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FEATURES OF HEAT AND MASS TRANSFER IN BUNDLES OF SPIRAL TUBES

B. V. Dzyubenko

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An analysis of published data on heat and mass transfer and a study of the process of equalization of velocity-field nonuniformity due to an inlet pipe are used to make recommendations on calculation of the velocity and temperature fields in bundles of spiral tubes.

Heat exchangers in which the fluid flow is twisted by spiral tubes of oval cross section are distinguished by favorable dimensional characteristics thanks to the intensification of heat transfer as the heat-transfer agent flows in the space between tubes and inside the tubes [1]. Such heat exchangers are also characterized by intensive equalization of temperature- and velocity-field nonuniformities in the cross section of the tube bundle [2-5]. The velocity-field nonuniformities may be the result of a nonuniform heat-carrier temperature field due in turn to a specified distribution of heat supply about the radius of the bundle or to inlet pipes bringing the heat carrier into the intertube space. The process of equalization of these nonuniformities for an axisymmetric problem is described in the equation of motion by the diffusion term $1/r\partial/\partial r(r\bar{\rho}D_t\partial\bar{u}/\partial r)$ and the term $\xi\bar{\rho}\bar{u}^2/2d_e$ characterizing the effect of sources of fluid resistance:

$$\bar{\rho}\,\bar{u}\,\frac{\partial\bar{u}}{\partial z}=-\frac{dp}{dz}+\frac{1}{r}\,\frac{\partial}{\partial r}\left(r\bar{\rho}D_t\,\frac{\partial\bar{u}}{\partial r}\right)-\xi\,\frac{\bar{\rho}\,\bar{u}^2}{2d_e}. \tag{1}$$

Here, we use a flow model in which the bundle is replaced by a porous mass with a diameter equal to the bundle diameter. The mass contains a flow of a homogenized medium — the flow of heat carrier with sources of volumetric energy release q_v and fluid resistance $\xi\bar{\rho}\bar{u}^2/2d_e$ distributed in it. The intensity of these sources changes over the radius of the bundle [3]. Having determined the displacement thickness of the boundary layer δ^* and having hypothetically accumulated a layer of material of equal thickness δ^* on the tube wall, we can examine a free flow in new boundaries with slip of the homogenized medium. Here, we assume that the velocity vector is parallel to the bundle axis and that $\partial p/\partial r = 0$. Thus, in Eq. (1) $\bar{u}$ is the velocity in the core of the flow (outside the boundary layer), while there are no convective terms with transverse velocity components in the left side of (1). The diffusion term in (1) accounts for the effect of different transport mechanisms on the velocity field in cross sections of the bundle [3]. Thus, replacement of the flow in an actual tube bundle by a flow of a homogenized medium is an engineering method which should be substantiated experimentally for use in calculating the velocity and temperature fields of the heat carrier.

One of the goals of the present study is to experimentally validate equation of motion (1) in the case of flow being examined, as well as to analyze data from different investigations of heat and mass transfer in bundles of spiral tubes.

To calculate the velocity and temperature fields in a bundle of coiled tubes, it is necessary to solve a system of differential equations which, apart from (1), includes equations of energy, flow rate, and state:

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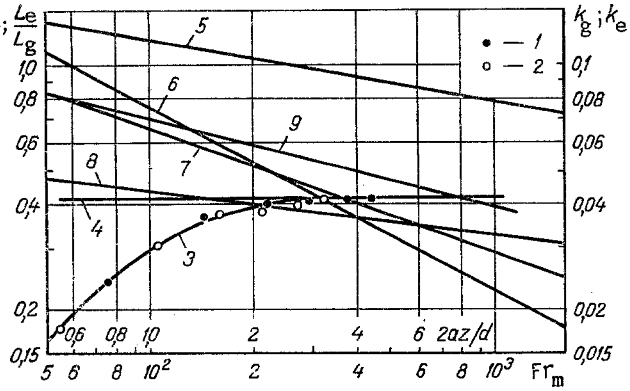

Fig. 1. Effect of the number Fr_m on characteristics of heat and mass transfer in bundles of spiral tubes: 1-3) change in the standard deviation of the temperature distributions relative to the transformed longitudinal coordinate; 1) test data from [2] for $Fr_m = 314$; 2) same for $Fr = 1530$; 3) curve generalizing the test data; 4-9) Eqs. (19), (11), (10), (13), (15), and (20), respectively.

$$\bar{\rho}\,\bar{u}c_p\frac{\partial\bar{T}}{\partial z}=q_v\frac{1-m}{m}+\frac{1}{r}\frac{\partial}{\partial r}\left(r\bar{\rho}c_pD_t\frac{\partial\bar{T}}{\partial r}\right), \tag{2}$$

$$G=2\pi m\int_0^{r_k}\bar{\rho}\,\bar{u}r\,dr, \tag{3}$$

$$p=\bar{\rho}R\bar{T}. \tag{4}$$

Equations (1) and (2) ignore the transfer of momentum and heat by molecular diffusion, heat liberation during dissipation, and the kinetic energy of the flow and turbulent diffusion in the longitudinal direction. The effect of homogenization is taken into account in (2) by introduction of the multiplier (1 − m)/m. The problem is solved by numerical methods [3] with the boundary conditions:

$$\bar{T}(0,\ r)=\bar{T}_{in},\quad \bar{u}(0,\ r)=\bar{u}_{in},\quad p(0,\ r)=p_{in}, \tag{5}$$

$$\left.\frac{\partial\bar{T}(z,\ r)}{\partial r}\right|_{r=r_k}=0,\quad \left.\frac{\partial\bar{u}(z,\ r)}{\partial r}\right|_{r=r_k}=0, \tag{6}$$

$$\left.\frac{\partial\bar{T}(z,\ r)}{\partial r}\right|_{r=0}=0,\quad \left.\frac{\partial\bar{u}(z,\ r)}{\partial r}\right|_{r=0}=0, \tag{7}$$

if the coefficients D_t and ξ, determined by experiment, are known. Numerical values of the coefficient D_t or the dimensionless value

$$k=\frac{D_t}{u_{av}d_e} \tag{8}$$

can be established by different methods [2-5].

In determining these coefficients, it is necessary to consider features of heat and mass transfer in bundles of spiral tubes connected with features of their design [1]. It is also necessary to consider the method of calculation used. Turbulence is generated in such bundles not only by a stationary wall but also due to the interaction of the flows in the helical and through channels of the bundle. These flows have different velocities and directions [6]. Additional agitation of the flow occurs when it moves past sites where the tubes are in contact with each other. Thus, the turbulence intensity in bundles of spiral tubes is several times greater than on the axis of a straight, circular tube [6]. The level of turbulence increases with a decrease in the relative pitch S/d or the number

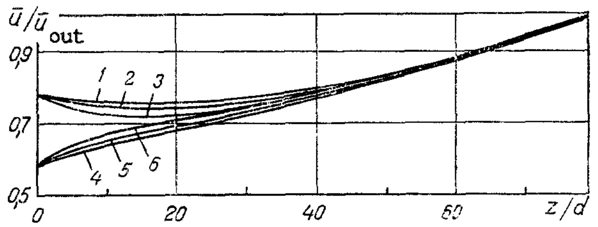

Fig. 2

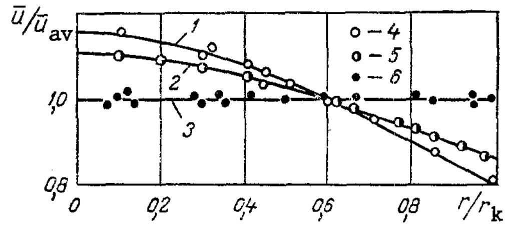

Fig. 3

Fig. 2. Change in velocity along a bundle of spiral tubes: 1-3) calculation with $\bar{r} = 0$ and $k_e \approx 0$, 0.09, and 0.625, respectively; 4-6) same with $\bar{r} = 1$.

Fig. 3. Comparison of experimental data with results of calculation of velocity fields with $k_e = 0.03$: 1) velocity profile (21) at z = 0; 2, 3) calculation for z/d = 7.7 and 79.5; 4-6) test data for z/d = 0, 7.7, and 79.5, respectively.

$Fr_m = S^2/dd_e$. Apart from turbulent mixing of the heat carrier, a significant contribution is made to the coefficient D_t by convective transfer on the scale of the cells of the bundle resulting from the presence of transverse velocity components [6]. The effect of this transport mechanism is taken into account empirically through a certain coefficient D_t, since system (1-4) does not reflect the three-dimensional character of the flow typical of an actual bundle of spiral tubes. In the description of flow by means of system (1)-(4), the coefficient D_t also accounts for the organized, forced transfer of heat carrier from cell to cell over the radius and azimuth of the bundle along the helical channels relative to the axis of the tubes. The above-mentioned mechanisms of heat and mass transfer lead to equalization of the flow parameters in the cross section of the bundle and are accounted for in the equations of energy and motion by diffusion terms.

The study [4] generalized test data of different authors on the coefficient k for bundles of rods with spiral fins and proposed the following relation for the numbers $Re \geq 10^4$

$$k_e = 1{,}902\,Fr_m^{-0{,}53} m^{1{,}086}, \tag{9}$$

which is valid for the range of numbers Fr_m = 43-3300 and $m \geq 0.27$. This formula agrees satisfactorily with experimental data on the coefficient k_e for bundles of helical tubes obtained by the method of heat diffusion from a system of linear sources based on an Eulerian description of turbulent flow [3-5]. In accordance with this method, with a nonuniform heat-release field in the outlet section of the bundle, a comparison is made between experimental and theoretical solutions of system (1)-(4) with boundary conditions (5-7) for the temperature field, and the most reliable values of the coefficient k_e are determined. The effect of the Reynolds number on k_e at $Re < 10^4$ was established in [5]. Then the formula for calculating this coefficient has the form

$$k_e = 3{,}1623\,(1{,}902\,Fr_m^{-0{,}53} m^{1{,}086})\,Re^{-0{,}125}. \tag{10}$$

Equation (10) is compared in Fig. 1 at $Re = 8\cdot10^3$ and m = 0.475 with the expression

$$k_g = 0{,}0356\,(1 + 8{,}1\,Fr_m^{-0{,}278}), \tag{11}$$

obtained in [2] by the heat diffusion method based on a statistical Lagrangian description of a turbulent field in a study of the history of motion of individual particles continuously generated by a point source. It is evident that if the difference between k_e and k_g is small at small Fr_m, then the difference will increase with an increase in Fr_m. If, following [5], we assume that

$$k_e/k_g = L_e/L_g, \tag{12}$$

then in the interval Fr_m = 50-1500 we obtain the following for the ratio of the spatial scales of turbulence:

$$L_e/L_g = 3{,}338\,Fr_m^{-0{,}356}. \tag{13}$$

A similarity equation was obtained in [5] for k_e which is valid for the narrower range Fr_m = 55-1080 than Eq. (9) and which has the form

$$k_e = 3{,}1623\,[0{,}136\,Fr_m^{-0{,}256} + 10\,Fr_m^{-0{,}66}(m - 0{,}46)]\,Re^{-0{,}125}. \tag{14}$$

Using (14) and (11), we have [5]:

$$L_e/L_g = 0{,}785\,Fr_m^{-0{,}127}. \tag{15}$$

It can be seen from (13) and (15) that a decrease in Fr_m is accompanied by an increase in L_e/L_g (Fig. 1), since there is an increase in the intensity of turbulent pulsations in this case [6]. This result is similar to that obtained by Michelson [7], who found that the ratio L_e/L_g on the axis of a circular tube increases with an increase in fluctuation velocity.

The appreciable difference between k_g and k_e in Fig. 1 at large Fr_m may be connected with the use of the method of diffusion from a point source of heat for a bundle of spiral tubes, where the source had finite dimensions [2]. For a source of heat diffusion of finite dimensions, the temperature distributions at different distances from the source have the form [2]:

$$\frac{\overline{T}-\overline{T}_0}{\overline{T}_m-\overline{T}_0}=\exp\left(\frac{\overline{r}_0^2-\overline{r}^2}{8\overline{\overline{y}}^2}\right)f(\overline{r},\ \overline{r}_0,\ \overline{\overline{y}}^2), \tag{16}$$

$$f(\overline{r},\ \overline{r}_0,\ \overline{\overline{y}}^2)=\frac{\overline{r}^6\overline{r}_0^6+576\,(\overline{\overline{y}}^2)^2\,\overline{r}^4\overline{r}_0^4+147{,}5\cdot10^3\,(\overline{\overline{y}}^2)^4\,\overline{r}^2\overline{r}_0^2+9{,}47\cdot10^6\,(\overline{\overline{y}}^2)^6}{\overline{r}_0^{12}+576\,(\overline{\overline{y}}^2)^2\,\overline{r}_0^8+147{,}5\cdot10^3\,(\overline{\overline{y}}^2)^4\,\overline{r}_0^4+9{,}47\cdot10^6\,(\overline{\overline{y}}^2)^6}, \tag{17}$$

where $\bar{r} = 2r/b$, $\bar{r}_0 = 2r_0/b$, $\bar{\bar{y}}^2 = \bar{y}^2/b^2$, is different from which Gaussian distribution (at $\bar{r}_0 = 0$)

$$(\overline{T}-\overline{T}_0)/(\overline{T}_m-\overline{T}_0)=\exp(-r^2/2\overline{\overline{y}}^2). \tag{18}$$

The dimensionless standard deviation σ/b of distribution (16) increases along the stream issuing from the source (Fig. 1) and only at the distances $2az/d \geq 2.56$ is there an asymptotic approach to σ/b for distribution (18), which is equal to

$$\sigma/b=0{,}423=\text{const}\,(2az/d). \tag{19}$$

In (19), a is the structure factor of the stream. It was determined empirically in [8]:

$$a=0{,}0745+11{,}37\,\text{Fr}_m^{-1}+246\,\text{Fr}_m^{-2}.$$

It is evident that the greater the number Fr_m, the longer the bundle must be for a source of finite dimensions to be taken as a point source in determining the coefficient k_g. Given the same length of a bundle of coiled tubes but different numbers Fr_m [2], the accuracy of k_g will be less at large Fr_m. Here, use of the method in [2] will lead to some overstatement of k_g. At the same time, values of k_e calculated for bundles of coiled tubes from (9) and (10) are roughly 1.5 times less than the experimental data for $Fr_m = 1050$ [3]. However, these values are within the confidence interval for the experimental value of k_e. Thus, if on the basis of comparison of k_e and k_g we take empirical data on k_e from [3], where $k_e = 0.10$ for $Fr_m = 64$, $k_e = 0.053$ for $Fr_m = 232$, and $k_e = 0.030$ for $Fr_m = 1050$, then the ratio L_e/L_g will accordingly be equal to 0.787, 0.535, and 0.387 and the interpolational relation for L_e/L_g will have the form (Fig. 1):

$$L_e/L_g=2{,}263\,\text{Fr}_m^{-0{,}254}. \tag{20}$$

In the present case we established a mean value $L_e/L_g \approx 0.6$ for the range $Fr_m = 64$-1050, as in the Michelson tests [7]. It was discovered in the latter tests that the mean value is $L_e/L_g \approx 0.6$ on the axis of a circular tube in the range $Re = 2\cdot10^5$-$6\cdot10^5$. This could be expected, since the velocity profiles in the boundary layer in bundles of coiled tubes in the above range of Fr_m and with $Re \approx 10^4$ are described by the same power laws as the velocity profiles in a circular tube at $Re \geq 10^5$ [9]. Thus, the temperature and velocity fields can be calculated by using empirical data on the coefficient k_e from [3] or Eqs. (9) and (10), which describe the test data of different authors with a confidence level of 0.95.

The use of system of differential equations (1)-(4) to determine the mutual effect of the temperature and velocity fields with a nonuniform field of heat release over the radius of the bundle was illustrated in [10]. The same study showed that with a uniform velocity field in the flow core, a nonuniform heat-release field at the bundle inlet forms nonuniform velocity fields. The latter fields are partially equalized along the bundle. A certain degree of velocity nonuniformity remains at the bundle outlet over the radius of the heat exchanger. Meanwhile, the theoretical and experimental velocity distributions in the flow core agree with each other, which confirms the validity of Eqs. (1-4).

The use of Eq. (1) can also be substantiated by studying the process of equalization of velocity-field nonuniformity caused by an inlet pipe in an adiabatic flow of air. The results of such a study are presented here. Tests were conducted on heat-exchanger models with 127 spiral tubes of oval profile. The relative spacing of the tubes S/d = 16, while

$Fr_m = 470$. We used the experimental unit described in [1]. The bundle inlet was axisymmetric. The nonuniformity of the velocity field was created by a system of inlet grids. The level of turbulence after the grids was 6%. Flow velocity was measured in the outlet sections of bundles of different lengths by a pitot tube. The pitot tube had a low sensitivity to the downwash angle of the flow up to ±20° [9]. The length of the bundles corresponded to distances from the inlet of 11d, 18.7d, and 90.5d. Here, the inlet conditions were constant, the number $Re \approx 10^4$, and $\bar{T}_{in} = 305°K$. The standard deviation of velocity was 3%.

The velocity profile for the flow core at the distance 11d turned out to be described by a zeroth-order Bessel function of the first kind

$$\bar{u}_{in}/\bar{u}_{av} = 0{,}8 + 0{,}35\,J_0(2{,}405\,\bar{r}), \tag{21}$$

where $\bar{r} = r/r_k$. This bundle section was taken as the initial section in performing the theoretical calculations. The calculations were performed by solving system (1-4) with boundary conditions (5-7) by the grid method and the explicit scheme in [3]. Figures 2 and 3 show results of calculation of the velocity fields over the length and radius of the bundle. Figure 2 shows curves of change in velocity along the bundle for $\bar{r} = 0$ and $\bar{r} = 1$ with different values of the coefficients $k_e \approx 0$, 0.09, and 0.625. It is evident that the initial velocity nonuniformity is equalized at $z \approx 70d$ for $k_e \approx 0$, at $z = 50d$ for $k_e = 0.09$, and at $z = 32d$ for $k_e = 0.625$. These findings confirm the need to use the equation of motion in the form (1).

Results of calculations in Fig. 3 performed for $k = 0.03$ agree well with test data for the bundle cross sections $z = 7.7d$ and 79.5d, which are represented as the most probable values of velocity for each value of r obtained by statistical analysis of a large volume of experimental data [9].

It should be noted that there is little stratification of the theoretical curves $\bar{u} = u(r)$ with different values of k_e, particularly in the range of coefficients $k \leq 0.09$. This shows that it is not possible to determine this coefficient by comparing experimental and theoretical velocity fields, as is done in determining temperature fields. In the latter case, there is significant stratification of the theoretical curves [3].

The diffusion term in Eq. (1) may makes its greatest contribution during transience. According to the data in [11], with an abrupt increase in the thermal load and a constant flow rate, the nonsteady heat-transfer coefficient k_n may be several times greater than the quasisteady coefficient k_e determined from Eqs. (9) and (10). The relationship between the two coefficients here is determined by the formula

$$\varkappa = k_n/k_e = 0{,}81\cdot 10^{-4}\mathrm{Fo}_b^{-2} - 0{,}978\cdot 10^{-2}\mathrm{Fo}_b^{-1} + 1{,}21, \tag{22}$$

where

$$\mathrm{Fo}_b = \lambda_b\tau/(c_p\rho_b d_k^2). \tag{23}$$

The effect of distributed fluid resistance $\xi\rho\bar{u}^2/2d_e$ during equalization of velocity nonuniformity can be evaluated as follows. We will assume that the diffusion term in (1) has the deciding effect on this process. Then, by solving the equation

$$\frac{\partial\Delta\bar{u}}{\partial z} = \frac{D_t}{\bar{u}}\left(\frac{\partial^2\Delta\bar{u}}{\partial r^2} + \frac{1}{r}\,\frac{\partial\Delta\bar{u}}{\partial r}\right) \tag{24}$$

with the boundary conditions

$$\Delta\bar{u}(0,\ r) = \Delta\bar{u}_m J_0(2{,}405\bar{r}) = f(r), \tag{25}$$

$$\Delta\bar{u}(z,\ r)|_{r=r_k} = 0, \tag{26}$$

$$\Delta\bar{u}(z,\ r) \text{ is finite}, \tag{27}$$

by the method of separation of variables we obtain the general solution

$$\Delta\bar{u}(z,\ r) = \sum_{n=1}^{\infty} A_n J_0(\mu_n\bar{r})\exp\left[-\frac{D_t z}{\bar{u}}\left(\frac{\mu_n}{r_k}\right)^2\right], \tag{28}$$

where

$$A_n = \frac{\int\limits_0^{r_k} \overline{T}(r) J_0(\mu_n \overline{r}) r dr}{\int\limits_0^{r_k} J_0^2(\mu_n \overline{r}) r dr}. \quad (29)$$

With allowance for (25), we obtain $A_1 = 0.35$, $A_2 = A_3 = \ldots = A_n = 0$. Finally, the general solution of Eq. (28) reduces to the form

$$\frac{\Delta \overline{u}(z, r)}{\overline{u}_{av}} = 0{,}35\, J_0(2{,}405\, \overline{r}) \exp\left[-\frac{D_t z}{\overline{u}_{av}} \left(\frac{2{,}405}{r_k}\right)^2\right]. \quad (30)$$

Comparing the theoretical and experimental velocity distributions, we find the value of the coefficient k. In the present case, k = 0.625. However, in reality for this bundle k_e = 0.03 according to (10). Thus, the term in Eq. (1) which accounts for the equalizing effect of fluid resistance is important.

The above study made it possible to substantiate system of differential equations (1-4) and to make recommendations on calculating velocity and temperature fields in bundles of spiral tubes with a prescribed velocity profile at the inlet due to an inlet pipe.

NOTATION

D_i, effective coefficient of turbulent diffusion; z, r, longitudinal and radial coordinates; r_0, radius of the diffusion source; ρ, density; $\overline{u}$, velocity in the flow core; p, pressure; $\overline{T}$, temperature in the flow core; ξ, fluid resistance coefficient; d_e, equivalent diameter; c_p, specific heat; q_v, density of bulk heat sources; m, porosity of bundle in regard to the heat carrier; d, maximum size of tube oval; S, pitch of spiral tube; Fr_m, criterion characterizing features of flow in a bundle of spiral tubes; Re, Reynolds number; L, spatial scale of turbulence; $\bar{y}_2$, statistical mean square of displacement; Fo_b, Froude number; λ, thermal conductivity; τ, time; $d_k = 2r_k$, diameter of bundle; b, doubled median radius of the stream (at the point where $(\overline{T} - T_0)/(T_m - T_0) = 0.5$); $\Delta\overline{u}$, excess velocity in the flow core. Indices: in, inlet; av, mean value in a cross section; e, with Eulerian description of turbulent flow; g, with Lagrangian description of turbulent flow; m, maximum; n, nonsteady; b, at the mean-mass temperature.

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CONVECTIVE HEAT TRANSFER IN BOUNDED-VOLUME CHAMBERS OF VARIOUS CONFIGURATIONS

É. M. Malaya

UDC 536.24:536.41

The results are given from a comprehensive study of the heat transfer, aerodynamics, and combustion processes in regions of separation, reattachment, and subsequent evolution of flow in channels and chambers of various configurations.

The investigation of one of the most widespread techniques of combustion stabilization, namely by rational organization of the recirculation zones, presents a complex but timely theoretical and experimental problem. To determine the heat-transfer laws in bounded volumes we have carried out comprehensive studies of the aerodynamics (Fig. 1), heat transfer (Fig. 2), and combustion process (Fig. 3) in chambers of various configurations. The results can be analyzed to solve problems encountered in flow with recirculation zones and to show that the combustion and heat transfer in industrial plants of various configurations are intimately related to the aerodynamic and turbulent structure of the flow. We have investigated flows in cylindrical chambers in the form of sudden plane expansions (one-sided and two-sided steps with exit flow from one and three slots).

An important consequence of flow reattachment is a strong variation of the flow structure in the form of decay and scale reduction of the vortices. In the downstream zone from the reattachment point a sharply defined boundary is preserved as a result of the flow perturbation during separation and interaction with the separation zone. The combustion process is characterized by the correlation of the fluctuations $\overline{v't'}$.

It is important to note that the intensity of the velocity fluctuations varies insignificantly near the walls of chambers of different configurations in all operating regimes in comparison with its variation at the separation boundary; this fact further underscores the stability of wall flow against perturbations.

We have established the similarity of the turbulent transfer characteristics. The greatest structural changes and deviations from the uniform-flow heat-transfer laws for the investigated types of chambers take place immediately after the flow reattachment point in the zone for a symmetrical sudden plane expansion $\bar{x} = x/d_0{}^h = 18\text{-}20$, for flow from three slots $\bar{x} = x/d_0{}^h = 42\text{-}45$, and for cylindrical expansions $\bar{x} = x/d_0{}^h = 2.5\text{-}6$.

The equilibrium state of the flow sets in nonmonotonically, and the amplification of turbulent energy dissipation in the separation zone promotes an increase in heat transfer, as shown in Fig. 3. The maxima for $\overline{U'V'}$ (see [2]) and $\overline{v't'}$ practically coincide and are explained by the influence of the previous history of interaction of the flow with the separation zone. The following relations are recommended for engineering calculations (in the case of plane and axisymmetrical channels; Fig. 4, curves 1 and 2):

$$\log(\overline{v't'})\, m = x \log \alpha. \tag{1}$$

An analysis of the results has led to a number of conclusions characterizing the fundamental laws of flow in plane and cylindrical abrupt expansions prior to the onset of self-similarity for Re > 1000 for the axisymmetrical case and for Re $\leq$ 1000 in the self-similar regime in the plane case.

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