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NUMERICAL MODELING OF TURBULENT FLOW INSIDE A WIND-DRIVEN PLANT WITH ALLOWANCE FOR THE FORCES ON THE IMPELLER

S. A. Isaev,[a] **P. A. Baranov,**[a]
A. E. Usachov,[b] **V. V. Mitrofovich,**[b]
A. D. Kolosov,[b] **and M. V. Ponomarev**[b]

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The operating efficiency of a wind-driven plant based on a confuser-diffuser accelerator is evaluated within the framework of the multiblock approach to solution of steady-state Reynolds equations closed with the use of a two-parameter dissipative turbulence model.

The concept of improving the efficiency of wind-driven plants due to the preacceleration of the air flow in front of the impeller with the use of specially profiled channels has actively been discussed in recent years. Turbulent air flow inside a specially profiled channel of a wind-driven plant has been modeled numerically to evaluate the acceptability of the concept. It is common knowledge that physical experiments involving measurement of the parameters of flow on the impeller and in the vicinity of it require considerable financial expenditures, and it is not always possible, especially at almost inaccessible places. Therefore, in solving the problem, one employs multiblock computational technologies allowing correct representation of different-scale elements of the flow in the plant [1]. A simplified mathematical model taking account of the action of forces on the impeller is proposed for correct description of the process of motion of the flow on the impeller of a wind-driven plant.

The mathematical model of turbulent, axisymmetric, low-velocity air flow about the multielement configuration of a wind-driven plant (Fig. 1a) is based on a system of complete steady-state Reynolds equations for an incompressible viscous fluid which are closed with the use of a high-Reynolds version of a two-parameter dissipative turbulence model. The standard Launder–Spalding model employed in combination with the wall-function method is modified within the framework of the Rodi–Leschziner concept so as to take account of the influence of the curvature of streamlines on the characteristics of turbulence [2]. Such an approach is tested in calculations of separated flow about bodies of different geometry, including the cases of the presence of a mobile shield, on monoblock structurized grids [3, 4].

The implicit factorized procedure for solution (by the finite-volume method) of the initial equations written in generalized form for increments in dependent variables in curvilinear nonorthogonal coordinates is based on the concept of splitting by physical processes [3, 5]. The SIMPLEC method is used for interpretation of the interrelation of the velocity and pressure fields; on centered grids, this method is supplemented with the Rhee–Chou approach with a selected empirical relaxation coefficient of 0.1. The original features of the algorithm proposed include the approximation of convective terms on the implicit side according to the upwind scheme with one-sided differences and the introduction of scheme diffusion in it for smoothing nonphysical oscillations of the solutions in the case of large Reynolds numbers. By the traditional method characteristic of methodological support of the majority of universal application packages (of the Fluent and Star CD type), one discretizes convective terms on the explicit side of the momentum equations with the use of the upwind scheme with quadratic interpolation (Leonard's one-dimensional scheme). A variant of the TVD scheme is employed for representation of convective terms in the equations for the characteristics of turbulence. The algebraic equations are solved by the method of incomplete matrix factorization.

The multiblock strategy of solution of the initial equations on intersecting structurized grids with their partial overlapping has been developed systematically from the mid-1990s primarily as applied to the aeromechanics of bodies

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[a]Academy of Civil Aviation, 38 Pilotov Str., St. Petersburg, 196210, Russia; email: isaev@SI3612.spb.edu; [b]State Scientific-Research Center at the Prof. N. E. Zhukovskii Central Aero-Hydrodynamics Institute, 17 Radio Str., Moscow, 107005, Russia. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 76, No. 6, pp. 45–48, November–December, 2003. Original article submitted April 9, 2003.

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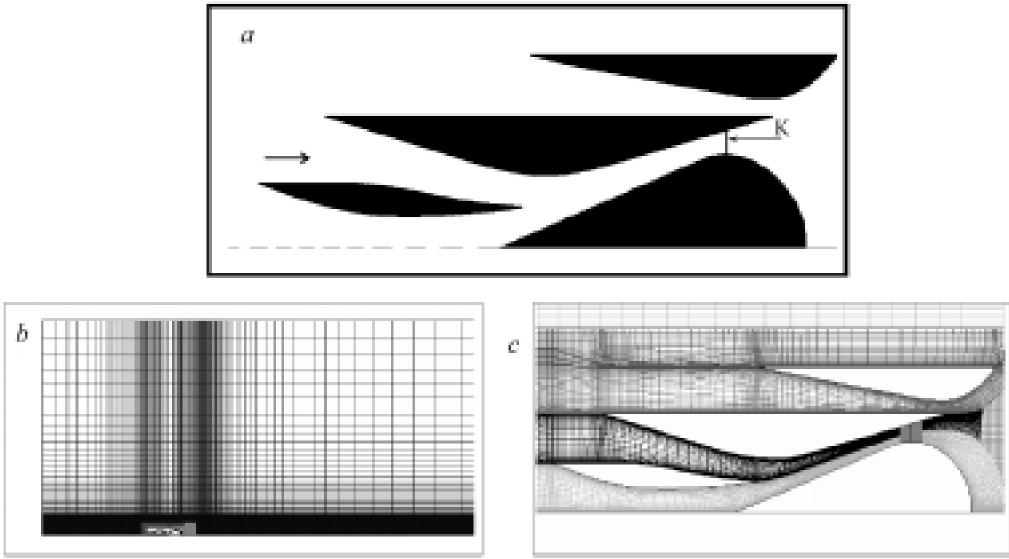

Fig. 1. Configuration of a wind-driven plant (a) with an accelerator (K is the turbine wheel) and multiblock computational grids in the computational region (b) and in the vicinity of the wind-driven plant (c). Dashed curve, axis of symmetry.

with vortex cells [1]. It was successfully tested in solving internal and external problems of hydrodynamics with flow separation [6–12] and those of vortex heat exchange in the case of flow about the reliefs of holes [13–21]. Therefore, it seems expedient to employ it for solution of the problem in question.

The configuration of an axisymmetric object, presented in Fig. 1, consists of a number of curvilinear elements. The distance from the axis of symmetry to the external shell of the last nozzle is equal to 5 m, while the length of the system of nozzles from the initial edge to the final edge is equal to 15 m.

Modeling of the turbulent flow about the multielement object is carried out on a set of multiblock grids shown in Fig. 1b. The computational region is covered with a Cartesian rectangular grid with a nonuniform step inside which there are different-scale curvilinear nonorthogonal grids. A more detailed arrangement of the block with six curvilinear grids is given in Fig. 1c. As is seen, the multiblock approach enables one to describe the geometry of the initial bodies in the best manner and to take account of the special properties of separated turbulent flow. A very fine curvilinear grid adapted to the mathematical interpretation of a windwheel is employed for modeling of the flow of the impeller.

For calculation we employ a simplified mathematical model of the impeller which enables us to transfer the basic physical conditions of flow in bladings to the "numerical experiment." In so doing, we consider a wind-driven plant with the flow going axially out of the impeller. Such conditions can be realized in the wind-driven plant consisting of the nozzle apparatus and the impeller. In the model in question, both bladings are geometrically represented by a fairly narrow disk whose thickness is one row of the cells of the narrow grid. In this work, the basic parameter characterizing the action of the impeller on the flow is the dimensionless quantity ψ:

$$\psi = \Delta\rho^* \Big/ \left(\frac{\rho V_\infty^2}{2} \right),$$

where $\rho V_\infty^2/2$ is the velocity head in an undisturbed flow.

The value of the specific work ψ is taken to be constant over the span of the impeller vane. This is consistent with the law (widely used in axial turbomachines) of profiling of bladings for circulation constant over the radius. The efficiency of the windwheel is also considered to be constant over the radius. We can show that in this case the distribution of the axial velocities in passage from the inlet to the outlet of the turbine is preserved and the radial position of the stream jet is not changed. In carrying out variant calculations, we assume that a windwheel realizing the

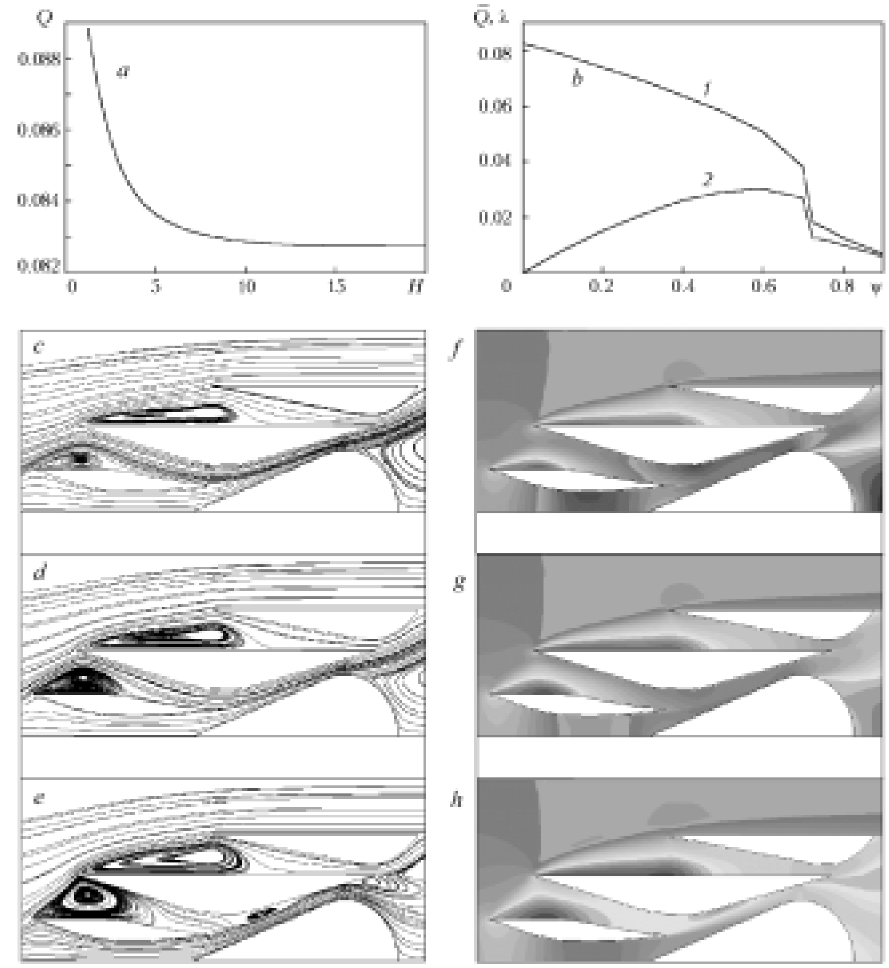

Fig. 2. Dimensionless value of the flow rate vs. distance to the upper boundary (a), dimensionless flow rate (curve 1) and coefficient λ (curve 2) vs. specific work ψ (b) and the patterns of flow about the wind-driven plant (c, d, e) and the fields of constant values of the axial velocity component (f, g, h) for three values of ψ: 0 (c, f), 0.6 (d, g), and 0.8 (e, h).

prescribed constant value of ψ for the axial-velocity distribution obtained for this variant is installed in each case in the extension of the wind-driven plant.

From consideration of the Reynolds equations (see, for example, [5]) it becomes clear that the parameter mentioned can be modeled as the external force F acting on the volume element of the gas. From this proposition, we can easily calculate the projections of F onto the coordinate axes.

Numerical modeling of axisymmetric gas flow without allowance for the swirl of the flow in the windwheel with negligibly small Mach numbers is carried out for $\mathrm{Re} = 3.123 \cdot 10^6$. The transverse linear dimension from the external edge of the nozzle to the axis of symmetry $L = 5$ m was selected as the characteristic linear dimension, and the velocity of the incoming flow $V_\infty = 10$ m/sec was taken as the characteristic velocity.

To ensure the fulfillment of the exact physical conditions of flow in the wind-driven plant we carry out test calculations with the aim of determining the rational position of the upper boundary of the computational region so as to eliminate the possible influence of the boundary conditions on calculation results. The absence of the influence of the dimensions of the computational region on the flow rate in the cross section of the windwheel bladings is selected as the condition of sufficiency of these dimensions. As is clear from Fig. 2a, the value of the flow rate is independent of the distance to the upper boundary beginning from a value of the distance equal to ten characteristic linear dimensions (10L). Therefore, the width of the computational region is selected to be 18 characteristic linear dimensions (18L).

Numerical modeling of flow inside the channel of the wind-driven plant is carried out for different values of the specific work ψ removed from the flow; the range of these values varies from 0 to 0.9. Figure 2 shows some of the results obtained on the diagnostics of the flow in the wind-driven plant.

In the absence of a turbine ($\psi = 0$), we observe a significant growth in the dimensionless value of the horizontal velocity component u in the vicinity of the turbine (to a value of 1.6). As the values of the specific work ψ removed from the flow increase, the pattern of flow inside the turbine channel begins to rapidly change. The value of the horizontal velocity component in the vicinity of the turbine rapidly drops. Thus, for $\psi = 0.6$ the dimensionless value of u in the vicinity of the turbine is less than unity. This fact indicates a rapid decrease in the rate of flow through the turbine cross section and hence a drop in the value of the power removed.

Further growth in ψ results in the occurrence and development of separation zones directly in the channel in front of the tube, which indicates insufficient operation of the wind-driven plant. As follows from Fig. 2e, a large recirculation zone keeping the air from moving is formed directly in the channel in front of the turbine. When $\psi = 0.9$, the regime of choking of the flow through the wind-driven plant is realized.

Figure 2b shows the dependence of the dimensionless flow rate of air and the dimensionless power removed from the wind turbine $\lambda = \overline{Q}\psi$ on the specific work ψ for an aerodynamic efficiency of the turbine of $\eta = 1$. The quantity λ is equal to half the factor of utilization of the wind energy adopted in wind-power engineering and represents the ratio of the capacity of the wind turbine N to the available power (rate) of the flow $\lambda = N/(\rho V_\infty^3 \pi L^2)$. We can see that the wind-driven plant considered has a maximum value of $\lambda = 0.03$. For the dimensional parameters employed in this work this corresponds to a capacity of 2.87 kW. With allowance for the fact that the aerodynamic efficiency of the turbine cannot be higher than 0.9–0.92 the shaft power will not exceed 2.6 kW, i.e., the wind-driven plant considered multiply ranks below the axial analogs without extensions in this parameter.

It should be noted that the rate of flow through the turbine of the wind-driven plant in question must be nearly an order of magnitude higher to attain an energy efficiency at least comparable to the efficiency of axial-type wind-driven plants without extensions.

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NOTATION

ρ and V_∞, density and velocity of the incoming flow; L, characteristic dimension; Δp^*, difference of the total pressure on the turbine; ψ, dimensionless value of the specific work removed from the flow; λ, ratio of the capacity of the wind turbine to the available rate of the flow; Re, Reynolds number; F, external force acting on the volume element of the gas; η, aerodynamic efficiency of the turbine; Q and $\overline{Q}$, flow rate and its dimensionless value; H, distance to the upper boundary of the computational region.

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