LOCAL HEAT TRANSFER IN THE INTERTUBE SPACE OF A HEAT EXCHANGER WITH SPIRAL TUBES

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The article explains the methods and presents the results of the experimental investigation of local heat transfer in bundles of spiral tubes with heat supply to the heat carrier that is nonuniform over the cross section.

Heat exchangers with spiral tubes of oval profile, assembled into a densely packed bundle, may be very compact on account of the intensified heat exchange in them. This is why heat exchangers with such tubes [1-5] arouse considerable interest. Heat transfer in longitudinal flow around spiral tubes was investigated in [3-5]. Dzyubenko and Dreitser [3] detected a marked intensification of heat transfer upon flow of the heat carrier, both inside the tubes with a relative pitch of the spirals $S/d = 6.5$ and in the space between the heat-exchanger tubes, and they obtained criterial dependences for calculating the heat-transfer coefficient. Dzyubenko and Ievlev [4] and Ievlev et al. [5] presented different forms of processing the data on heat transfer, and they established new generalizing criterial dependences for calculating heat transfer in a wide range of changes of the similarity criterion Re and $Fr_M = S^2/dd_e$ characterizing flow in bundles of spiral tubes. The most expendient forms of processing the experimental data, found in these works, broadened the possibilities of modeling heat exchange in such bundles. The investigation of heat transfer in [3-5] was carried out by generally accepted methods with uniform heat supply to all the tubes of the bundle over its radius on the section of stabilized flow of the heat carrier.

Since it is not obvious that the experimental regularities of heat transfer obtained in uniform heating of a bundle of spiral tubes can be extended to the case of nonuniform heat supply to the heat carrier over the cross section, a special investigation was carried out involving the modeling of axisymmetric nonuniform heat supply over the radius of the bundle. The simplest case of axisymmetric nonuniformity was examined: heat supply was effected stepwise to a group of central spiral tubes of the bundle by passing an electric current through them, whereas the peripheral group of the spiral tubes was not heated. The nonuniform field of heat supply formed a nonuniform temperature field of the heat carrier over the radius of the spiral tubes in the bundle, and this was partially compensated by the lateral mixing of the flow. Thereby differing conditions of heat removal from the heated spiral tubes radially and along the bundle were induced.

Thus, to determine the local heat-transfer coefficients in the examined case of flow, it was indispensable to measure the distributions of the wall temperature of spiral tubes,

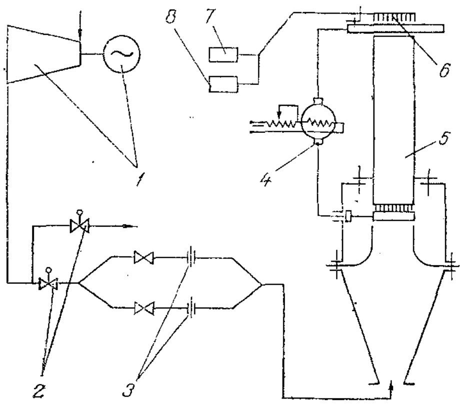

Fig. 1. Basic diagram of the experimental rig: 1) turbocompressor with motor; 2) control shutters; 3) throttle flow meters; 4) dc generator; 5) experimental section; 6) comb of sensors with coordinate mechanism; 7) gathering data on temperature measurement; 8) gathering data on measurement of dynamic head.

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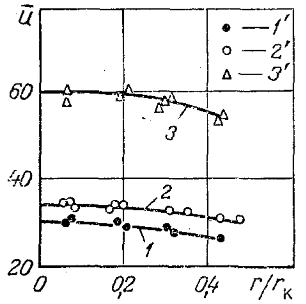

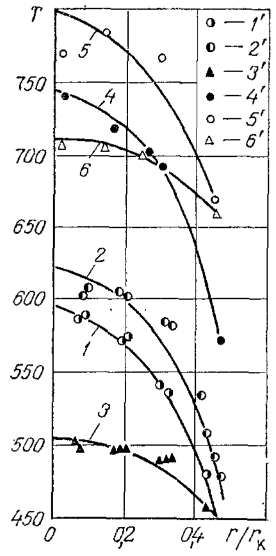

Fig. 2 Fig. 3

Fig. 2. Velocity distribution in the flow core over the radius of a bundle of spiral tubes: 1) $Fr_M = 57$; 2) 236; 3) 1082 (the lines are drawn through the experimental points 1', 2', 3' for the velocity).

Fig. 3. Temperature distribution of the heat carrier and of the tube wall over the radius of the bundle: 1) $Fr_M = 57$; 2) 236; 3) 1082 (the lines are drawn through the experimental points 1', 2', 3' for the temperature of the heat carrier); 4) $Fr_M = 57$; 5) 236; 6) 1082 (the lines are drawn through the experimental points 4', 5', 6' for the wall temperature).

of the heat flow density, of the temperature of the heat carrier, and of the speed of flow outside the near-wall layer over the radius of the bundle in the given cross section. The experimental temperature fields of the heat carrier and the flow speeds in the cross section of the bundle are in good agreement with the calculated temperature and velocity fields determined for a homogenized model of the flow described in [5, 6]. In accordance with this model, a layer of material with thickness δ^* is built up on the walls of the tubes and of the jacket, and for the new boundaries of the bundle of tubes we examine the free flow with slip of a homogenized medium with distributed sources of energy release and hydraulic resistance. Since the given investigation is based on the use of such a flow model, it is an advantage to determine the heat-transfer coefficient by the parameters on the outer boundary of the near-wall layer $\bar{T}$ and $\bar{u}$, using the expression [8]:

$$\alpha_m = \frac{q_0}{\rho_m \bar{u} c_{pm} (T_c - \bar{T})}, \tag{1}$$

which is correlated with the determining similarity criterion in the following way [4]:

$$\alpha_m = \alpha_m (Z,\ Z_m,\ \mathrm{Pr}_m). \tag{2}$$

In the dependence (2)

$$Z = \frac{\mathrm{Re}_\theta}{\alpha} = \frac{\bar{u}}{\alpha\mu} \int_0^\delta \rho \frac{u}{\bar{u}} \left(1 - \frac{u}{\bar{u}}\right) dy, \tag{3}$$

$$Z_m = \frac{\mathrm{Re}_\theta}{\alpha_m} = \frac{\bar{u}}{\alpha_m \mu} \int_0^\delta \rho \frac{u}{\bar{u}} \frac{T - \bar{T}}{T_c - \bar{T}} dy \tag{4}$$

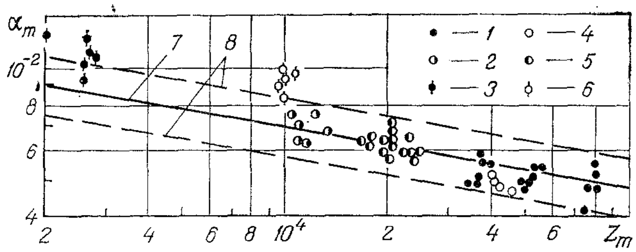

Fig. 4. Dependence of the dimensionless heat-transfer coefficient in a bundle of spiral tubes on the Reynolds number constructed in a special manner: 1) $Fr_M = 1082$, $L = 500$ mm; 2) 236 and 500; 3) 57 and 500; 4) 1050 and 750; 5) 232 and 750; 6) 57 and 750 mm, respectively; 7) dependence (15); 8) lines characterizing the deviations of the heat-transfer coefficient.

are Reynolds numbers constructed in a special way [8]. The number Re_ϑ in expression (3) is determined by the pulse thickness ϑ in the near-wall layer, and Re_θ in (4) is determined by the energy thickness θ.

Since the velocity and temperature fields in the near-wall layer of the bundle of spiral tubes are described by the same regularities, and the thicknesses of the thermal and dynamic near-wall layers are practically equal [4, 7], we can show that $Z \approx Z_m$. We will therefore seek the dependence (2) in the form

$$\alpha_m = \alpha_m (Z_m, \ \mathrm{Pr}_m).$$

The object of the present investigation is the experimental determination of dependence (2) with nonuniform heat supply to the heat carrier in the cross section of a bundle of spiral tubes.

The investigation of heat transfer was carried out on experimental rigs with bundles of 37 and 127 tubes covered with electrically insulating varnish, the heat carrier used being air. To create on these rigs a stepwise axisymmetric distribution of heat release over the radius of the bundle, the electric energy was supplied to 7 and 37 central spiral tubes, respectively. The rig with 37 spiral tubes was described in detail in [3]. We therefore confine ourselves to the basic features of the experimental rig with 127 tubes whose basic diagram is shown in Fig. 1. This rig was an open-type circuit in which air was supplied with the aid of a turbocompressor. The throughput of air was measured by double diaphragms, and it was remotely controlled by shutters 2 mounted in the main and bypass lines (Fig. 1).

The air to the bundle was supplied from below upward through a diffuser, a chamber with flow-stabilizing screens, and the inlet nozzle patterned on a Vitoshinskii profile. After having passed through the bundle, the air was released into the atmosphere.

The group of 37 spiral tubes was heated by a dc generator whose power was controlled by changing the current intensity in the excitation circuit of the generator. For voltage stabilization a special electronic device with feedback was used. The current intensity was measured by the voltage drop on the shunt.

The temperature of the tube walls and of the air was measured by Chromel—Alumel thermocouples which were connected to the automatic system for data gathering and recording. The thermocouples indicating the wall temperature were welded to the inner surface of the spiral tubes, and the thermocouple leads were led out through the lower part of the bundle and through the wall of the inlet chamber to the measuring instrument. Pitot tubes were used for measuring the flow velocities in the outlet section of the bundle. Temperature and speed were measured in the section of the tube that was 25 mm upstream from the outlet. The temperature and flow-velocity sensors were mounted on a special coordinate mechanism. The above-described devices made it possible to investigate heat transfer with nonuniform heat supply over the radius of a bundle of spiral tubes. The investigation was carried out in the following range of changes of parameters: $Fr_M = 57$–1082, $Re_f = (0.05$–$21) \cdot 10^4$, $q_0 = (1.2$–$1.9) \cdot 10^4$ W/m², $T_c \leq 780$°K, $T_f = 460$–610°K, $T_c/T_f \leq 1.45$. The maximum error in determining the coefficient α_m was ±15%.

To determine the coefficient α_m it was chiefly necessary for each experimental regime characterized by the Re and q_0 numbers to measure the distribution of the flow velocities and temperatures of the heat carrier and of the walls of the spiral tubes over the radius of the bundle. Typical experimentally measured distributions $\bar{u}$, $\bar{T}$, and T_c over the radius of a bundle with 127 tubes are presented in Figs. 2 and 3. Taking this distribution into account, the values of α_m and Z_m for the specified radius are calculated in the following way. The dimensional heat-transfer coefficient $\bar{\alpha}$ is introduced:

$$\bar{\alpha} = \frac{q_0}{(T_c - \bar{T})} \tag{5}$$

and the similarity criteria are determined

$$\overline{Re}_{\delta n} = \bar{u}\bar{\rho}\delta/\mu_f, \tag{6}$$

$$\overline{Re}_{\delta m} = \frac{\bar{u}\bar{\rho}\delta}{\mu_m}\left(\frac{\bar{T}}{T_m}\right), \tag{7}$$

$$\overline{Nu}_{\delta m} = \bar{\alpha}\delta/\lambda_m, \tag{8}$$

where

$$\bar{\rho} = P/R\bar{T};\quad \mu_f = \mu(\bar{T});\quad T_m = (T_c + \bar{T})/2;$$
$$\mu_m = \mu(T_m);\quad \lambda_m = \lambda(T_m).$$

The mean integral thickness of the near-wall layer is determined by the formula [4]

$$\delta = 0.5\left(1 + \frac{3.6}{Fr_M^{0.357}}\right)^{-4} d_e. \tag{9}$$

In [3, 4] it was shown that the coefficient of hydraulic resistance does not depend on the magnitude of T_c/T_f. Therefore, in accordance with [4], the friction coefficient can be determined by using the formula

$$\alpha = \frac{0.266\,(1 - 4\delta^*/d_e)^{1.75}}{8\overline{Re}_{\delta f}^{0.25}}, \tag{10}$$

where δ^*, in accordance with [8], can be determined by the dependence

$$\delta^* = \frac{1.3 Z_m \delta \mu_m \alpha}{Re_{\delta f}\,\mu_f}. \tag{11}$$

Then we transform expression (4) into the form

$$Z_m = \overline{Re}_{\delta m}\left(1 - \frac{2\sqrt{\alpha}}{0.39}\right)\Big/ 0.39\sqrt{\alpha}. \tag{12}$$

In determining the dependence (10), the following correlation between the maximum velocity $\bar{u}$ (the velocity in the flow core) and the mean mass velocity u_{me} was taken into account:

$$u_{me}/\bar{u} = 1 - 4\delta^*/d_e. \tag{13}$$

The sought value of α_m (2) can be expressed through the criterion $\overline{Nu}_{\delta m}$ (8) and $\overline{Re}_{\delta m}$ (7) in the following way:

$$\alpha_m = \overline{Nu}_{\delta m}/\overline{Re}_{\delta m}\,Pr_m. \tag{14}$$

The experimental data on the heat transfer coefficient α_m in dependence on the determining criterion Z_m are presented in Fig. 4, where they are being compared with the dependence

$$\alpha_m = (30.4 Z_m^{0.174} Pr_m^{0.6} + 14.65 Z_m^{0.09} - 11.2)^{-1}, \tag{15}$$

obtained in [4] for the uniform heating of spiral tubes over the radius of the bundle. The agreement of the experimental data on the coefficient α_m for Fr_M numbers ≈ 200–1082 and non-uniform heat supply with dependence (15) is good. The experimental data on the coefficient α_m for $Fr_M = 64$ are ≈20% higher than those calculated by (15); this is in agreement with the data of [3]. Evidently, when $Fr_M < 200$, there occurs additional turbulization of the flow on account of the flow being detached from the spiral surfaces of the tubes; this is not encountered in the range of change of the Fr_M numbers between 200 and 1082.

On the basis of the above we may conclude that the suggested method of experimental investigation of local heat transfer in bundles of spiral tubes makes it possible to determine with sufficient accuracy the heat-transfer coefficients upon nonuniform heat supply to the heat carrier over the radius of the bundle. The obtained results concerning the coefficient α_m indicate that it is possible to use a homogenized flow model for calculating heat transfer by the local flow characteristics using the law of heat transfer (15).

NOTATION

S, spiral pitch of the profile of the spiral tube; d, maximum dimension of the profile; $d_e = 4F/B$, equivalent diameter; F, clear cross-sectional area of the bundle; L, length of the bundle; B, wetted perimeter of the bundle; Re, Reynolds number; Fr_M, criterion characterizing the effect of centrifugal forces on the flow in a bundle of spiral tubes; δ^*, thickness of the boundary layer displacement; $\bar{T}$, temperature in the flow core; $\bar{u}$, velocity in the flow core; T_c, wall temperature; α_m, dimensionless heat-transfer coefficient; q_o, specific heat flow on the wall; ρ, density; c_p, specific heat; Pr, Prandtl number; λ, thermal conductivity; μ, viscosity; δ, thickness of the near-wall layer; P, pressure; α, coefficient of friction; Z_m, specially constructed Reynolds number; r, radial coordinate of the bundle; r_K, radius of the bundle jacket. Subscripts: m, physical properties of the heat carrier determined at the mean temperature of the near-wall layer; f, physical properties of the heat carrier determined at the flow temperature.

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