STIRRING THE HEAT CARRIER IN A HEAT EXCHANGER BY TWISTING THE STREAM

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A theoretical–experimental study was made of the effective diffusion coefficient in longitudinally streamlined helically twisted tubes with an oval profile, using as the basis a homogenized model of flow and applying the method of diffusion from linear heat sources of finite dimensions.

Heat exchangers with stream twisting were considered in several studies concerning longitudinal and transverse streamlining of helically twisted tubes with an oval profile in the intertube space [1, 2]. In such heat exchangers, with the Reynolds number $N_{Re} < 5 \cdot 10^4$, there occurs an appreciable intensification of heat transfer due to flow between tubes as well as through them. One should also expect a much better stirring of the heat carrier within the intertube space, where the flow is characterized by large transverse components of velocity. It is worthwile, therefore, to study the transfer characteristics of a stream flowing along a bundle of twisted tubes. In one study [3] the transfer characteristics of a stream in bundles of twisted tubes were analyzed by the method of diffusion from a point heat source and a relation was obtained for calculating the asymptotic value of the dimensionless effective diffusion coefficient

$$k_{ac} = D_t/ud_e = 0.0356\,(1 + 8.1\mathrm{Fr}_M^{-0.278}). \tag{1}$$

According to relation (1), k_{ac} depends almost solely on the dimensionless number $N_{Fr,M}$

$$\mathrm{Fr}_M = S^2/d_e d, \tag{2}$$

characterizing the peculiarities of flow through a bundle of coiled tubes. In practice, however, the temperature field in the heat carrier within the intertube space of a heat exchanger becomes nonuniform not so much due to point heat sources but due to linear heat sources of finite dimensions such as, e.g., in the case of a nonuniform admission of heat through the tube walls. It is worthwile, therefore, to study the effective diffusion coefficient in a bundle of coiled tubes by the method of heat diffusion from an array of linear sources of finite dimensions, this method utilizing the homogenized flow model [4] more successfully than does the identical to it method of heating the center tube [5]. In this case one can single out a group of tubes in the center and heat it by passing an electric current through them, after having electrically insulated them from the remaining tubes. During their heating the temperature field in the bundle will become nonuniform, but this nonuniformity will then be partly smoothed along the bundle as a result of transverse stirring. The effective diffusion coefficient, which characterizes the heat transfer process in a bundle of twisted tubes, is determined on the basis of a comparison of the temperature fields measured experimentally at the exit section of the bundle with the temperature fields calculated theoretically by methods of mathematical statistics.

Transverse stirring of the heat carrier in a bundle of helically twisted tubes was studied in an experimental test stand shown schematically and described in an earlier report [1]. The bundle of 37 coiled tubes with an oval profile was 750 mm long. The tubes were made of grade Kh18N10T steel, with the largest dimension of the profile d = 12.31 mm and the wall thickness 0.2 mm, electrically insulated from one another with a heat-resistant varnish coating. Electric power from the model OSU-100 welding transformer was pumped into the center group of seven coiled tubes. Air purified of moisture, oil, and solid impurities served as the heat carrier. The air left the tubes into the open axisymmetrically so that the temperature fields could be measured with a thermocouple mounted on a coordinate plotter. On the same coordinate

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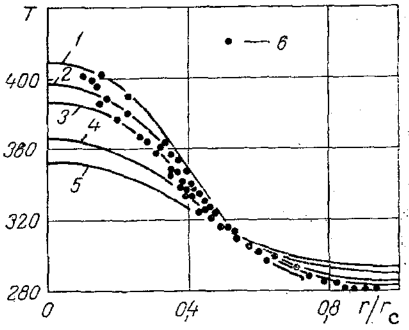

Fig. 1

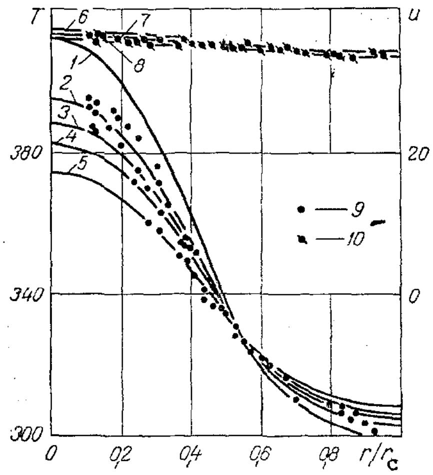

Fig. 2

Fig. 1. Comparison of the measured temperature fields T(K) and the calculated temperature fields at various values of the coefficient k, for $N_{Fr,M}$ = 1050: 1-5) results of numerical solution of the system of Eqs. (3)-(6) with k = 0.020, 0.025, 0.030, 0.04, 0.06, respectively; 6) experimental data.

Fig. 2. Comparison of the measured temperature and velocity fields and the calculated ones at various values of the coefficient k, for $N_{Fr,M}$ = 232: 1-5) results of numerical solution of the system of Eqs. (2)-(6) with k = 0.030, 0.040, 0.045, 0.050, 0.060, respectively for the temperature T(K); 6-8) results of numerical solution of the system of equations (3)-(6) with k = 0.030, 0.045, 0.060, respectively, for the velocity u (m/sec); 9) experimental data for the temperature; 10) experimental data for the velocity.

plotter was mounted a total-pressure Pitot tube for measuring the velocity head. For easier measurement of temperature and velocity fields, the tubes in the bundle had been arranged as as to leave a free space between the rows in the form of slits in the plane in which the probes were moved. The measurement procedure and the method of mathematical data processing that the maximum error of a coefficient k determination did not exceed ±50%.

Experiments were performed with $N_{Fr,M}$ = 64-1050, $N_{Re,b} = (1.2\text{-}1.7)\cdot 10^4$, and $q = (1.2\text{-}1.9)\cdot 10^4$ W/m^2.

The graphs in Figs. 1-2 depict the experimentally determined temperature fields in the heat carrier and the velocity fields in the exit section of the bundle of coiled tubes, with $N_{Fr,M}$ = 1050, 232, and 64, respectively, as well as the theoretically determined temperature and velocity fields, based on the numerical solution of the system of equations

$$\rho u \frac{\partial u}{\partial x} = -\frac{dP}{dx} + \frac{1}{r}\frac{\partial}{\partial r}\left(\rho r D_t \mathrm{Pr}_t \frac{\partial u}{\partial r}\right) - \xi \frac{\rho u^2}{2d_e}, \tag{3}$$

$$\rho u c_p \frac{\partial T}{\partial x} = \frac{1}{r}\frac{\partial}{\partial r}\left(\rho r c_p D_t \frac{\partial T}{\partial r}\right) + q_v \frac{1-m}{m}, \tag{4}$$

$$G = 2\pi m \int_0^{r_c} \rho u r dr, \tag{5}$$

$$P = \rho R T, \tag{6}$$

describing the flow in the homogenized model [4] where

$$q_v = \frac{Q}{\pi h^2 L m}$$

and the boundary conditions for the problem are

$$T(0, r) = T_{in},\ u(0, r) = u_{in},\ P(0, r) = P_{in}, \tag{7}$$

$$\left.\frac{\partial T(x, r)}{\partial r}\right|_{r=0} = 0,\quad \left.\frac{\partial u(x, r)}{\partial r}\right|_{r=0} = 0, \tag{8}$$

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

This nonlinear system of equations of the parabolic kind was solved by the method of grids according to the explicit scheme [6]. Upon introduction of the dimensioness variables

$$\hat{T} = \frac{T}{T_{in}},\quad \hat{u} = \frac{u}{u_{in}},\quad \hat{P} = \frac{P}{P_{in}},\quad \hat{r} = \frac{r}{r_c},\quad \hat{x} = \frac{x}{L}, \tag{10}$$

the system of equations (3)-(6) can be reduced to a dimensionless system in finite differences. The constants c_1, ..., c_6 in the finite-difference analogs of the original differential equations are

$$c_1 = k\frac{L}{r_c}\frac{d_e}{r_c}, \tag{11}$$

$$c_2 = \frac{Lq_v}{\rho_{in} u_{in} c_p T_{in}}\frac{1-m}{m}, \tag{12}$$

$$c_3 = c_1 \mathrm{N}_{\mathrm{Pr}_t}, \tag{13}$$

$$c_4 = \xi\frac{L}{2d_e}, \tag{14}$$

$$c_5 = \frac{P_{in}}{\rho_{in} u_{in}^2}, \tag{15}$$

$$c_6 = 2\pi m \rho_{in} u_{in} r_c^2, \tag{16}$$

while the equation of motion and the energy equation become

$$\hat{u}(\hat{x} + h_x, \hat{r}_c) = \hat{u}(\hat{x}, \hat{r}) + Z_u(\hat{x}, \hat{r})h_x - c_5\frac{\hat{T}(\hat{x}, \hat{r})}{\hat{u}(\hat{x}, \hat{r})}\frac{\hat{P}(\hat{x} + h_x) - \hat{P}(\hat{x})}{\hat{P}(\hat{x})}, \tag{17}$$

$$\hat{T}(\hat{x} + h_x, \hat{r}) = \hat{T}(\hat{x}, \hat{r}) + Z_T(\hat{x}, \hat{r})h_x, \tag{18}$$

where

$$Z_u(\hat{x}, \hat{r}) = c_3\frac{\hat{T}(\hat{x}, \hat{r})}{\hat{u}(\hat{x}, \hat{r})\hat{r}}D\left(\hat{u}\frac{\hat{r}}{\hat{T}}, \hat{u}\right) - c_4\hat{u}(\hat{x}, \hat{r}); \tag{19}$$

$$Z_T(\hat{x}, \hat{r}) = c_1\frac{\hat{T}(\hat{x}, \hat{r})}{\hat{u}(\hat{x}, \hat{r})\hat{r}}D\left(\hat{u}\frac{\hat{r}}{\hat{T}}, \hat{T}\right) + c_2\frac{\hat{T}}{\hat{P}\hat{u}}. \tag{20}$$

Functions $D\left(u\frac{\hat{r}}{\hat{T}}, \hat{u}\right)$ and $D\left(\hat{u}\frac{\hat{r}}{\hat{T}}, \hat{T}\right)$ in expressions (19) and (20) can in this case be described in the form

$$D(f, \varphi) = \frac{\partial}{\partial r}\left(f\frac{\partial\varphi}{\partial r}\right)$$

or in the form of finite differences

$$D(f, \varphi) = \frac{1}{2h_r^2}[(f_{i+1} + f_i)(\varphi_{i+1} - \varphi_i) - (f_i + f_{i-1})(\varphi_i - \varphi_{i-1})]. \tag{21}$$

The pressure at point $\hat{x} + h_x$ is found from the equation of a continuous stream, viz.,

$$G_0 = \hat{P}(\hat{x} + h_x)\int_0^1 \frac{\hat{u}(\hat{x} + h_x, \hat{r})rdr}{\hat{T}(\hat{x} + h_x, \hat{r})} = \hat{P}(\hat{x})\int_0^1 \frac{\hat{u}(\hat{x}, \hat{r})\hat{r}d\hat{r}}{\hat{T}(\hat{x}, \hat{r})}. \tag{22}$$

The system of equations (17)-(22) can be further simplified, assuming that $\hat{T}(\hat{x} + h_x, \hat{r})$ does not depend on $\hat{P}(\hat{x} + h_x)$, and that $\hat{u}(\hat{x} + h_x, \hat{r})$ depends on $\hat{P}(\hat{x} + h_x)$ linearly.

An algorithm was constructed according to this method and programmed on a model BÉSM-4M high-speed computer. For the computations pertaining to the given bundle of tubes were stipulated their geometrical characteristics (F, m, d, r_h, r_c, L) as well as the pressure, the temperature, and the velocity of the heat carrying stream at the entrance to the bundle, also q_v and the physical properties of the heat carrier. On the r axis N nodes and the step h_r were selected, then the step $h_x = 1/5h_r^2$ was selected. The numbers of steps along the radius and along the height were selected on the basis of accuracy and minimum machine time considerations. Both requirements for the given problem were met by 82 steps along the radius and 1355 steps along the height. The accuracy of calculations was checked by comparing the mean-mass temperature of the heat carrier at the exit from the bundle with the mean temperature of the calculated temperature field at that exit section. These temperatures agreed within 1%.

For determining the effective diffusion coefficient k, the theoretically calculated temperature fields and the experimentally measured ones (Figs. 1-3) were compared by two methods. In the first method for each calculated $T = T(\bar{r})$ curve with a given value of coefficient k was found the square root of the sum of squares of deviations of all n experimental points from that curve and then the graph of the relation

$$\sqrt{\sum_{i=1}^{n} (\delta T_i)^2} = f(k), \tag{23}$$

was plotted.

The minimum of function (23) corresponds to the maximum reliable value of coefficient k with which the closest agreement between experimental and theoretical data is obtained.

The second method involved determining from the experimental sample the most likely confidence intervals of k variation. In the implementation of this data grouping method each experimental point on the $T = T(\bar{r})$ curve was assumed to correspond to a definite value of coefficient k. The grid of $T = T(\bar{r})$ curves plotted on the graph together with experimental data (Figs. 1-3) had m intervals of length h with respect to coefficient k. Calculations were performed in the following sequence. First the number of points falling within the interval (k_i, k_i + h) was determined, they were assigned to the center of this interval, the mean value of k in this sample calculated as

$$\bar{k} = \frac{1}{n} \sum_{i=1}^{m} n_i \bar{k}_i, \tag{24}$$

and the adjusted dispersion of k was calculated as

$$\sigma^2 = \frac{1}{n-1} \sum_{i=1}^{m} n_i (\bar{k}_i - \bar{k})^2 - \frac{h^2}{12}. \tag{25}$$

Then, based on the hypothesis of k having a normal distribution, the confidence interval for k was found to be

$$k = \bar{k} \pm 1.96\sigma, \tag{26}$$

with $\sigma = \sqrt{\sigma^2}$ denoting the standard deviation of k.

The mean values of coefficient k thus found by the two methods and referred to k_{ac} (1) are in Fig. 4 compared with the relation

$$\frac{\bar{k}}{k_{ac}} = 1 - \exp\left[-0.0504 \frac{2ax}{d}\right], \tag{27}$$

proposed in an earlier study [4] for bundles of rods with helical finning. On the same diagram are also plotted the confidence intervals for k according to relation (26). The structural factor a in expression (27), characterizing the stream in bundles of coiled tubes, was calculated according to the relation [4]

$$a = 0.0745 + 11.37 N_{Fr,M}^{-1} + 246\ N_{Fr,M}^{-2}. \tag{28}$$

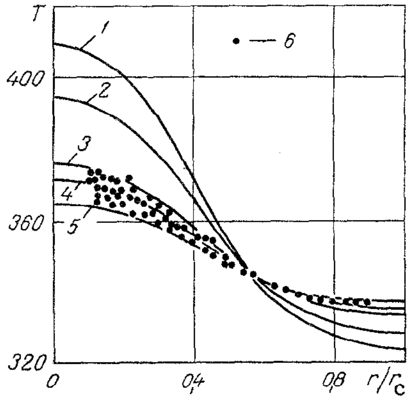

Fig. 3

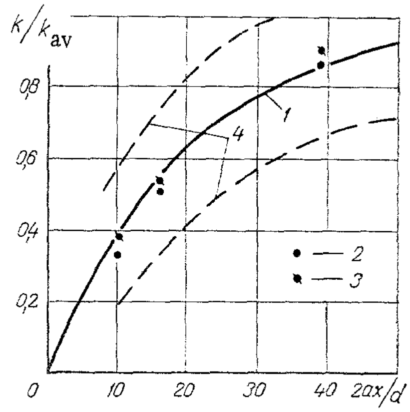

Fig. 4

Fig. 3. Comparison of the measured temperature fields and the calculated temperature fields at various values of the coefficient k, for $N_{Fr,M}$ = 64: 1-5) results of numerical solution of the system of Eqs. (3)-(6) with k = 0.045, 0.060, 0.095, 0.110, respectively; 6) experimental data.

Fig. 4. Relative effective diffusion coefficient as a function of the referred longitudinal coordinate: 1) according to relation (27); 2) experimental data based on relation (24); 3) experimental data based on relation (23); 4) confidence intervals based on relation (26).

According to the graph in Fig. 4, the experimental data on the relative diffusion coefficient k/k_{ac} for a bundle of coiled tubes agree closely with relation (27) for bundles of finned rods. This indicates that the mechanisms of transfer in bundles of coiled tubes and in bundles of finned rods are practically the same, also that the mean value of the effective diffusion coefficient depends equally in both cases on the referred length $2ax/d$ of the bundle. Therefore, taking into account relations (1) and (27), one can propose for the mean effective diffusion coefficient for a bundle of coiled tubes the expression

$$\bar{k} = 0.0356\,(1 + 8.1\,N_{Fr,M}^{-0.278})\left\{1 - \exp\left[-0.1008\,(0.0745 + 11.37\,N_{Fr,M}^{-1} + 246\,N_{Fr,M}^{-2}\,\frac{x}{d}\right]\right\}, \quad (29)$$

which will yield the values $\bar{k}$ for heat exchangers with helically twisted tubes of any length and with the given kind of nonuniformity of the temperature fields in the heat carrier.

A comparison of the earlier results [3] with the results of this study reveals that for a determination of the coefficient k_{ac} the method of diffusion from a point heat source requires shorter bundles of coiled tubes than does the method of diffusion from a group of linear heat sources. No dependence of the coefficient $\bar{k}$ on the Reynolds number over the given range of the latter was discovered in this study.

The theoretical—experimental studies of the effective diffusion coefficient indicate the possibility of using this homogenized flow model for the design of intertube stirring of the heat carrier in longitudinally streamlined bundles of helically coiled tubes.

NOTATION

D_t, effective diffusion coefficient; u, stream velocity; d_e, equivalent diameter; S, pitch of the tube profile; d, largest profile dimension; $N_{Re,b}$, Reynolds number; q_v, volume heat emission; ρ, density; x and r, coordinates; P, pressure; $N_{Pr,t}$, Prandtl number for turbulent flow; ξ, hydraulic drag coefficient; c_p, specific heat; T, temperature; G, mass flow rate of heat carrier; m, porosity of the bundle of tubes relative to the heat carrier; L, length of the bundle of tubes; r_h, radius of heated tubes; r_c, radius of the bundle of tubes; and k, dimensionless effective diffusion

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HEAT EXCHANGE IN TWO-PHASE FLOW ABOUT A SURFACE LOCATED IN A RECTANGULAR CHANNEL

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Similitude equations are obtained on the basis of the principle of superposition of separate effects to calculate heat exchange between surfaces with complex-shaped cross sections located in a rectangular channel during their cooling by a two-phase flow.

The design of equipment characterized by high levels of heat release and which satisfies the requirement of maintaining the thermal state of the heat-exchanging surfaces within a specified temperature range involves the use of efficient methods of cooling. Mass-exchange cooling has come into wide use in this regard. One form of mass-exchange cooling is cooling with a two-phase flow of a gas and a finely dispersed liquid. The main advantages of this method compared to convective cooling are: a reduction in the temperature of the surface, with the direct contact of the liquid droplets with the wall; a reduction in the temperature of the coolant as a result of evaporation of moisture in the gas flow; an increase in the specific heat of the coolant (the gas—vapor—liquid mixture); an increase in the heat-transfer coefficient as a result of additional agitation of the flow by the liquid particles. In some cases, it proves best to employ a combination method — convective cooling together with the use of a two-phase flow during the operation of units under "peak" loads. Here, it is necessary to know the limits of applicability of the convective cooling and the consumption of liquid required in relation to the service conditions of the unit (the heat flux, time of operation under "peak" load, number of startups of the water-supply system, etc.).

However, the data available in the literature pertains mainly either to the flow of two-phase mixtures in channels or to the interaction of individual droplets with a hot surface. Furthermore, the final results cannot be generalized so as to yield a suitable design of cooling system for hot surfaces of complex shape located in a channel admitting a two-phase flow.

The present work presents results of a study of heat exchange and the thermal state of electrically heated metal surfaces (resistors) of length L, width H, and thickness 2δ when the inequalities $L \gg 2\delta$, $H \gg 2\delta$, $L \gg H$ are observed and the resistor is cooled on both sides. Here the temperature field of the resistor depends only on the coordinate y and, assuming that $\lambda = \text{const}$, the mathematical description of the heat-conduction process for steady-state conditions has the form

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