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NUMERICAL MODELING OF LAMINAR SEPARATION FLOW AND HEAT EXCHANGE IN TUBE BANKS WITH THE USE OF MULTIBLOCK COMPUTATIONAL TECHNIQUES

S. A. Isaev, P. A. Baranov, and N. A. Kudryavtsev

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A multiblock approach to the solution of steady-state Navier–Stokes equations has been approved and an original procedure of mean-mass temperature correction has been proposed for calculation of separation flow and heat exchange an in-line bank of round tubes.

For years the designing of heat-exchange apparatus and investigation of physical processes occurring in them have attracted the attention of specialists in thermal physics and heat engineering [1, 2]. It should be noted that up to now the experimental and semiempirical integral methods based on simple criteria relations for the heat-transfer coefficient have been prevailing in this field of convective heat transfer.

A few computational works carried out at the initial stage of development of numerical modeling of hydrodynamics and heat exchange (CFD) were generalized fifteen years ago in monograph [3]. Since then, no investigation of such a scale has appeared despite the impressive progress made in the field of computational technics associated, first of all, with the appearance and rapid development of personal computers and the improvement of computational techniques (multiblock, in particular, [4]) realized in universal and specialized packets of applied programs or in codes, including "heavy" packets developed for hydrodynamic and thermophysical purposes: FLUENT, Star CD, CFX, and others.

The calculation of a heat-exchange apparatus representing an ordered bank of heated elements identical in shape is based on the assumption that flows of identical structure are formed in the repetitive isolated modules, of which, as of boxes, the bank is composed (see, for example, [3]). Periodic boundary conditions are set at the flow boundaries of each of the modules. It is assumed that the velocity profiles are the same in the geometrically similar cross sections. This formulation of the problem is fully suited for multirow heat exchangers, especially for the elements positioned deep in the bank.

The genesis of numerical modeling of viscous fluid flow about elements periodic in distance is closely related to the evolution of numerical methods of solving Navier–Stokes equations. At the initial stage of development of numerical modeling of hydrodynamics and heat exchange, performed with the personal computers then available characterized by low storage capacity and a low speed of response, the initial equations were written, for economy of computational resources, in transformed variables: vorticity–stream function. For the same purpose, the problems with limited dimensions of the computational regions, in particular the problems with periodic boundary conditions, such as problems on viscous fluid motion in a channel with bulges periodic in distance, were considered [5]. It should be noted that the use of the stream function as a the dependent variable made the solution of the problem much simpler because, in this formulation the fluid flow rate in the flow section was fixed automatically [6].

In the 1970s, the progress that had been made in computational resources was favorable for wide approbation of semiempirical differential models of turbulence, in particular, the two-parameter dissipative model. Because of the difficulties associated with the formulation of correct boundary conditions in transformed variables, pressure and velocity components had come into use as dependent variables. However, there arose a problem characteristic of the calculation of channel-type flows and associated with the determination of the pressure difference corresponding to the

Academy of Civil Aviation, 38 Pilotov Str., St. Petersburg, 196210, Russia; email: isaev@S13612.spb.edu.
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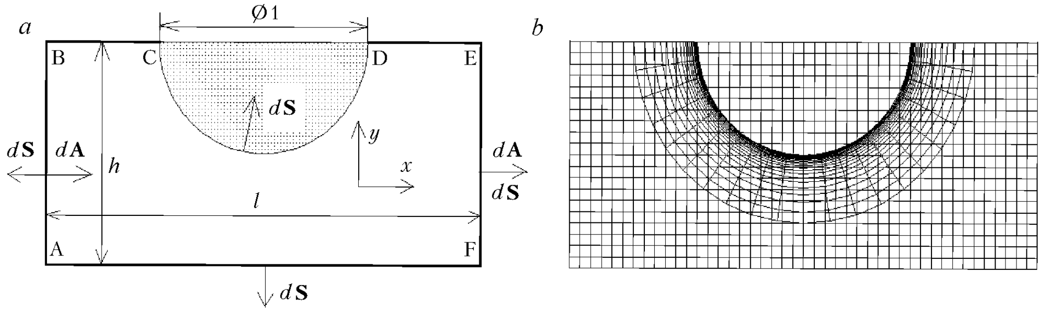

Fig. 1. Scheme of the computational region (a) and multiblock computational grid (b).

prescribed fluid flow rate. The solution of the heat problem presents analogous difficulties: heating of the fluid in a channel is enhanced with increase in the number of heated elements periodic in distance, and this process is nonlinear. The indicated problems have been partially resolved in [7–9], where problems on convective heat exchange in laminar longitudinal and transverse flows about blocks of thin and thick plates were considered.

In subsequent investigations [10–12], the methodology of solving problems with periodic boundary conditions has been developed as applied to tube heat exchangers, container transport, and instrument engineering. Of special interest is the proposed interpretation of the iteration procedure of correction of the pressure gradient related to the prescribed fluid flow rate. Until the present time this procedure has not experienced substantial changes and has been formalized in the majority of applied-program packets developed. However, the heat problem has not been solved adequately. In [11], it is proposed to determine the dimensionless temperature with the use of the pressure difference between the neighboring vertical rows of tubes, which makes it possible to compare heat transfer in tubes of different various arrangement only qualitatively. The approach proposed in [8] is based on the construction of an ordinary differential equation for the mean-mass temperature gradient and presents great difficulties when it is used for multiblock curvilinear grid structures.

The present work is, on the whole, methodological in character. In it, the aim is to propose a procedure of mean-mass temperature correction, approve a multiblock computational algorithm by the example of modeling of laminar flow and heat exchange in an in-line bank of roundtubes, and investigate the influence of the viscosity on the characteristics of the flow and heat exchange.

Features of the Formulation of the Procedure of Mean-Mass Temperature Correction. Patankar et al. [8] argue that, in the case where the temperature of the walls within a region is constant (see. e.g., Fig. 1a), the periodic component of the temperature field can be separated using the expression

$$\theta = \frac{T(\mathbf{r}) - T_{\mathrm{w}}}{T^{*}(x) - T_{\mathrm{w}}}, \quad \text{in this case } \theta(x) = \theta(x + L) = \theta(x + 2L)\ldots, \tag{1}$$

where

$$T^{*}(x) = \frac{\int\limits_{A} T(\mathbf{r})\, |\rho \mathbf{v}\cdot d\mathbf{A}|}{\int\limits_{A} |\rho \mathbf{v}\cdot d\mathbf{A}|}, \tag{2}$$

and the integrals should be taken over the region cross section normal to the direction of the main flow. In the case where a reverse flow is absent, the quantity $T^{*}(x)$ represents the mean-mass temperature. The distribution of this quantity over the region is not known in advance and should be determined using an iteration procedure in the process of solution.

For simplicity, we will assume that the distribution of $T^{*}(x)$ over the region is linear, i.e.,

$$T^{*}(x) = T^{*}_{\mathrm{inlet}} + \beta_T x. \tag{3}$$

Let us also assume that reverse-flow regions are absent in the inlet cross section or they are small; therefore, it may be assumed that $T^{*}_{\text{inlet}} = T^{\text{b.t}}_{\text{inlet}}$, where

$$T^{\text{b.t}}_{\text{inlet}} = \frac{\int\limits_{\text{inlet}} T(\mathbf{r})\, \rho \mathbf{v}\cdot d\mathbf{A}}{\int\limits_{\text{inlet}} \rho \mathbf{v}\cdot d\mathbf{A}}$$

is the inlet mean-mass temperature.

To determine the unknown mean-mass temperature gradient, we use the energy equation written in the form

$$\nabla\cdot(\rho \mathbf{v} T) = \alpha \nabla^2 T .$$

Integrating it over the entire computational region, we obtain

$$\int\limits_{\text{inlet}} T\rho \mathbf{v}\cdot d\mathbf{S} + \int\limits_{\text{outlet}} T\rho \mathbf{v}\cdot d\mathbf{S} = \int\limits_{\text{w}} \alpha \nabla T\cdot d\mathbf{S} ,$$

where the vector $d\mathbf{S}$ is directed along the outer normal to the boundary of the region, or

$$-\int\limits_{\text{inlet}} T\rho \mathbf{v}\cdot d\mathbf{A} + \int\limits_{\text{outlet}} T\rho \mathbf{v}\cdot d\mathbf{A} = -\int\limits_{\text{w}} \alpha \frac{\partial T}{\partial n}\, ds ,$$

here, the derivative on the right side is taken with respect to the normal to the wall.

Taking into account (3), we write

$$-T^{\text{b.t}}_{\text{inlet}} \int\limits_{\text{inlet}} \rho \mathbf{v}\cdot d\mathbf{A} + (T^{\text{b.t}}_{\text{inlet}} + \beta_T L) \int\limits_{\text{outlet}} \rho \mathbf{v}\cdot d\mathbf{A} = -\int\limits_{\text{w}} \alpha \frac{\partial T}{\partial n}\, ds .$$

Since, because of the periodicity of the flow,

$$\int\limits_{\text{inlet}} \rho \mathbf{v}\cdot d\mathbf{A} = \int\limits_{\text{outlet}} \rho \mathbf{v}\cdot d\mathbf{A} ,$$

the expression for β_T takes the form

$$\beta_T = -\frac{\int\limits_{\text{w}} \alpha \frac{\partial T}{\partial n}\, ds}{L \int\limits_{\text{inlet}} \rho \mathbf{v}\cdot d\mathbf{A}} . \tag{4}$$

In the process of iteration solution of the problem, in addition to the determination of the mean-mass temperature gradient, it is necessary to maintain a predetermined temperature of the incoming flow (inlet mean-mass temperature).

If the inlet mean-mass temperature is equal to the predetermined temperature, the quantity

$$\int\limits_{\text{inlet}} \theta(\mathbf{r}) \left|\rho \mathbf{v}\cdot d\mathbf{A}\right| = 1 .$$

Otherwise, the dimensionless-temperature field is corrected as

$$\theta^{\text{new}}(\mathbf{r}) = \frac{\theta^{\text{old}}(\mathbf{r})}{\int\limits_{\text{inlet}} \theta(\mathbf{r}) \left|\rho \mathbf{v}\cdot d\mathbf{A}\right|} .$$

Formulation of the Problem. Two-dimensional motion of an incompressible viscous fluid and convective heat exchange in a lane bank of cylindrical tubes of round cross section arranged with longitudinal and transverse steps l and h are numerically analyzed. The diameter of the cylinder D is taken as the linear scale and the mean-mass velocity U is taken as the characteristic velocity. An ABCDEF computational module is separated. Periodic boundary conditions are set at its AB and EF boundaries and symmetry conditions are set at the AF and BE boundaries. The adhesion conditions are fulfilled on the heated isothermal CD wall of the cylinder about which fluid flows. The heat problem and the dynamic problem are solved separately using the velocity fields calculated in advance. The Reynolds number is varied from 40 to 10^3 and the Prandtl number is varied from 0.7 to 4000. The geometric dimensions of the computational module are taken as $l = 2$ and $h = 1$. The wall of the cylinder is assumed to be isothermal and heated (it is equal to 1.27 in the dimensionless form). The mean-mass temperature in the inlet cross section is taken as the characteristic temperature (it is taken to be 293 K in the dimensional form; in this case, the overheating of the tube is 100^{o}).

A multiblock grid (Fig. 1b) is used. It consists of two different-scale grids having a different structure: a Cartesian grid (with uniform pitches in the transverse direction equal to 0.02) covering the entire computational region and a polar grid (with a uniform pitch along the peripheral coordinate and a nonuniform pitch along the radial coordinate) extending a distance of 0.15 from the surface of the cylinder. The near-wall pitch is taken to be 0.0001. The number of cells in the peripheral direction of the polar grid is varied from 40 to 90.

Interpretation of the Multiblock Computational Algorithm. Laminar flow and heat exchange in the computational module separated are described based on the finite-volume solution of the Navier–Stokes equations and the energy equation by the factorized implicit method of global iterations constructed within the framework of the concept of splitting by physical processes [13]. This original method in completed form is described in detail for multiblock grids in [14].

The core of the computational algorithm is the known SIMPLEC procedure of pressure correction. This two-step procedure of the "predictor–corrector" type is meant for determining the Cartesian velocity and pressure components. The original features of the finite-volume algorithm developed in the late 1980s are associated with (a) representation of the initial equations in increments of dependent variables, (b) approximation of convective terms in the explicit side of the momentum equation by the univariate Leonard upwind scheme with quadratic interpolation, (c) approximation of the convective terms in the implicit side by the upwind scheme with one-sided differences, (d) introduction of artificial diffusion into the implicit side for damping of the high-frequency oscillations with a coefficient of increasing the kinematic viscosity OTL (>1), (e) use of the Rhie–Chow monotonizer in the block of pressure correction because of the centered computational template with an empirically determined coefficient of 0.1, and (f) solution of the difference equations by the method of incomplete matrix factorization in the Stone version (SIP).

Construction of a multiblock algorithm is associated with numerical modeling of vortex flows in multiply connected regions within the framework of the approach based on the decomposition of a computational region of complex geometry (Fig. 1) into fragments followed by use of intersecting grids of simple topology. The parameters in the region of intersection of grids are determined using the procedure of linear interpolation [15]. The multiblock computational technoloques developed initially for the calculation of a two-dimensional flow about bodies with vortex cells have been verified (see, e.g., [16, 17]) and extended to the case of steady-state, three-dimensional turbulent flows with flow separation [4].

A feature of the formulation of the problem under consideration is that periodic components are separated from the pressure and temperature fields [3]. It should be noted that these periodic components are used as dependent variables in the construction of the equation for pressure and energy correction. In addition to the successive iteration solution of the Navier–Stokes equations and the equation for pressure correction derived from the continuity equation, additional iteration cycles are introduced into the procedure of global iterations. The pressure difference between the inlet and outlet cross sections is determined with the use of the known iteration procedure of pressure-gradient correction based on the constancy of the flow rate [13], and the heat problem is solved using the methods of mean-mass temperature correction described in the present work.

The relaxation coefficients are taken to be 0.5, 0.8, and 0.9 in the calculations the increments of the velocity components, the pressure corrections, and the temperature increment, respectively. The E factor is taken, as ever, to be 2.5 in the calculation of the dynamic problem and 100 in the calculation of the heat problem [13, 14] to provide high convergence in the latter case.

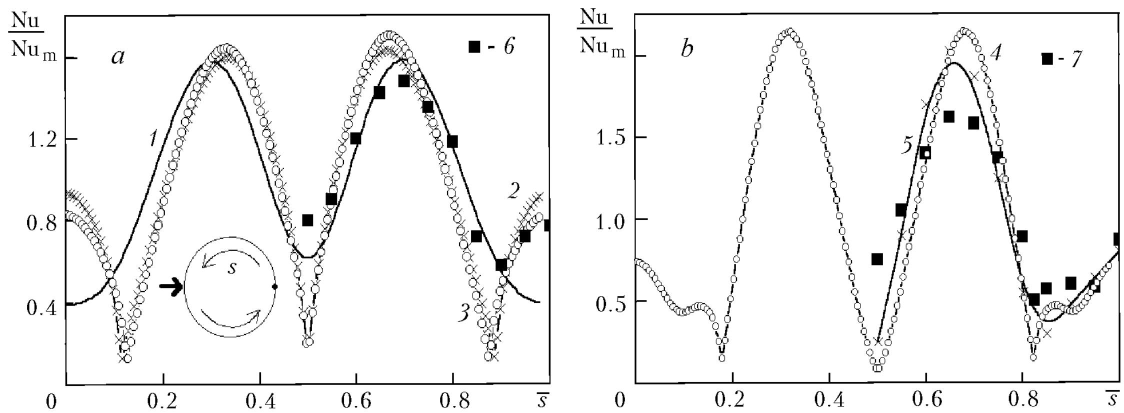

Fig. 2. Distribution of the local Nusselt number related to the contour-average Nusselt number over the surface of a tube in the in-line bank for laminar flows with low (a) and high (b) Reynolds numbers. Calculation curves: 1) Re = 40 and Pr = 0.73; 2) 60 and 0.73; 3) 60 and 4000; 4) 10^3 and 321; 5) 10^3 and 0.73 [3]; experimental data: 6) Re = 57 and Pr = 3947; 7) 1140 and

Testing of the Computational Algorithm. Our main concern in the present work was the verification of the multiblock algorithm developed. For this purpose, the calculated local heat flows on the surface of a tube in the in-line bank selected were compared to the calculated [3] and experimental [2] data obtained at close regime parameters. The wholly satisfactory agreement between the data presented in Fig. 2 at low and high Reynolds numbers points to the acceptability of the approach developed, including the case of laminar flow of a medium such as oil.

It should be noted that the distribution of local heat flows in the separation zone between the cylinders changes abruptly when the Reynolds number increases from 40 to 60, and the local minimum of heat transfer is positioned at the point of flow separation. It is notable that the influence of the Prandtl number is not very marked, which lends support to the validity of the comparison of the calculation data on heat transfer obtained for air in [3] with the experimental data obtained for oil media in [2].

Investigation of the Viscosity Effect. Some of the calculation data on the influence of Re on the structure of the separation flow about a cylinder from the in-line bank under consideration on the local and integral characteristics of the drag and heat exchange are presented in Figs. 3–5. The working medium is air (Pr = 0.73) in all the cases.

As noted in a number of works (see, e.g., [14, 15]), an increase in the Reynolds number in a flow about a cylinder leads to a progressing intensification of the separation flow, an increase in the dimension of the circulation zone in the near wake, and a decrease in the drag. In the case of flow about a cylinder positioned in a in-line bank, in which the tubes are arranged fairly closely together, as in the case under consideration ($h = l/2 = 1$), undoubtedly the blocking of the flow section exerts a great action on the formation of the flow structure, i.e., the flow in the computational module is of the channel type. However, as is seen from Fig. 3, the characteristic features of the influence of Re, noted for an unbounded flow, remain in the case of fluid motion in a tube bank too. Even though the blocking effect, appearing when the separation zone occupies the entire space between the cylinders, is observed at even R = 40, the transverse dimension of the zone and the velocity of the reverse flow increase with increase in the Reynolds number. It should be noted that the largest changes in the vortex structure are observed at small and moderate Re, whereas in the ranges of high (of the order of 500 and higher) Re the dimensions of the circulation region stabilize.

It is important to determine the interrelation between the evolution of the vortex structure and the changes in the temperature field caused by an increase in Re. As the Reynolds number increases, the core of a large-scale vortex is gradually heated, which serves to decrease heat transfer in the separation zone. At the same time, in the core of the flow part of the module the temperature field becomes layered in character and somewhat cooled in the central zone (the temperature is approximately 5% lower than the mean-mass temperature), and the temperature layer near the part of the cylinder surface about which fluid flows without separation becomes gradually thinner, which increases heat transfer in this region.

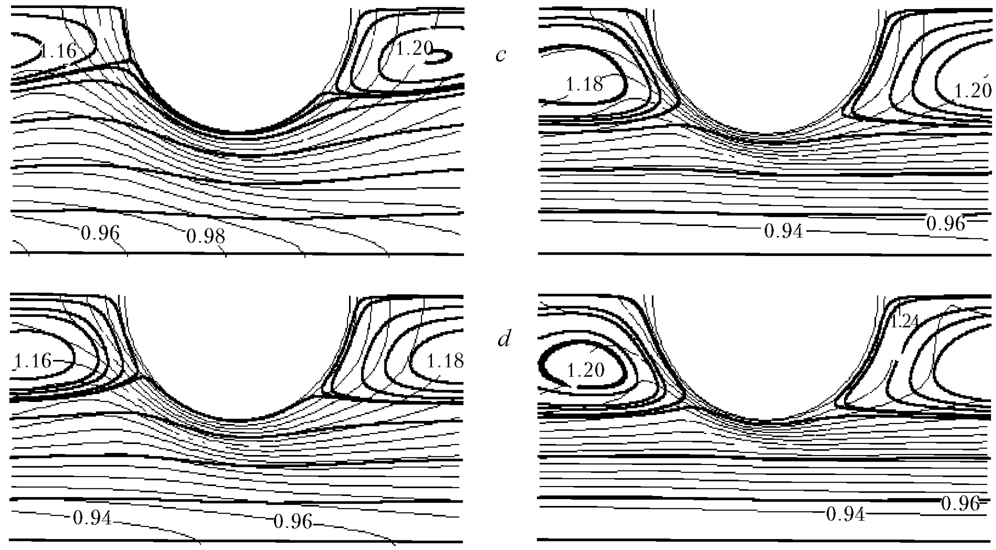

Fig. 3. Evolution of the temperature fields (the isotherms are drawn with a step of 0.02) and the pattern of flow (the lines of flow are shown by heavy lines) about a cylinder in the tube bank with increase in the Reynolds number: a) Re = 40; b) 100; c) 250; d) 500.

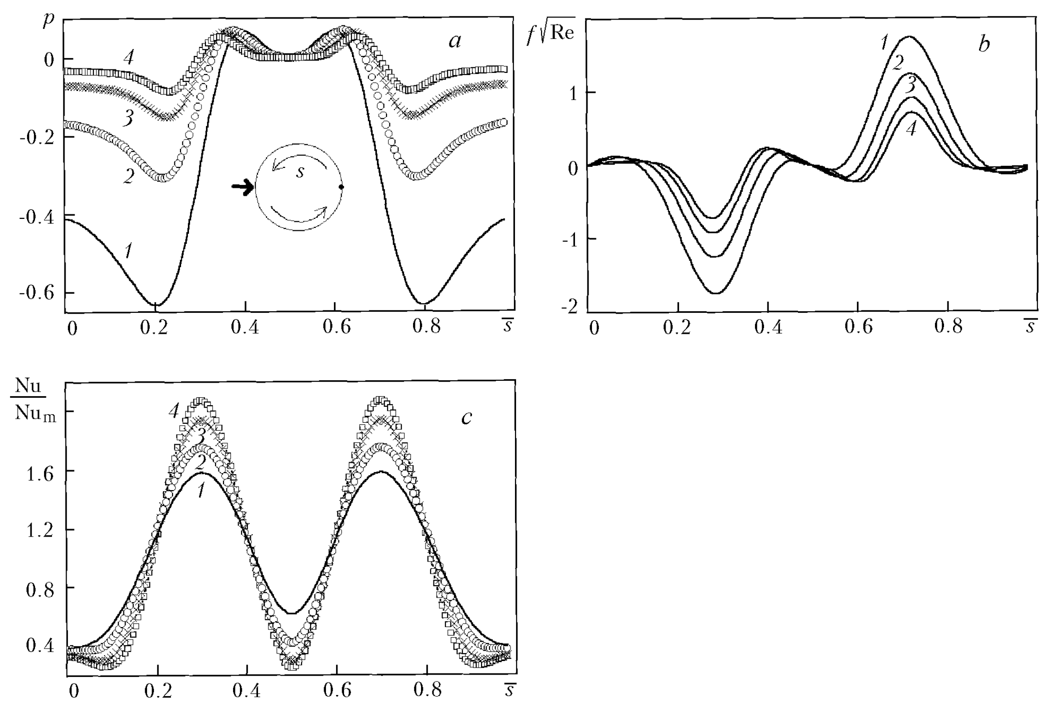

Fig. 4. Evolution of the surface distributions of the pressure (a), friction stress (b), and local Nusselt number related to the perimeter-average Nu_m (c) with increase in the Reynolds number: 1) Re = 40; 2) 100; 3) 250; 4) 500.

In interpreting the pressure profiles (Fig. 4a), all were referred to the front critical point of the cylinder (p = 0). First of all, the large pressure differences between the back and front critical points at law Reynolds numbers, caused by the large pressure gradients in the module selected, have engaged our attention. In this case, the pressure is minimum in the region of the flow-separation point and is maximum in the zone of flow attachment. As Re increases, the indicated pressure extrema approach each other, since, as suggested in Fig. 3, the dimensions of the separation

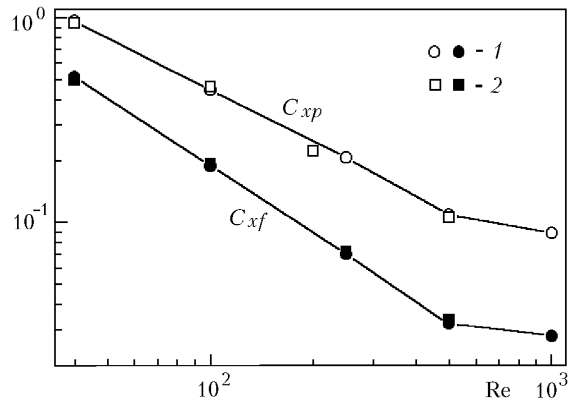

Fig. 5. Dependence of the coefficients of pressure resistance C_{xp} and friction resistance C_{fx} of a tube in the in-line bank on Re: 1) present work; 2) [3].

TABLE 1. Dependence of the Nusselt Number Averaged over the Perimeter of the Tube and the Drag Coefficient of the Cylinder on the Reynolds Number

Re	40	100	250	500	1000
Nu_m	3.351	3.415	3.646	3.823	4.42
C_x	1.48	0.635	0.279	0.142	0.117
C_x [3]	1.44	0.656	0.296	0.140	–

zone increase and the separation and attachment points move sideways from the symmetry plane of the cylinder. At high Reynolds numbers, the differences between the extremum values of the pressure also decrease and the profiles in the separation zone smooth out, i.e., the separation zone becomes isobaric.

An analysis of the surface distributions of the friction stress (Fig. 4b) confirms the earlier assumption that the intensification of the flow in the separation zone increases with increase in Re and shows that the extremum values of the friction in the zones of separation flow and flow without separation about the cylinder approach each other.

The evolution of the profiles of relative local heat transfer from the cylinder (Fig. 4c) points to the following tendency: the level of heat transfer in the separation zone decreases monotonically (in this case, the local minimum moves sideways from the symmetry plane) with gradual increase in the maximum of heat transfer in the zone of flow without separation. The reasons for such behavior of heat transfer were discussed above when the temperature fields were analyzed (Fig. 3). Even though the above-mentioned processes of change in the local heat transfer are opposite in character, they compensate each other and, in doing so, cause weak intensification of heat exchange (see Table 1). An opposite tendency in the behavior of Nu_m(Re) was noted in [3], which is apparently due to the insufficiently correct calculation of the temperature fields.

As is seen from Fig. 5, the total force loads on the circular cylinder in the tube bank behave with increase in Re in the same manner as in the case of a free space [14]. As already mentioned, the blocking effect characteristic of a flow about a in-line bank of tubes progresses with increase in Re, which leads to a gradual decrease in the drag. A comparison of the drag coefficients (see Table 1) and drag components calculated in the present work and in [3] shows that they are in close agreement, which can be considered as an additional guide for the verification of the multiblock algorithm developed.

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NOTATION

C_x, C_{xp}, and C_{xf}, coefficients of frontal resistance, pressure resistance, and friction resistance, respectively; c_p, heat capacity at a constant pressure, kJ/(kg·K); D, diameter of the cylinder, m; $d\mathbf{S}$ and $d\mathbf{A}$, vectors of elemental areas along the boundaries of the computational module and at the flow boundaries ($d\mathbf{S}$ coincides with the direction of the outer normal), m; E, relaxation parameter (E factor); f, friction stress related to the double kinetic head, fractions of ρU^2; L, dimensional length of the computational module, m; l and h, length and width of the computational module, fractions of D; Nu, Nusselt number determined as $d\theta / dn$; x and y, axial and radial coordinates, m; Pr, Prandtl number, $\mathrm{Pr} = c_p \mu / \lambda$; p, pressure related to the double kinetic head, fractions of ρU^2; Re, Reynolds number, $\mathrm{Re} = \rho UD / \mu$; $\mathbf{r}$, radius-vector, m; s, $\bar{s}$, and n, coordinates measured along the contour of the body about which fluid flows along the length of the perimeter and on the normal to it, m; T, temperature, K; U, mean-mass velocity, m/sec; $\mathbf{v}$, velocity vector, m/sec; α, thermal diffusivity, m^2/sec; β_T, mean-mass temperature gradient; λ, heat-conductivity coefficient, W/(m·K); μ, viscosity, Pa·sec; θ, dimensionless temperature; ρ, density, kg/m^3; ∇, Hamiltonian. Subscripts: inlet, parameters in the inlet cross section; outlet, parameters in the outlet cross section; m, averaging over the cylinder parameter; w, perimeters on the wall; b.t, mean-mass temperature; *, characteristic integral temperature in the absence of separation equal to the mean-mass temperature; overscribed bar, dimensionless quantity expressed in fractions of the cylinder perimeter; old and new, values of a parameter in previous and subsequent iterations.

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