Journal of Engineering Physics and Thermophysics, Vol. 81, No. 4, 2008

NUMERICAL SIMULATION OF LAMINAR FLOW OF OIL AND HEAT TRANSFER NEAR A CIRCULAR CYLINDER WITH ARCHED GUIDE PLATES

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UDC 532.517.2

We have investigated the intensification of the heat transfer from a cylinder in an oil medium due to the setting of arched guide plates on the basis of the numerical solution of the Navier–Stokes and energy equations with the help of multiblock computer-aided technologies realized in the VP2/3 package. Calculations have been made for the laminar nonstationary cross flow around a heated cylinder in a viscous medium with a strong temperature dependence of its thermophysical characteristics. Comparison has been made between the numerical forecasts of the drag, local heat transfer, thermohydraulic efficiency, and total Nusselt number for a cylinder with plates and for a single cylinder.

Introduction. Intensification of the heat and mass transfer in power plants is a topical problem in modern thermophysics [1]. Many methods of intensifying the heat-transfer processes correspond to the analogous approaches to the control of the flow past bodies with flow separation [2]. One nontraditional control method consists of using the throttling effect for the jet-vortex generation with fluid flow in the jacket from the higher-pressure region in the front part of the body into the low-pressure region in the near wake [3]. In a series of numerical studies [4–8], it has been shown that in the flow of air around a circular cylinder with an outflow of low-head jets into the bottom part of bodies being flown, a change in the vortex structure and a considerable decrease in the lateral alternating load occur. An increase in the window sizes leads to the transformation of the outer shell of the cylinder into arched guide plates, which, when positioned in the sternmost part of the cylinder, resemble spoilers [9]. In [10, 11], the results of the numerical simulation of nonstationary heat transfer in a laminar air flow around a circular heated cylinder are presented and an estimate of the influence on the heat transfer of the geometry of arched guides is given.

One of the most important problems remains the development of computing methods for predicting the characteristics of the flow and heat transfer near circular cylinders in heterogeneous media [12]. The multiblock computing technologies (MCT) developed in [3, 13] for solving Navier–Stokes (Reynolds) and energy equations have been realized in the VP2/3 package (velocity–pressure, two-dimensional and three-dimensional versions) intended for hydrodynamics and energetics problems. In [14], the MCT and VP2/3 have been modified for solving related problems of convective heat transfer near circular cylinders in heterogeneous media (of the type of oils) with account for the thickness of the walls and their thermophysical properties.

The aim of the present work is to estimate the influence of arched guide plates on the intensification of convective heat transfer near a heated cylinder in an oil medium.

Formulation of the Problem. We consider a laminar nonstationary flow of an incompressible fluid around a circular cylinder at a Reynolds number Re = 186 in the absence of influence of mass forces. The temperature difference is 80^{o}. The choice of the Reynolds number is due to the fact that the three-dimensional character of the flow in the wake manifests itself beginning with Re > 190 [15]. It is known that the velocity and pressure distributions in the shear layers formed by shedding vortices are determined by the quasi-two-dimensional vortices. Direct and indirect measurements [16–18] have shown that the detached flow behind the cylinder at Re = 186 is two-dimensional and is adequately described in this case by Navier–Stokes equations from the point of view of the two-dimensional approach.

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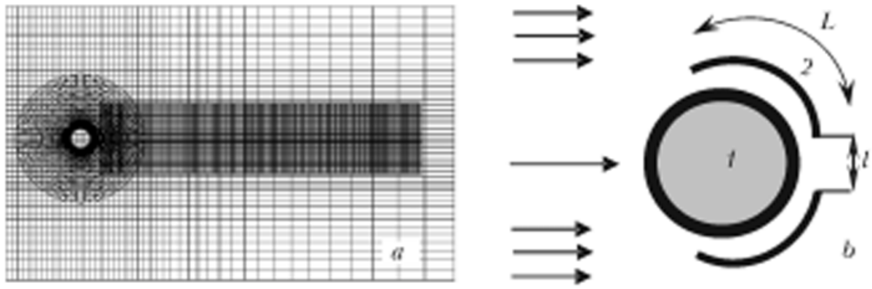

Fig. 1. Calculation grid (a) and object of investigation (b): 1) circular cylinder; 2) guide elements.

The flow in the wake behind a heated cylinder is physically a more complex process because of the buoyancy phenomenon existing in addition to the viscous effects. Heat transfer by the fluid from the heated cylinder surface can be realized by forced convection, free (natural) convection, or mixed convection, depending on the Richardson number [19]. In the forced-convection regime (Ri << 1), where the buoyancy phenomenon is negligibly small, the heat transfer is only determined by the Reynolds and Prandtl numbers. In the free-convection regime (Ri >> 1), where the forced convection is negligibly small, the heat transfer is determined by the Grashof and Prandtl numbers. In the case of mixed convection conditions, taking account of both the free and mixed convection is important and, therefore, the heat transfer is characterized by the Grashof, Reynolds, and Prandtl numbers, as well as by the chosen directions of the flow.

The Richardson number in the present case was determined as $\mathrm{Ri} = \mathrm{Gr}/(\mathrm{Re})^2$, and the Grashof number as $\mathrm{Gr} = gD^3\beta\Delta T/\nu^2$. The coefficient of volumetric expansion at a constant pressure p can be given as $\beta = -\frac{1}{\rho}\left(\frac{\partial\rho}{\partial T}\right)$ [20], and for the case of the cross flow of MS-20 oil around a heated cylinder it is $\beta_{\mathrm{oil}} = 6.37\cdot10^{-4}$. Thus, $\mathrm{Gr}_{\mathrm{oil}} = 5.95\cdot10^{-3}$ and, accordingly, $\mathrm{Ri}_{\mathrm{oil}} \sim 2\cdot10^{-7}$. The estimate of the Richardson number permits concluding that in the considered case of the cross flow of MS-20 oil around a heated cylinder the forced convection regime is realized and the buoyancy effect can be neglected.

The equations are solved in dimensionless form; the coordinates are dedimensionalized to the characteristic scale D (cylinder diameter), the velocity — to the characteristic velocity U_0, the pressure — to the double velocity head ρU^2, and the time — to the characteristic time scale. The specific features of the solution are given in [13]. The outer boundaries of the calculation region are fairly distant from the surface of the investigated object. At the inlet boundary of such a region the incident flow parameters are given, and at the outlet boundaries the boundary conditions are mild. On the flown-over surfaces of the investigated object the adhesion conditions are realized.

The calculations were carried out on a multiblock grid [13] consisting of an external rectangular mesh, a three-layer mesh for describing the region near the cylinder, an additional mesh for describing the wake, and meshes for describing the region of the guide elements (Fig. 1a). The external rectangular mesh of size $(15.2 \times 17.2)D$ contained 150×124 cells with a minimum width of $0.005D$ before the investigated object, in the middle and in the region of the near wake, and $0.02D$ for the region of the far wake. In the calculation region, near the cylinder surface a three-layer polar mesh made to agree with the center of the cylinder was used. The ring zone adjoining the cylinder surface had a thickness of $0.1D$ and contained 300×26 cells with a minimum width of $0.001D$. The next layer of the polar mesh covered a distance of $0.4D$ from the cylinder surface and included 300×20 cells; the external layer of the polar mesh extended for a distance of $0.5D$ and consisted of 300×14 cells. For a detailed description of the wake behind the cylinder an additional rectangular mesh of size $(20 \times 2)D$ containing 468×125 cells with a minimum width of $0.025D$ was used. For the cylinder with guide plates, additional meshes for the guide elements with 93×47 cells and a minimum width of $0.017D$ were constructed.

To simulate the heat transfer, tabular values of the variables depending on the temperature, density, heat conductivity, and heat capacity were introduced. The enthalpy was calculated as $c_p(T)/T$. In the course of the calculations the Prandtl number was determined as a function of temperature.

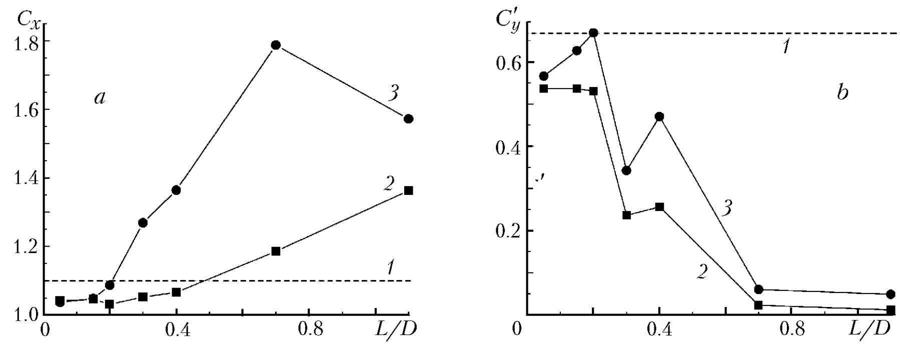

Fig. 2. Total drag coefficient of the cylinder C_x (a) and amplitudes of the lift coefficient of the cylinder C'_y (b) as a function of the plates: 1) single cylinder; 2) cylinder in the cylinder–plates system; 3) cylinder with plates.

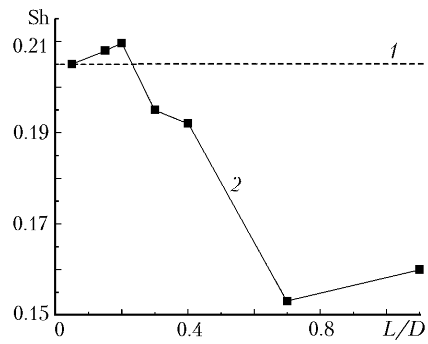

Fig. 3. Strouhal number as a function of the length of the plates: 1) single cylinder; 2) cylinder with plates.

In the calculations made, the Strouhal number was calculated from the oscillation period of the lift coefficient. Thus, from the determination of the Strouhal number a value of Sh = 0.2055 has been obtained. The discrepancy between this value and the value of Sh = 0.2 obtained experimentally in [20] is 3%.

The calculated value of the total drag coefficient of the single cylinder C_x = 1.1 (with an accuracy of ~9%) agrees with the value of C_x = 1.199 obtained on the basis of the formula $C_x = -2.2 + 2.5/(\text{Sh} + 0.53)$ from [21].

The aim of the present study was to establish the relations hip between the hydrodynamic and thermal characteristics at a laminar cross flow around a cylinder with arched guide plates of a viscous medium with a strong temperature dependence of the thermophysical characteristics.

The object of the investigation was a cylinder with guide plates which had an equal length and were equidistant from the outer surface of the cylinder and the frontal critical point (Fig. 1b), forming in this case channels for the flow of the medium. The plate length was varied from $L = 0.05D$ (minimum) to $L = 1.1D$ (maximum), and the coordinate of the rear edge of the plate was held constant (the size of the clearance between the plates was $l = 0.03D$).

Results and Discussion. The total drag coefficient C_x and the oscillation amplitude of the lift coefficient C'_y depend on the character of the flow around the cylinder (Fig. 2). An increase in the size of the vortex zones leads to a loss of the kinetic energy of the flow and an increase in the drag of the whole of the system.

With decreasing length of the plates the total drag coefficient of the cylinder decreases, and the least drag of the whole of the system thereby is attained already with plates of length $L < 0.2D$ and constitutes ~95–98% of the total drag of the single cylinder (Fig. 2a). It should be noted that as the length of the plates is further decreased the drag of the cylinder in the cylinder–plates system and the total drag of the whole of the structure practically coincide and approach a value characteristic of the single cylinder.

A decrease in the length of the plates leads to an increase in the oscillation amplitude of C'_y (Fig. 2b), which points to an increase in the intensity of the vortex system. The peak amplitude is observed at a length of the plates $L = 0.2D$ and practically coincides with the value for the single cylinder. A further decrease in the length of the plates leads to a decrease in the oscillation amplitude of C'_y and in the vortex intensity.

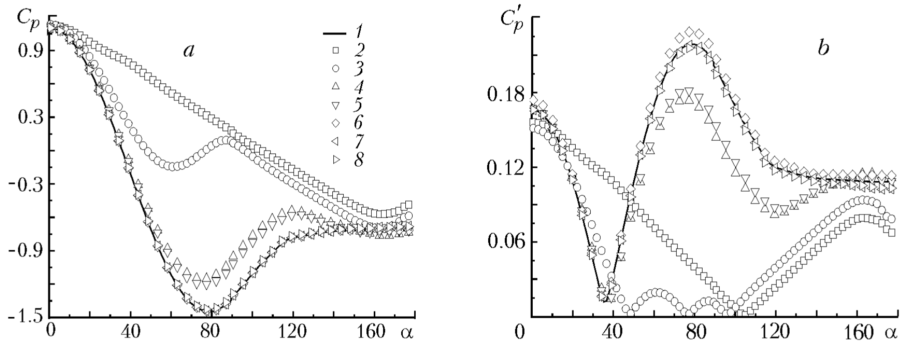

Fig. 4. Distribution of the pressure coefficient (a) and its rms oscillations (b) over the cylinder contour: 1) single cylinder; 2) L = 1.1D; 3) 0.7D; 4) 0.4D; 5) 0.3D; 6) 0.2D; 7) 0.15D; 8) 0.05D.

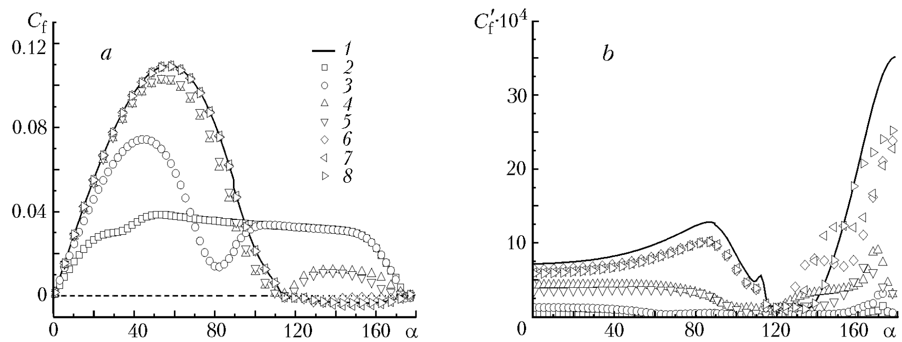

Fig. 5. Distribution of the surface friction coefficient (a) and its rms oscillations (b) over the cylinder contour. Designations 1–8 same as in Fig. 4.

At a length of the plates L = 0.2D the frequency of vortex formation (Strouhal number) is maximum (Fig. 3). Since at this length of the plates C_x is lower than for the single cylinder and the C_y' amplitude is maximum, it may be expected that the intensity of the vortex system behind the cylinder with plates of length L = 0.2D is maximum.

The setting of plates of length L = 1.1D causes a monotonic decrease in the pressure coefficient C_p throughout the cylinder perimeter (Fig. 4a). A decrease in the length of the plates to $L \le 0.4D$ leads to the formation of a detached flow near the edge of the plate, and at $L \le 0.2D$ the character of the flow around the cylinder with plates is similar to the flow around the single cylinder.

In the case of the windward flow around the cylinder, the pressure gradient is negative, the flow velocity increases, and the thickness of the boundary layer increases with receding from the frontal stagnation point at which the oscillations of the pressure coefficient C_p' are maximum. For plates with $L \le 0.7D$ a minimum of C_p' in the vicinity of the cylinder point with an angular coordinate α = 40° (Fig. 4b) is observed, and for plates with $L > 0.7D$ this minimum shifts to the windward side of the cylinder. The presence of the minimum is due to the damping of the pressure-coefficient oscillations resulting from the disturbance caused by the flow-cylinder interaction. Above the point with a coordinate $\alpha \sim 40°$ the pressure-coefficient oscillations grow due to the increase in the pressure on the cylinder surface compared to the static pressure in the flow, since $dp/dx > 0$. Note that C_p' is the higher the shorter the plate. The pressure-coefficient oscillations for the cylinder with plates of length $L < 0.2D$ are higher throughout the perimeter of the cylinder than in the case of the single cylinder.

The increase in the friction coefficient C_f to the cylinder point with an angular coordinate $\alpha \sim 60°$ (Fig. 5a) is due to the acceleration of the fluid flow because of the negative pressure gradient. In the region of maximum C_f, the pressure gradient is first equal to zero and then changes its sign. Subsequently C_f drops to zero and the boundary layer separates at the point where C_f = 0. Plates of length L = 1.1D slow down the fluid flow, and beginning with $\alpha \sim 50°$ the flow velocity monotonically decreases. With decreasing length of the plates the character of the flow around the cylinder with plates becomes identical to the behavior of the flow in the case of the single cylinder. The

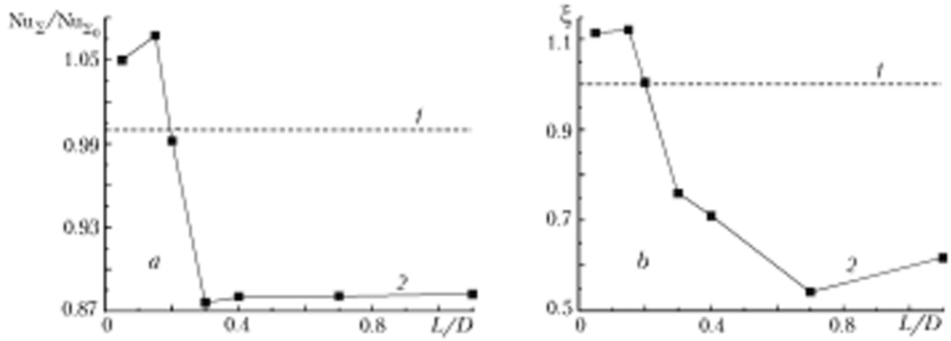

Fig. 6. Total heat transfer (a) and thermohydraulic efficiency (b) as a function the length of the plates: 1) single cylinder; 2) cylinder with plates.

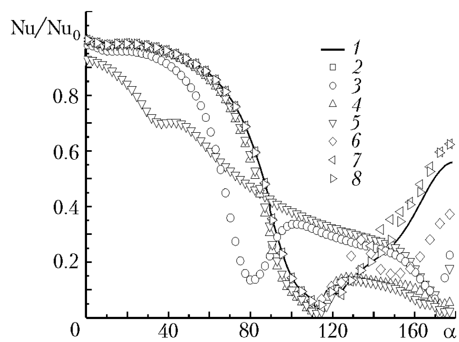

Fig. 7. Local heat-transfer distribution over the cylinder contour. Designations 1–8 same as in Fig. 4.

sloping portion of curves 2–5 (Fig. 5a) is due to the interaction of the flow with the plates. The negative values of the C_{f} distribution (curves 1, 6–8, Fig. 5a) in the region of the cylinder with an angular coordinate varying over the range of $\alpha \sim 120$–180^{o} are due to the appearance of a reverse flow.

The oscillations of the friction coefficient C_{f}' are maximum for the single cylinder (Fig. 5b), and in the vicinity of the stern critical point they are ~40% higher than the oscillations for the cylinder with plates of $L = 0.05D$, i.e., setting the plates leads to a decrease in the friction-coefficient oscillations. The first maximum in the range of $\alpha \sim 80$–90^{o} (for plates of length $L \le 0.2D$, curves 1, 6–8, Fig. 5b) corresponds to the appearance of a boundary layer of maximum length; the minimum in the range of $\alpha \sim 100$–120^{o} corresponds to the separation zone. The further growth of C_{f}' can be explained by the shedding of vortices and their interaction with the cylinder surface in the region of the stern critical point.

In the physical sense, the Prandtl number Pr is a measure of similarity of the temperature and dynamic fields. The thickness of the temperature and dynamic layers is related as $\delta_T/\delta \sim \mathrm{Pr}^{-0.5}$. Since in fluids Pr > 1, the dynamic boundary layer is thicker than the temperature one. Because in media of the type of oils the Prandtl number is much greater than unity, the dynamic boundary layer is much thicker than that of the temperature layer. For such media, the temperature boundary layer develops in the viscous sublayer.

The total heat transfer of the cylinder with plates of length $L = 0.15D$ and $0.05D$ is practically constant (Fig. 6a) and exceeds the total heat transfer of the single cylinder by ~5%. With plates of larger length the total heat transfer is lower than for the single cylinder. The thermohydraulic efficiency (Fig. 6b) is defined as $\xi = \dfrac{\mathrm{Nu}_\Sigma/\mathrm{Nu}_{\Sigma_0}}{C_x/C_{x0}}$. With plates of length $L < 0.2D$ it is ~10% higher than for the single cylinder.

The necessity of a deep insight into the processes of heat transfer between a body and the flow around it requires the determination of not only such integral thermal characteristics as the Prandtl number and the thermohydraulic efficiency. In considering the physics of the process, it is necessary to analyze the character of the change in the local heat transfer (Fig. 7). The local Nusselt number is maximum in the frontal part of both the single cylinder and the cylinder with plates. The minimum heat transfer is realized at the point of flow separation from the cylinder surface. Its further increase is determined by the interaction of the shedding vortices with the cylinder surface in the vicinity of the stern critical point. Thus, installing plates of a certain length in the stern of the cylinder makes it possible to control the flow-separation point and, consequently, the heat transfer between the cylinder and the medium flowing around it.

Conclusions. With plates of length $L = 0.2D$ the Strouhal number is maximum, which, jointly with the fact that the total drag of the cylinder with plates is ~7% lower than in the single cylinder, makes it possible to draw the conclusion that the intensity of the vortex system formed behind the cylinder is higher than in the single cylinder. The increase in the heat transfer of the cylinder with plates is due to the interaction of the shedding vortices with the cylinder surface in the region of the stern critical point and constitutes ~12% compared to the single cylinder. And the total heat transfer of the cylinder with plates exceeds the heat transfer of the single cylinder by ~5%.

This work was supported by the Belarusian and Russian Basic Research Foundations under project Nos. T08R-101, 080890001, 08-01-00059, and 08-0800065.

NOTATION

C_x, drag; C'_y, lift coefficient; c_p, isobaric heat capacity; C_p, pressure coefficient; C_f, friction coefficient; D, cylinder diameter, m; f, shedding frequency of vortices; Gr, Grashof number; g, gravitational acceleration, m^2/sec; l, clearance between the plates in the region of the stern critical point, in fractions of D; L, length of each plate, in fractions of D; Pr = ν/a, Prandtl number; Re, Reynolds number, Re = $\rho UD/\mu$; Ri, Richardson number; Sh = $L/\Delta tU$, Strouhal number; T, temperature, K; t, time in fractions of D/U; U, incoming flow velocity, m/sec; x and y, horizontal and vertical coordinates, in fractions of D; α, angular coordinate on the cylinder contour, deg; β, volumetric expansion coefficient, 1/K; δ, thickness of the dynamic boundary layer, m; δ_T, thickness of the temperature boundary layer, m; ΔT, temperature difference between the wall and the heat-transfer medium, deg; μ, dynamic viscosity coefficient, kg/(m·sec); ν, kinematic viscosity coefficient, m^2/sec; ρ, density, kg/m^3. Subscripts: 0, single cylinder; p, pressure; f, friction; oil, oil; Σ, total quantity.

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