NONSTATIONARY HEAT AND MASS TRANSFER WITH A REDUCTION OF THE HEAT LOAD IN A HEAT EXCHANGER WITH TWISTED TUBES

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The results of an investigation of the nonstationary mixing of the heat-transfer agent accompanying the reduction of the heat load in a bundle of twisted tubes are presented, and a generalizing dependence for calculating the effective coefficient of diffusion is derived.

The problems of nonstationary heat and mass transfer in channels are now of great interest, since transient processes associated with a change in the operating conditions, actuation or stopping of heat-exchangers, can in many cases have a deciding effect on their safe operation. The results of investigations of nonstationary heat transfer in circular pipes are presented in [1]. Stationary heat- and mass-transfer processes [2] as well as nonstationary mixing of the heat-transfer agent with increasing heat load [3-6] have been investigated quite fully for channels with a complicated shape, formed by bundles of twisted tubes. In so doing it was found that nonstationary transport processes in bundles of twisted tubes have a number of peculiarities, linked with the solutions of constructional problems associated with them. Investigations of the mixing of the heat-transfer agent with increasing thermal power and constant flow rate of the agent showed that the temperature fields in the agent are restructured within the first 10-20 sec, leading to intensification of transfer processes as compared with quasistationary transfer [3-6]. When the thermal load is reduced rapidly to zero with the maximum value of the derivative of the power with respect to time $(\partial N/\partial\tau)_m$ = 7.5-10 kW/sec, it was observed in [3-6] that the intensity of the mixing process drops during the first moments compared with the quasistationary operating conditions. A criterional dependence was derived for nonstationariness of this type for calculating the dimensionless effective diffusion coefficient $K_n = D_{tn}/ud_e$ in a bundle of twisted pipes with the number $Fr_m = S^2/dd_e = 220$, where S is the pitch of the twisting of the oval profile of the tubes:

$$K_n/K_{qs} = 0.454\cdot 10^{-5}Fo_b^{-2} - 3.86\cdot 10^{-3}Fo_b^{-1} + 1.24. \quad (1)$$

In the expression (1) the quasistationary value of the effective diffusion coefficient K_{qs} is determined by the dependences in [7], while the Fourier criterion Fo_b is given by the expression

$$Fo_b = \frac{a\tau}{d_k^2} = \frac{\lambda_b\tau}{c_p\rho_b d_k^2}. \quad (2)$$

In this paper the data of [3, 6] and new results of an investigation of nonstationary mixing in a bundle of twisted tubes with $Fr_m = 57$, obtained with a reduction of the thermal load and a transition into the reduced power regime are generalized in order to determine the criterional dependence for calculating the dimensionless effective diffusion coefficient K_n with this type of nonstationariness.

The experimental investigations of nonstationary heat and mass transfer were performed on the experimental setup described in [3, 6] by the method of diffusion from a system of linear heat sources. The central zone, consisting of 37 tubes, in a bundle of 127 twisted tubes was heated with an electric current. The temperature distribution in the heat-transfer agent was measured in the output section of the bundle, which was 0.5 m long, with the help of a rack of 10 Chromel—Alumel thermocouples with wire 0.1 mm in diameter, placed at

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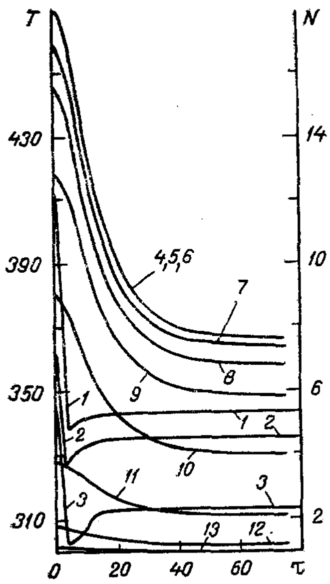

Fig. 1

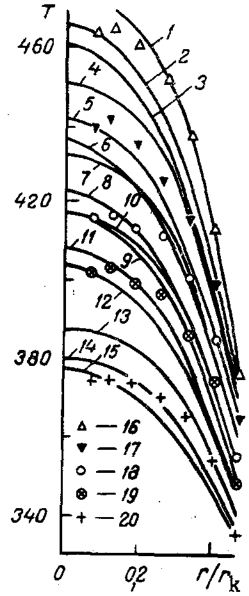

Fig. 2

Fig. 1. Thermal load and temperature of the heat-transfer agent as a function of time with a transition to a different, lower power operating regime: 1-3) change in the power for the numbers Re = $1.25\cdot10^4$; $8.9\cdot10^3$, $5.1\cdot10^3$, respectively; 4-13) change in the temperature of the heat-transfer agent for Re = $1.25\cdot10^4$ with r/r_k = 0.073; 0.128; 0.193; 0.265; 0.334; 0.408; 0.479; 0.624; 0.770; 0.916, respectively. T, K; N, kW; τ, sec.

Fig. 2. Temperature distribution in the heat-transfer agent in the output section of the bundle for Re = $1.25\cdot10^4$: 1-15) computed temperature distribution: 1, 4, 7, 10, 13) with K_n = 0.03; 2, 5, 8, 11, 14) with K_n = 0.06; 3, 6, 9, 12, 15) with K_n = 0.075; 16-20) experimental data for τ = 4, 8, 12, 16, and 32 sec, respectively.

characteristic points of the flow with the coordinates r/r_k = 0.073; 0.128; 0.193; 0.265; 0.334; 0.408; 0.479; 0.624; 0.770; 0.916. The time constant of the thermocouples equaled 0.04-0.2 sec, which is acceptable for purposes of these experiments.

The experiments were performed for Reynolds numbers in the range Re = $5.1\cdot10^3$-$1.25\cdot10^4$ and rates of cooling of the wall $(\partial N/\partial\tau)_m$ = 1.075-1.875 kW/sec. The thermal load as a function of time for the operating regimes of the bundle with the Reynolds numbers Re = $5.1\cdot10^3$; $8.9\cdot10^3$; $1.25\cdot10^4$ is shown in Fig. 1. The figure also shows the typical change in the experimentally measured temperatures of the heat-transfer agent as a function of time for Re = $1.25\cdot10^4$. One can see that as the thermal load changes over a period of 16 sec the temperature of the heat-transfer agent emerges to a new stationary level at each point of the flow practically within 60-76 sec. In Fig. 2 the experimentally measured temperature distributions in the transfer section of the bundle for different times and for Re = $1.25\cdot10^4$ are compared with the theoretically computed temperature distributions for different values of the coefficient K_n.

The temperature distributions were calculated with the help of the model of the flow of a homogenized medium [2, 3, 6, 7] and a system of differential equations including the equations of energy, motion, and continuity and the equation of state, as well as the heat-conduction equation, describing the temperature distribution in the "solid phase" — in the twisted tubes. For the problem at hand, this system of equations has the form:

$$\rho_{\text{т}} c_{\text{т}} \frac{\partial T_{\text{т}}}{\partial \tau} = q_v - \frac{4\alpha m}{(1-m)\, d_e}(T_{\text{т}} - T) + \frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda_{\text{т}r}\frac{\partial T_{\text{т}}}{\partial r}\right) + \frac{\partial}{\partial x}\left(\lambda_{\text{т}x}\frac{\partial T_{\text{т}}}{\partial x}\right), \tag{3}$$

$$\rho c_p \frac{\partial T}{\partial \tau} + \rho u c_p \frac{\partial T}{\partial x} = \frac{\partial P}{\partial \tau} + \frac{4\alpha}{d_e}(T_{\text{т}} - T) + \frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda_{\text{э}}\frac{\partial T}{\partial r}\right) + \frac{\partial}{\partial x}\left(\lambda_{\text{э}}\frac{\partial T}{\partial x}\right), \tag{4}$$

$$\rho u\frac{\partial u}{\partial x} = -\frac{\partial P}{\partial x} - \xi\frac{\rho u^2}{2d_e} + \frac{1}{r}\frac{\partial}{\partial r}\left(r\rho\nu_e\frac{\partial u}{\partial r}\right), \tag{5}$$

$$G = 2\pi m\int_0^{r_k}\rho u r dr, \tag{6}$$

$$P = \rho RT. \tag{7}$$

Thus, in the case of a stationary process a one-temperature model of the flow [2], when only the temperature distribution in the heat-transfer agent is determined, was employed, whereas for the nonstationary case a two-temperature mode, in which the change in the temperature of the tubes as a function of time is also taken into account, is employed in order to take into account the thermal inertia of the twisted tubes. The system of equations (3)-(7) is supplemented by the following boundary conditions: at the input into the bundle (x = 0):

$$T_T(r, 0, \tau) = T_{T\,in}(r, \tau), \quad T(r, 0, \tau) = T_{in}(r, \tau),$$
$$u(r, 0, \tau) = u_{in}(r, \tau), \quad P(r, 0, \tau) = P_{in}(\tau), \tag{8}$$

at the output from the bundle (condition of no heat transfer):

$$\left.\frac{\partial T_T(r, x, \tau)}{\partial x}\right|_{x=l} = 0, \quad \left.\frac{\partial T(r, x, \tau)}{\partial x}\right|_{x=l} = 0, \tag{9}$$

on the axis of the bundle (condition of axial symmetry):

$$\left.\frac{\partial T_T(r, x, \tau)}{\partial r}\right|_{r=0} = 0, \quad \left.\frac{\partial T(r, x, \tau)}{\partial r}\right|_{r=0} = 0, \quad \left.\frac{\partial u}{\partial r}\right|_{r=0} = 0, \tag{10}$$

on the outer boundary of the bundle:

$$-\lambda_{Tr}\left.\frac{\partial T_T(r, x, \tau)}{\partial r}\right|_{r=r_k} = 0, \quad -\lambda_e\left.\frac{\partial T(r, x, \tau)}{\partial r}\right|_{r=r_k} = 0, \quad \left.\frac{\partial u}{\partial r}\right|_{r=r_k} = 0. \tag{11}$$

The starting conditions are found from the solution of the stationary problem at the time $\tau = 0$. In solving the system (3)-(7) the quantities in front of the derivatives were first averaged over the differentiation variables and were removed from the differentiation operator; then they were refined in iteration cycles. The equations of gas dynamics (5) and (6) are written in the quasistationary approximation, since in the experiments the conditions under which the perturbations of the parameters determining the flow process are small and the duration of the perturbations is much longer than the propagation time of a sound wave along the length of the bundle were realized. In the heat-conduction equation (3) the coefficient of thermal conductivity of the "solid phase" is written down taking into account the directional anisotropy of the properties of this phase. In so doing, the coefficient of thermal conductivity in the radial direction was determined by the dependence

$$\lambda_{Tr} = \left[\frac{1-\varepsilon}{\lambda_T} + \frac{\varepsilon}{\lambda_h}\right]^{-1}, \tag{12}$$

and in the longitudinal direction by the formula

$$\lambda_{Tx} = \lambda_T(1-\varepsilon) + \lambda_h\varepsilon, \tag{13}$$

where the concept of an equivalent coefficient of thermal conductivity was employed [4]. Taking into account the volume occupied by the heat-transfer agent in the tubes, the density of the "solid phase" was assumed to equal

$$\rho_r = \rho_m(1-\varepsilon) + \rho_h\varepsilon, \tag{14}$$

while the heat capacity of the "solid phase" was determined by the relation

$$c_r = c_m(1-\varepsilon) + c_h\varepsilon. \tag{15}$$

In the expressions (12)-(15) ε is the ratio of the area of the flow-through section of the tube to the total cross-sectional area of the tube, the indices t and m refer to the tube material, and the index g refers to the heat-transfer agent filling a tube. The energy

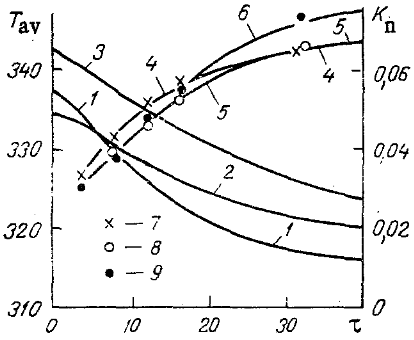

Fig. 3

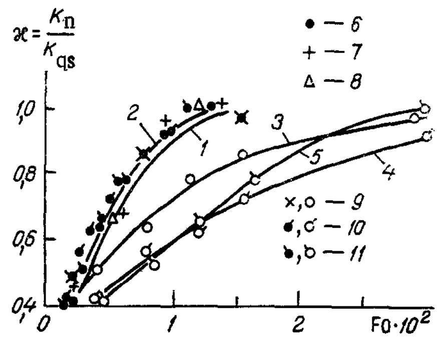

Fig. 4

Fig. 3. Mean mass temperature and the coefficient K_n as a function of time with a reduction of the heat load for $Fr_m = 57$: 1-3) change in the mean mass temperature for Re = $1.25 \cdot 10^4$; $8.9 \cdot 10^3$; $5.1 \cdot 10^3$; 4-6) change in the coefficient K_n for Re = $1.25 \cdot 10^4$; $8.9 \cdot 10^3$; $5.1 \cdot 10^3$; 7-9) experimental data for the same Re numbers.

Fig. 4. Coefficient κ as a function of the Fourier number with a reduction of the heat load: 1) the dependence (1); 2) the dependence (20); 3, 4, 5) the dependence $\kappa = f(Fo_b)$ for $(\partial N/\partial\tau) = 1.875$, 1.75, and 1.075 kW/sec; 6, 7, 8) experimental data for a bundle with $Fr_m = 220$ with Re = $8.9 \cdot 10^3$ and $(\partial N/\partial\tau)_m$ = 7.5 kW/sec; Re = $1.36 \cdot 10^4$ and $(\partial N/\partial\tau)_m$ = 1.075 kW/sec. Re = $1.75 \cdot 10^4$ and $(\partial N/\partial\tau)_m$ = 10 kW/sec, respectively; 9) experimental data for $Fr_m = 57$, Re = $1.25 \cdot 10^4$ and $(\partial N/\partial\tau)_m$ = 1.875 kW/sec with analysis using the Fo_m (black dots) and Fo_b (white dots) numbers; 10) the same for Re = $8.9 \cdot 10^3$ and $(\partial N/\partial\tau)_m$ = 1.175 kW/sec; 11) the same for Re = $5.1 \cdot 10^3$ and $(\partial N/\partial\tau)_m$ = 1.075 kW/sec.

equation (4) and the heat-conduction equation (3) were solved by the method of variable directions. The numerical analogs of the equations were constructed according to an implicit scheme and solved by the method of alternate intervals. The equations of gas dynamics (5) and (6) were solved by the method of alternate intervals with the help of Sumini's substitution. These solutions were then linked through the equation of state and the iteration cycles [3-6]. The coefficients λ_e and ν_e in (4) and (5) can be uniquely related with the experimentally determined coefficient K_n ($K_n = \nu_e/ud_e$; $K_n = \lambda_e/\rho c_p u d_e$), setting as an approximation the effective turbulent Lewis and Prandtl numbers equal to unity [3].

To determine the effective coefficient of diffusion K_n the experimentally measured temperature distribution in the heat-transfer agent (Fig. 2) are compared with the temperature distributions computed theoretically by the modified method of least squares [3]. Comparison of the experimental temperature distributions with the theoretical distributions $T = T(r/r_k, \tau, K)$ indicates that the coefficient K_n decreases during the first few moments compared with the quasistationary value. The coefficient K_n as a function of time is shown in Fig. 3. One can see that the coefficient K_n becomes practically stationary within a time τ = 30-40 sec for $(\partial N/\partial\tau)_m$ = 1.075-1.875 kW/sec. At the same time, for $(\partial N/\partial\tau)_m$ = 7.5-10 kW/sec K_n for a bundle with $Fr_m = 220$ assumed a quasistationary value within 10-13 sec [3, 6]. For this reason when the experimental data for $(\partial N/\partial\tau)_m$ = 1.075-1.875 are analyzed in the form

$$\varkappa = K_n/K_{qs} = f(Fo_b) \tag{16}$$

a deviation from the dependence (1) is observed (Fig. 4). This is attributable to the fact that the criterion Fo_b does not take into account the change in the derivative of the wall temperature $\partial T_w/\partial\tau$, with which the time derivative of the thermal load $\partial N/\partial\tau$ is related. To take into account the change in the turbulence structure of the flow in the layer at the wall accompanying a change in the temperature of the wall, in [4] an effective time, defined by the dependence

$$\tau_e = (\tau - \tau_0)\left[a + b\left(\frac{\partial N}{\partial\tau}\right)_m\right], \tag{17}$$

was introduced into the Fourier criterion instead of the real time; here, τ is the real time, measured from the moment at which the thermal operating regime of the apparatus starts to change; $a = 0.043$; $b = 0.263$ sec/kW; and, τ_0 is the period of time preceding the moment at which the thermal load is increased sharply. In addition, the modified Fourier criterion, calculated from the formula

$$Fo_m = \frac{a\tau_e}{d_k^2} = \frac{\lambda_b(\tau - \tau_0)}{c_p\rho_b d_k^2}\left[a + b\left(\frac{\partial N}{\partial \tau}\right)_m\right]. \quad (18)$$

is employed as the determining criterion.

For the case of a sharp reduction in the thermal load the time τ_0 in the expressions (17) and (18) equals zero. If it is assumed that for the type of nonstationarity under study the effect of the parameter $(\partial N/\partial\tau)_m$ on the coefficient K_n is analogous to the effect of this parameter on K_n with an increase in the thermal load, i.e.,

$$Fo_m = \frac{\lambda_b\tau}{c_p\rho_b d_k^2}\left[0.043 + 0.263\left(\frac{\partial N}{\partial \tau}\right)_m\right], \quad (19)$$

then the experimental data for bundles with $(\partial N/\partial\tau)_m$ = 1.075-1.875 kW/sec and $Fr_m = 57$ agree well with the experimental data for $(\partial N/\partial\tau)_m$ = 7.5-10 kW/sec and $Fr_m = 220$ (Fig. 4). Here the experimental data on K_n for bundles with $Fr_m = 57$ and 220 are referred to the quasistationary values of the coefficient K_{qs}, obtained experimentally for each Reynolds number Re studied, the quantity $(\partial N/\partial\tau)_m$, and the number Fr_m. Then the experimental data on the relative coefficient $\kappa = K_n/K_{qs}$ for beams with the numbers $Fr_m = 57$ and 220 in a range of values of Re and $(\partial N/\partial\tau)_m$, encompassed by experiments [$Re = 5.1\cdot10^3$-$1.75\cdot10^4$, $(\partial N/\partial\tau)_m$ = 1.075-10 kW/sec], can be generalized by a single dependence

$$\varkappa = 0.454\cdot10^{-5}Fo_m^{-2} - 3.86\cdot10^{-3}Fo_m^{-1} + 1.28, \quad (20)$$

valid for the numbers $Fo_m \leq 1.4\cdot10^{-2}$. This dependence generalizes the experimental data for both the case of a reduction of the thermal load to zero (stopping of the heat-exchange apparatus) and the case of thermal-load reduction with a transition from one operating regime to another. An analogous result was obtained also for the type of nonstationarity associated with an increase in the thermal load [8].

The criterion Fr_m, characterizing the characteristics of the flow in the bundle of twisted tubes, affects differently the coefficient K_n for different types of nonstationarity. When the number Fr_m is reduced under conditions of nonstationary heating of the bundle of twisted tubes the process of equalization of temperature nonuniformities proceeds much more rapidly (the coefficient K_n assumes a quasistationary value more rapidly [8]), whereas for a reduction of the thermal load the number Fr_m is not observed to affect the value of κ.

The foregoing generalization of the experimental data suggests a dependence for calculating the nonstationary effective coefficient of diffusion for the operating regimes of heat-exchange apparatus and systems, associated with a reduction of the thermal load to zero as well as with a transition from one operating regime to another with a lower thermal power. This dependence can be employed to close the system of differential equations describing nonstationary heat- and mass-transfer in bundles of twisted tubes for the type of nonstationarity studied above.

The good agreement between the experimentally measured and theoretically computed temperature distribution for regimes with a reduction of the thermal load serves as an experimental justification for the flow model adopted, its mathematical description, and the methods for calculating nonstationary flow in bundles of twisted tubes for this case.

NOTATION

N, thermal load; τ, time; K_n, a dimensionless effective coefficient of diffusion; u, velocity; d_e, equivalent diameter; Fr_m, a criterion characterizing the characteristics of the flow in a bundle of twisted tubes; S, pitch of the twisting of a tube; d, maximum size of the profile of the tube; Fo_b, Fourier' criterion; a, coefficient of thermal diffusivity; d_c, diameter of the jacket of the heat exchanger (bundle of tubes); λ_b, coefficient of thermal conductivity; c_p, specific heat capacity; ρ_b, density; x and r, longitudinal and radial coordinates; r_k, radius of the bundle; Re, Reynolds number; α, heat-transfer coefficient;

q_V, volume density of energy release; P, pressure; κ, relative coefficient of diffusion; Fo_m, modified Fourier criterion; ξ, coefficient of hydraulic resistance; and m, porosity of bundle of tubes with respect to the heat-transfer agent.

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FLOW IN A ROTATING SLOT CHANNEL WITH INJECTION THROUGH ONE WALL AND COMPENSATING SUCTION THROUGH THE OTHER

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The results of calculating the velocity field and integral characteristics for laminar conditions are presented and analyzed.

The use of porous materials is regarded as a promising trend in the improvement of convective cooling systems, including rotary components of power machinery. As a consequence, it is necessary to investigate the flow of coolant media in rotating channels with walls through which there is injection or suction. In a general formulation, this problem is extremely complex, since the field of hydrodynamic characteristics is significantly three-dimensional. However, as in the case of motionless channels with permeable walls [1, 2], a series of simple model problems may be formulated and solved, thereby elucidating the basic specific effects and obtaining the corresponding quantitative estimates. One such problem forms the subject of the present work.

Suppose that a prismatic slot channel of constant height 2h, in which a viscous liquid moves isothermally, is uniformly rotated at angular velocity ω relative to an axis perpendicular to the wide wall forming the slot. A Cartesian coordinate system 0, x, y, z is introduced; this system is rigidly connected to the channel, and is oriented so that the y axis is directed along the axis of rotation, the z axis is parallel to the lateral boundary wall of the channel in the direction of flow, and the coordinate origin is in the median plane of the channel.

Assume that the wide walls are of uniform porosity over the surface. At the wall $y = h_0$, $v_0 = \text{const} < 0$, which corresponds to injection. At the wall $y = -h$, the normal velocity is also v_0, i.e., there is suction equal in intensity to the injection. The side walls are impermeable. In these conditions, the liquid flow rate along the axis 0z is constant and the problem may be formulated for calculating developed flow, in which the relative-velocity field does not depend on the coordinate z. Limiting consideration to flow in the region far from the side walls, the equations of relative motion may be written in the form

$$\nu \frac{d^2u}{dy^2} = \frac{\partial p^*}{\partial x} + 2\omega w + v_0 \frac{du}{dy}, \qquad (1)$$

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