EFFECT OF FLOW-TWISTING INTENSITY ON THE MIXING OF A HEAT-TRANSFER AGENT IN BUNDLES OF TWISTED TUBES

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UDC 621.564.71:536.423

Methods of mathematical statistics are used to generalize results obtained in studying the interchannel mixing of a heat-transfer agent in the inter-tube space of a heat exchanger with twisted tubes.

Intensification of heat and mass transfer by twisting the flow in bundles of twisted tubes makes it possible to achieve the maximum effect in high-temperature heat exchangers, since this method provides for intensive smoothing out of possible temperature variations in the cross section of the bundle [1-3]. Interchannel mixing of the heat carrier in bundles of twisted tubes was studied in a number of articles [1-3], and it was noted that the effective diffusion coefficient describing the intensity of lateral heat and mass transfer in such bundles exceeds the coefficients of turbulent diffusion along the axis of a round tube by more than an order of magnitude. Articles [1-3] deal with the effect of various parameters on this coefficient which, in dimensionless form can be presented as

$$K = D_t/(ud_e). \tag{1}$$

It was demonstrated in [1-3] that the numerical value of K might be dependent on the experimental research method. Thus, when using the heat-diffusion method based on the Lagrange description of a turbulent field in examining the history of the motion of an individual particle emitted from a point source, we find that K is almost twice as large as in the case in which we make use of the method of diffusion from linear heat sources. Moreover, it was demonstrated in the cited references that when the length of the twisted-tube bundle is

$$l > l_i = 8{,}019 d_e \mathrm{Fr}_s^{0,226}, \tag{2}$$

where ℓ_{in} is the length of the initial section required to obtain a stabilized temperature profile for the core of the flow, and thus to obtain a stabilized value for K_{st} which is the average value of K over the entire region of flow within the bundle, this average value is determined as part of the Euler description of the turbulent flow by comparison of the experimentally measured and theoretically derived temperature fields at the outlet section of the bundle, and this value is virtually equal to the stabilized value of the coefficient K_{st}. Since condition (2) in [1, 2] is satisfied, the authors of these articles noticed no effect of the level of entry flow turbulence (ε = 1-6%) on K, and the experimental data of the various authors were generalized by the following relationships:

for Re = $3.4 \cdot 10^3 - 10^4$

$$K_{st} = 3.1623\,[0.136\mathrm{Fr}_s^{-0,256} + 10\mathrm{Fr}_s^{-0,66}(m - 0.46)]\,\mathrm{Re}^{-0,125}, \tag{3}$$

for Re $\geq 10^4$

$$K_{st} = 0{,}136\,\mathrm{Fr}_s^{-0,256} + 10\mathrm{Fr}_s^{-0,66}(m - 0.46). \tag{4}$$

At the same time, a noticeable deviation from (3) and (4) is observed in the cited references for twisted-tube bundles 1.5 m in length, although this deviation falls within the limits of experimental error. This deviation is greater for bundles with small Fr_s numbers [2]. The Fr_s number is defined by

$$\mathrm{Fr}_s = s^2/(dd_e), \tag{5}$$

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 55, No. 5, pp. 709-715, November, 1988. Original article submitted July 16, 1987.

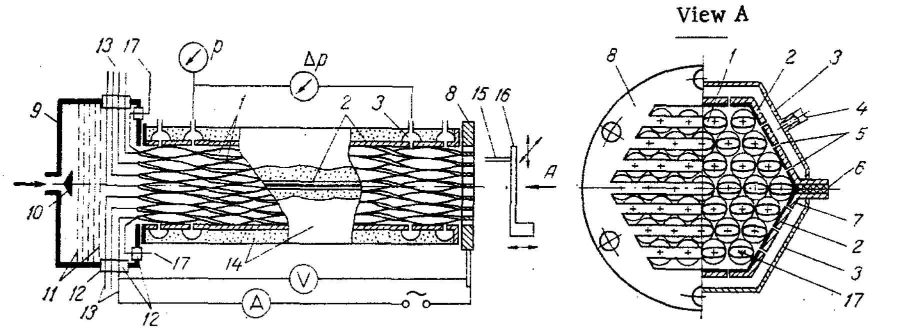

Fig. 1. Schematic of the experimental section: 1) twisted tubes; 2) housing; 3) collector; 4) static pressure sampling valve; 5) static pressure sensors; 6) sealing gasket; 7) Al_2O_3 electroinsulating coating; 8) outlet current lead; 9) inlet assembly; 10) conical flow splitter; 11) equalizing grids; 12) hermetically sealed inlet; 13) power leads; 14) thermally insulated jacket; 15) temperature and velocity heat sensors; 16) coordinating mechanism; 17) thermoelectric twisted-tube wall temperature sensors.

which is derived by using a model of quasisolid rotation [1], and which characterizes the intensity of heat-carrier flow twisting in the bundle. The nature of this deviation of K from K_{st} can be explained by the heat loss through the housing of the heat exchanger which in [2] has no thermal insulation. These losses increase with increase in mixing intensity (for the smaller values of the Fr_s number).

It therefore becomes necessary to conduct more precise experiments on interchannel mixing in twisted-tube bundles of various lengths, with minimum heat losses through the heat-exchanger housing wall, to refine the dependence of the coefficient K on the intensity of flow twisting (on the Fr_s number) for twisted-tube bundles exhibiting that heat-carrier porosity m = 0.501-0.544 most frequently encountered in actual practice.

The investigation of the mixing process in twisted-tube bundles was conducted on an experimental installation which is described in general outline in [1]. The experimental section housing the bundle is shown in Fig. 1. The housing of the 37-tube bundle has a horizontal release and is coated on the inside surface with aluminum oxide so as to electrically insulate the bundle from the housing. The outer surface of the housing is thermally insulated with an asbestos layer and a sheet of fiberglass, which virtually eliminates any loss of heat to the ambient medium. Air is used as the heat-transfer agent, and this air is fed into the experimental set-up through a system of equalizing grids designed to produce a uniform velocity field at the inlet to the bundle. Alternating current is used to heat the tubes. It was thus possible to simulate both axisymmetric and asymmetric fields of energy release in the bundle, heating individual groups of tubes. The temperature and velocity fields were measured at the outlet from the tubes by means of a thermocouple and a total-head sensor mounted on the coordinating mechanism. In addition, the escaping air flow was measured, as were the twisted-tube wall temperatures and the pressure drops across various sections of the bundle. The twisted tubes were 750 and 1000 mm in length, the maximum dimension of the oval profile was d = 12.33 mm, the wall was 0.2 mm thick, and the relative pitch of the twist was s/d = 6.5, 12.3, 12.9, and 26. Table 1 shows the geometric characteristics of the twisted-tube bundles.

TABLE 1. Geometric Characteristics of the Tube Bundles

Fr_s	$d\cdot10^3$, m	$d_e\cdot10^3$, m	s/d_e	l/d_e	m
63,6	12,30	8,15	9,82	92,02	0,544
232	12,32	8,06	18,86	93,05	0,539
286	12,16	7,11	22,11	140,7	0,501
1052	12,33	7,88	40,61	95,18	0,527

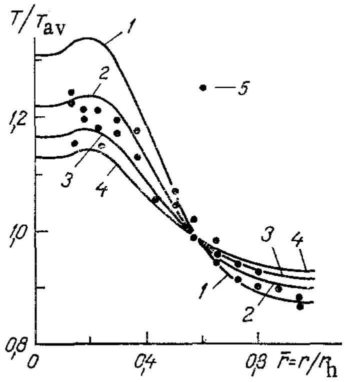

Fig. 2

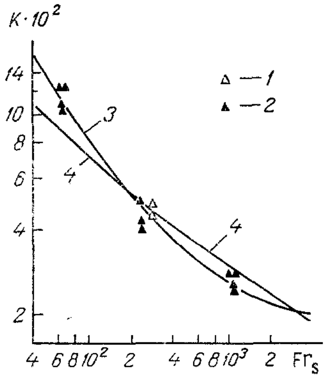

Fig. 3

Fig. 2. Comparison of experimental dimensionless temperature fields with calculation results for a bundle with Fr_s = 286 and ℓ = 1.0 m: 1-4) calculation with K = 0.03, 0.045, 0.06, and 0.075, respectively; 5) experimental data.

Fig. 3. Coefficient K as a function of Fr_s: 1, 2) experimental data for bundles with Fr_s = 286 and ℓ = 1 m; Fr_s = 63.6, 232, 1052, and ℓ = 0.75 m, respectively; 3) relationship (20); 4) relationship (4).

For purposes of determining the coefficient K, the experimentally measured temperature fields of the heating agent were compared with the theoretically calculated fields. The latter were calculated by solution of a system of equations describing the flow of a homogenized medium in x, r, φ coordinates which replaced the real flow in the bundle of twisted tubes [1, 3]:

$$\rho u \frac{\partial u}{\partial x} = -\frac{\partial p}{dx} + \frac{1}{r}\frac{\partial}{\partial r}\left(\rho r D_t \frac{\partial u}{\partial r}\right) + \frac{1}{r^2}\frac{\partial}{\partial \varphi}\left(\rho D_t \frac{\partial u}{\partial \varphi}\right) - \xi \frac{\rho u^2}{2d_e}, \tag{6}$$

$$\rho u c_p \frac{\partial T}{\partial x} = q_v \frac{1-m}{m} + \frac{1}{r}\frac{\partial}{\partial r}\left(\rho r c_p D_t \frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial}{\partial \varphi}\left(\rho c_p D_t \frac{\partial T}{\partial \varphi}\right), \tag{7}$$

$$G = m \int_0^{2\pi}\int_0^{r_h} \rho u r dr d\varphi, \tag{8}$$

$$p = \rho R T, \tag{9}$$

with the following boundary conditions:

$$u(0, r, \varphi) = u_{in}, \; T(0, r, \varphi) = T_{in}, \; p(0) = p_{in}, \tag{10}$$

$$\left.\frac{\partial u(x, r, \varphi)}{\partial r}\right|_{r=r_h} = 0, \quad \left.\frac{\partial T(x, r, \varphi)}{\partial r}\right|_{r=r_h} = 0, \tag{11}$$

$$u(x, r, \varphi) = u(x, r, \varphi + 2\pi), \; T(x, r, \varphi) = T(x, r, \varphi + 2\pi). \tag{12}$$

The system of equations (6)-(9) with boundary conditions (10)-(12) is solved numerically, and for its closure we use the experimentally derived coefficient K from (1) [1]. The method of solving a system of equations such as (6)-(9) is examined in [1].

Figure 2 shows a typical experimentally measured temperature field for a twisted-tube bundle with a relative twist pitch of s/d = 12.3 and length ℓ = 1.0 m, where the six central tubes are heated and these surround the unheated tube along the axis of the bundle; said temperature field is compared to the theoretically calculated temperature fields by solving the system of equations (6)-(9) for various values of the effective coefficient of diffusion. The methods of mathematical statistics were used to determine K, as was a modified method of least squares [1, 2]. The values of the coefficient K derived in this manner for various values of Fr_s, with bundle lengths of ℓ = 0.75 and 1 m and a porosity of m = 0.501-0.544 are shown in Table 2. Here we also find the results from the determination of the coefficient K both for the axisymmetric and asymmetric nonuniformities in heat supply for

TABLE 2. Parameters of the Experiment

Fr_s	Number of heated tubes	$Re \times 10^{-4}$	$q_s \cdot 10^2$ MW/m²	$K \cdot 10^2$	Fr_s	Number of heated tubes	$Re \times 10^{-4}$	$q_s \cdot 10^2$ MW/m²	$K \cdot 10^2$
63,6	6	1,52	2,22	10,6	232	15*	1,63	2,08	4,9
63,6	7	1,52	1,85	10,9	286	6	1,78	3,99	4,8
63,6	11*	1,54	2,14	12,3	286	6	1,79	3,70	4,4
63,6	11*	1,55	2,27	12,3	1052	9*	1,72	2,12	2,8
232	7	1,65	1,93	4,8	1052	9*	1,75	1,98	2,8
232	9*	1,65	2,04	3,9	1052	18	1,23	1,23	2,5
					1052	30	1,22	0,93	2,6

*Nonaxisymmetric location of heating zone in lateral cross section.

various numbers of heated twisted tubes. Here, all of the data for K presented in the table pertain to the turbulent flow region with Re = $(1.22\text{-}1.79)\cdot 10^4$, where in accordance with [1, 2] no effect of the Re number on the effective diffusion coefficient is observed. According to [1] the location of the heating zone also has virtually no effect on the coefficient K in the flow region separated from the housing wall through a distance (1-1.5)d. Therefore, bearing in mind that the porosity m for the subject bundles varied insignificantly, we find that the entire compilation of experimental data pertaining to K, as shown in Table 2, can be utilized to derive the relationship between this coefficient and the twisting intensity of the twisted tubes (i.e., the Fr_s number), by employing the methods of mathematical statistics.

The coefficient K as a function of Fr_s can be shown in the form of a quadratic polynomial in logarithmic coordinates:

$$\lg K = A_0 + A_1 \lg Fr_s + A_2 (\lg Fr_s)^2. \tag{13}$$

Selection of the quadratic polynomial (13) to describe processes of heat and mass transfer in a bundle of twisted tubes, which, after simple transformations, leads to the approximating relationship

$$K = 10^{A_0} Fr_s^{A_1 + A_2 \lg Fr_s}, \tag{14}$$

is associated with the complex nature of the flow in such bundles and the existence of additional transfer mechanisms, as opposed to straight tubes for which linear relationships are used to describe similar processes. For purposes of describing data pertaining to mixing in a bundle of twisted tubes and to demonstrate the validity of this selection, let us also examine a linear relationship of the form

$$\lg K = B_0 + B_1 \lg Fr_s. \tag{15}$$

The coefficients A_0, A_1, and A_2 in (13) were calculated with a computer by the method of least squares [4]. To estimate the agreement of the approximating relationship to the bulk of the experimental data, we employed a coefficient of multiple correlation [5]:

$$R^2 = 1 - \left[\sum_{i=1}^{n} (Y_{iex} - Y_i)^2 \Big/ \sum_{i=1}^{n} (Y_{iex} - \bar{Y}_{ex})^2\right], \tag{16}$$

where Y_{iex} is the experimental value of the studied parameter; Y_i is the calculated value according to the approximating relationship; $\bar{Y}_{ex}$ is the average value of the parameter from experimental data:

$$\bar{Y}_{ex} = \sum_{i=1}^{n} Y_{iex} / n, \tag{17}$$

where n is the number of experimental data (experimental values of the parameter). To evaluate the advantages of utilizing an approximating relationship of the selected type, (13) for example, in comparison to the mean value of the parameter in the studied range of changes in the argument, we made use of the Fisher criterion [5]:

$$F = [(n-k)/(k-1)]\left[\sum_{i=1}^{n} (Y_{iex} - \bar{Y}_{ex})^2 - \sum_{i=1}^{n} (Y_{iex} - Y_i)^2\right] \Big/ \sum_{i=1}^{n} (Y_{iex} - Y_i)^2, \tag{18}$$

where k is the number of degrees of freedom in the regression equation (the approximating relationship). Use of the Fisher criterion to compare dispersion of the experimental data relative to an approximating relationship of the form (13) with an average value (17) reduces primarily to comparison of the tabulated value for F_{cr} for the assumed significance level 0.01 and to the assumption of a "null hypothesis" for the dispersions and, consequently, for the equivalence of describing the experimental results with an average value for the parameter and for the selected approximating relationship. Moreover, having calculated the magnitude of the Fisher criterion F for two different approximating relationships (13) and (15), we can evaluate the advantage of either relationship on the basis of the relationships between the magnitudes of the F criterion. The quantity F_{cr} is taken from the tables in [5] in accordance with the assumed significance level (0.01) and the number of degrees of freedom for the dispersion of the experimental data relative to the mean value of $\bar{Y}_{ex}$ · (n − 1) as well as relative to the approximating relationship (n − q, where q is the number of determined coefficients in the relationship).

To estimate the scattering of the experimental data, we calculate the root-mean-square deviation σ of the experimental values for the parameter Y from relationships (14) and (18):

$$\sigma = \left[\sum_{i=1}^{n} (Y_{iex} - Y_i)^2/n\right]^{0,5}. \tag{19}$$

Here the quantity σ, for ease of analysis, was normed to the minimum calculated value of the parameter K according to the approximate relationship. Then the experimental data for the coefficient K (Table 2) with the quadratic polynomial (13) can be described as a function of the Fr_s number:

$$K = 10.35 Fr_s^{-1,4232+0,1857 \lg Fr_s}, \tag{20}$$

for which the quantities σ, F, and R^2 are: $\sigma \le \pm 22\%$; F = 225 and $R^2 = 0.978$. If these same experimental data are generalized by linear relationship (15), we derive the expression

$$K = 0.904 Fr_s^{-0,5171}, \tag{21}$$

for which $\sigma \approx \pm 36\%$, F = 197, and $R^2 = 0.947$. Comparison of the quantities σ, F, and R^2 for (20) and (21) demonstrates the advantage of approximation (13), which describes the experimental data with greater accuracy and to a greater degree corresponds to the bulk of the experimental data, since the values of F and R^2 for relationship (20) are greater than the values of F and R^2 for relation (21).

Relationship (20), shown in Fig. 3, is a good generalization of the experimental data for twisted-tube bundles of various lengths for the range of porosity m variation encompassed by the experiment. For purposes of comparison, Fig. 3 shows relationship (4) at an average value for the porosity of the tube bundles used in this study (see Table 1). We see that the values of K, calculated by means of (20), lie above the values of K determined from (4), with particularly noticeable divergence observed for the small Fr_s numbers at which the greatest intensification of the interchannel heat-carrier mixing is observed. This divergence can be explained by the fact that in this study the housing of the bundle of twisted tubes was thermally well insulated and the heat losses from the experimental section were negligibly small, which is borne out by the agreement in values of the coefficient K for bundles of varying lengths.

The derived relationship (20) can be recommended for calculation of the coefficient K in twisted-tube bundles with varying intensities of flow twisting and with a porosity m = 0.501-0.544 for Reynolds numbers Re > 10^4 and bundle lengths $\ell > \ell_{in}$. The greatest intensification of the process of interchannel mixing in the subject bundles is observed at small Fr_s numbers from 63 to 286. Relationship (20) allows us to close the system of equations (6)-(9) and thermohydraulically to design a heat exchanger with twisted tubes in which allowance is made for interchannel mixing.

NOTATION

D_t, effective diffusion coefficient; u, velocity; d_e, equivalent diameter; ℓ, length of bundle; Fr_s, criterion characterizing the flow-twisting effect; K, dimensionless effective diffusion coefficient; s, twisting pitch of the twisted tubes; d, maximum dimension of oval tube profile; x, r, φ, coordinates; ξ, drag coefficient; q_V, density of volumetric heat release; T, temperature; ρ, density; c_p, specific heat capacity; p, pressure; R, gas constant; σ, root-mean-square deviation; F, Fisher criterion; R^2, multiple correlation coefficient; Re, Reynolds number; m, porosity of the bundle relative to the heat-transfer agent; G, mass flow rate of the heat-transfer agent; q_s, heat flow density. Subscripts: st, stabilized; i, initial; h, housing; in, inlet.

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RADIANT—CONVECTIVE HEAT EXCHANGE IN TURBULENT MOTION OF A GAS SUSPENSION WITHIN A TUBE

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UDC 536.242.001.5

A calculation of the temperature field and radiant and convective components of the thermal flux density is performed for combined action of convection and radiation in a dusty gaseous medium.

At present there are available a large number of studies of the process of radiant—convective heat exchange [1-4]. However, for the case of flow of a gas suspension in a round tube this problem has been considered only in [5, 6]. Many questions such as the effect on heat exchange of the direction of the thermal flux, the parameters of the carrier gas and particles, and temperature conditions require further study.

In the general case radiant—convective heat exchange is described by a system of equations in which the energy equation is an integral-differential one. Numerical solution of the problem is possible only with significant expenditures of machine time, so that development of simple but reliable methods for engineering calculations of the radiant component of the thermal flux density on the tube surface during motion of a dusty gas therein is a problem of practical value.

The present study will present a simplified method and results of calculating radiant—convective heat exchange for flow of a gas suspension in a circular tube.

Relying on [7], we will assume that the solid particles found in the gas suspension flow are uniformly distributed over the tube section. The gas suspension is considered as a quasihomogeneous absorbing and radiating grey medium. Temperature difference between gas and particles will be neglected, as well as the effect of these temperatures on convective heat exchange. The latter assumption is satisfied well for tubes of small diameter if the particle mass flow concentration does not exceed the value two [8]. We will consider the flow of the gas suspension in a region far removed from the tube entrance. The tube wall is absolutely black. On the wall the boundary condition is q_W = const. Following [7], we write the energy equation for the gas suspension in the form

$$\rho_* w_x \frac{\partial h_*}{\partial x} = \frac{1}{r} \frac{\partial}{\partial r}(rq) + \eta_{\text{res}}. \quad (1)$$

Moscow Energy Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 55, No. 5, pp. 716-721, November, 1988. Original article submitted July 9, 1987.

