Influence of the Design Mode of the Turbine Design on the Gas Dynamics of the Flow in its Flow Part

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Abstract
On the basis of the optimal design method, the radial-axial turbine impellers are designed for various design pressures of the gas at the inlet. The design results showed that with an increase in the design gas pressure at the turbine inlet, the height of the impeller blades decreases. For the first time for a radial-axial turbine, the effect of the calculated gas pressure at the turbine inlet on the flow structure in its impeller is shown. As a result of the conducted studies, it was found that with a decrease in the calculated gas pressure at the turbine inlet, the area of flow separation in the impeller increases. For the experimental determination of the separation area, the braking pressure and the static pressure were measured and compared behind the impeller. The final judgment about the appearance of the separation area was registered when the braking pressure and the static pressure were equal.

Keywords 1
Height of the impeller blade, radial-axial turbine, speed profile, flow part, design mode, flow separation, current line.

1. Introduction

Radial-axial turbines are widely used in power plants for various purposes. These can be low-power gas turbine engines and microturbines, which are highly efficient autonomous sources of electrical and thermal energy; auxiliary power plants of aircraft; turbo-expander installations, as well as turbochargers used as supercharging units for reciprocating internal combustion engines. In turbochargers of supercharging units, the turbines operate in a non-stationary exhaust gas flow, which leads to large reserves in terms of improving their flow parts. The purpose of this work is to use a model of a two-dimensional flow of inviscid compressible gas in the flow part of a radial-axial turbine to investigate the effect of the design mode of the turbine design on the gas dynamics of the flow in its flow part. The relevance of the work is due to the need to improve radial-axial turbines in connection with the continuous tightening of requirements for marine power plants and internal combustion engines in terms of improving their environmental performance and saving fuel and energy resources.

2. A mathematical model for research

The vortex flow of an inviscid compressible fluid in radial-axial turbomachines is described by the following system of equations [1]:

the equation of motion in the Crocco energy form

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$$\boldsymbol{w} \times (\nabla \times c) = \nabla H^* - T\nabla S - \boldsymbol{F} + T\frac{\boldsymbol{w}}{w^2}\frac{dS}{dt}; \tag{1}$$

continuity equation

$$\nabla \cdot (\chi\rho\boldsymbol{w}) = 0; \tag{2}$$

equation of the first law of thermodynamics

$$\frac{dU}{dt} + p\frac{d\rho^{-1}}{dt} = T\frac{dS}{dt}; \tag{3}$$

gas equation of state

$$p = \rho RT, \tag{4}$$

where $= c_v dT$; $H^* = J + \frac{w^2-u^2}{2}$; $J = c_p T = \frac{kR}{k-1}T$; $\left(k = \frac{c_p}{c_v}\right)$.

The following notation is used in equations (1)-(4): ∇H^* - gradient of the field generalized heat content in the relative flow; J - enthalpy; ∇S - entropy gradient; T - absolute temperature; F - mass power; U - internal energy; k - indicator of isentropy; c_p and c_v - heat capacity at constant pressure and volume; p - pressure; ρ - density; R - gas constant; w – relative velocity; c - absolute velocity; u - circumferential velocity; χ - constraint coefficient; r - distance from the axis of rotation.

In this case, the steady-state vortex flow is considered. Moreover, the vorticity is caused by the presence of blades in the flow. The entropy varies across and along the current lines. The liquid is compressible and perfect. The main effect of viscosity is taken into account by the isentropy coefficient $\sigma = exp[(S_1 - S)/R]$, which is considered a function of the flow parameters and coordinates. The hypothesis of axial symmetry of the flow is accepted.

The solution of the problem under consideration is constructed on a semi-fixed grid by the method of straight lines. The change in the isentropy coefficient is given by the quadratic dependence. Partial derivatives are replaced by central differences. The equations of motion and continuity are represented by integral equations. The calculation is carried out by the method of successive approximations. The degree of approximation is estimated with an accuracy of 1%.

3. Design of the flow part

For the design of the flow part, the following parameters are set: the flow rate of the gas passing through the flow part; the speed of the turbine; the pressure of the torsion in front of the turbine. The number of revolutions is determined when calculating the optimal turbine parameters [2, 3]. The accepted parameters of the four design modes are given in the table.

Table
Parameters of the calculated modes

Parameters	Mode 1	Mode 2	Mode 3	Mode 4
Gas consumption Gt, kg / s	0,45	0,45	0,45	0,45
Stalled flow pressure at the stage inlet p_0^*, MPa	0,18	0,20	0,22	0,24
Degree of pressure reduction $\pi_т = p_0^*/p_2$	1,7	1,89	2	2,26
Speed of rotation nтк, min-1	35860	39880	43660	46860

Using the optimal design method [2, 3], we perform the design calculation of the flow parts for the previously defined design modes (Table). The design results are presented in (Fig. 1). Also in Fig. 1, for comparison, the flow part of the impeller of the standard turbine of the TKR-14C-27 turbocharger is shown.

In fig. 1 turbine № 1 is designed for mode № 1, turbine № 2 for mode № 2, turbine № 3 for mode № 3, and turbine № 4 for mode № 4.We will refer to them as experimental turbines.

As shown by the design results, with an increase in the design gas pressure at the turbine inlet, the height of the blades at the inlet and outlet of the impeller decreases (Fig. 1), and the optimal number of turbine revolutions increases, determined using the optimal design method (Table).

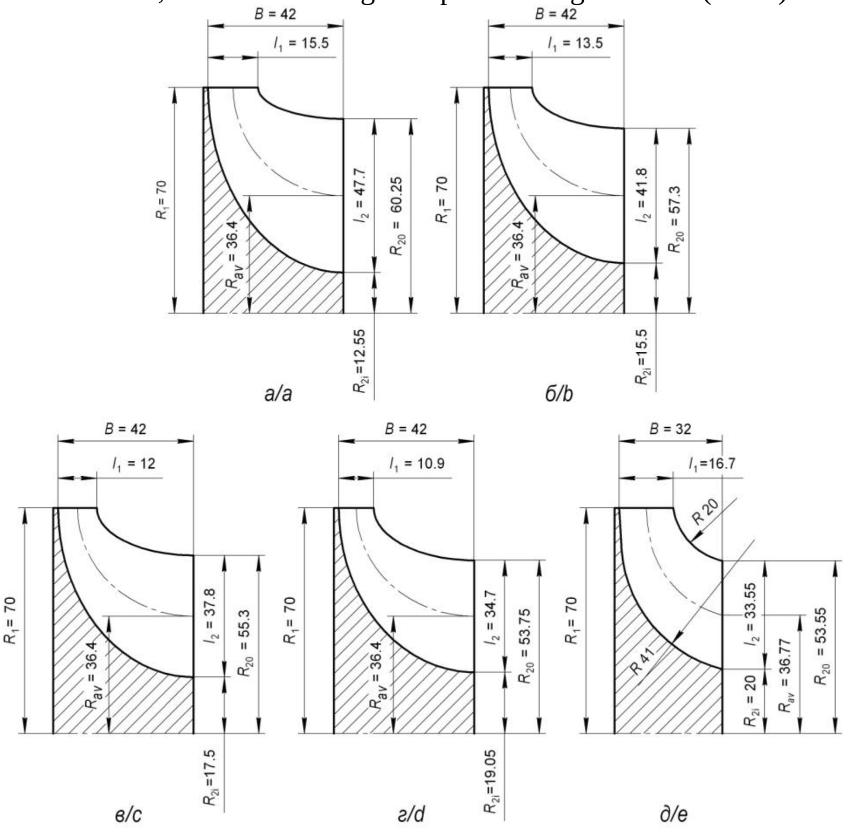


Figure 1: Impellers of a radial-axial turbine: a) turbine № 1, mode 1; b) turbine № 2, mode 2; c) turbine № 3, mode 3; d) turbine № 4, mode 4; e) standard turbine TKR-14C-27

4. Calculation results

To perform a two-dimensional calculation of the gas flow in the flow part of a radial-axial turbine, a program in the MATLAB programming language is compiled [4]. To perform the calculation, the flow part is divided into a grid. To do this, we draw a family of straight lines l to the outer line of the meridional contour (Fig. 2). After that, we apply the current lines s according to the principle of equal annular areas (Fig. 2 shows thin lines). To ensure the convergence of the approximations, it is essential that l differs little from the normal to the current line. The fulfillment of this condition together with the requirement $\Delta l \leq \Delta s$ practically ensures the convergence of successive approximations.

Figure 2 shows the results of modeling the dynamics of a two-dimensional flow of inviscid compressible gas carried out in the impellers of radial-axial turbines designed for different design modes. As shown in Figure 2, the current surfaces of the last approximation (thick lines) are significantly different from the current lines of the preliminary channel partitioning (thin lines).

In total, for the required convergence (2% in speed), it was necessary to perform from 12 to 15 approximations. The time taken by the machine for one approximation was 0,016 s.

Figure 2 shows that with increasing curvature of the lines of the meridional contour of the impeller, the current lines approach the turbine body. This is mainly expressed in turbine № 1 in the zone of straight lines $l5$, $l6$ and $l7$. This is less pronounced in turbines № 2 and 3 in the zone of straight lines $l6$ and $l7$. In the area of these lines, negative velocities appear on the inner contour of the meridional profile. This circumstance gives reason to assume that a flow separation (return currents) is formed in the turbine impellers. In Figure 2, the area of return flows is shaded. The boundary of the region of return flows is determined under the condition that the meridional velocity projection is equal to zero. With an increase in the design gas pressure at the turbine inlet, the current lines are somewhat aligned. For example, in the flow section № 4, the current lines are more even than in the flow sections № 1,2 and 3.

The results of the studies indicate that an increase in the height of the impeller blades at the inlet and outlet leads to an increase in the efficiency of the turbine, a drop in the effective power of the turbine, as well as an increase in the area of return flows in the flow part of the turbine (Fig. 2).

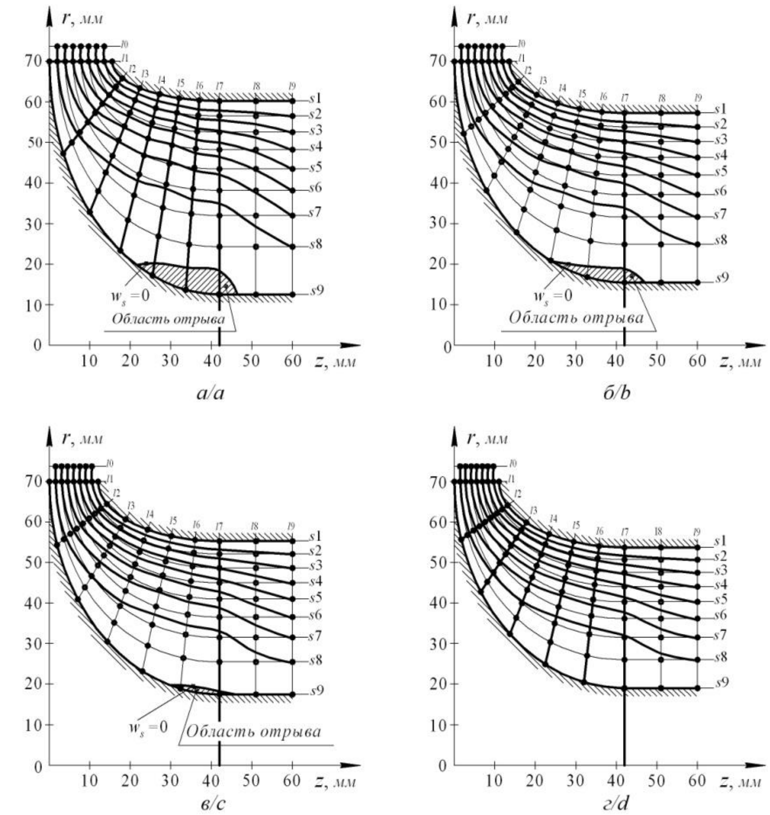


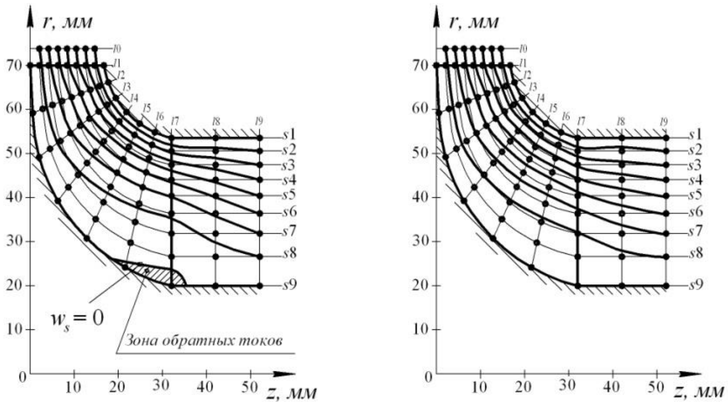
Figure 2: Results of calculating the flow in the flow parts of turbines: a) turbine № 1- mode 1; b) turbine № 2 – mode 2; c) turbine № 3 – mode 3; d) turbine № 4-mode 4

5. Experimental verification

For experimental verification of the mathematical model of a two-dimensional vortex flow of inviscid compressible gas in the flow part of a radial-axial turbine, the turbine of the TKR-14C-27 turbocharger was purged on a special stand. The stand allows the measured parameters to determine the efficiency of the turbine, as well as to measure the velocity field at the outlet of the flow part of the turbine. The description of the stand is given in [5-7]. To measure the flow parameters behind the turbine, in the area of the gas outlet from the impeller, a ball probe with a coordinate device is installed on the stand. In addition to measuring the total pressure, the probe allows you to measure the angle of the flow exit from the turbine.

To check the adequacy, the gas dynamics of the two-dimensional flow in the flow part of the standard turbine TKR-14C-27 was calculated. The calculation results are shown in Fig. 3.

Experimental check on the separation flow mode. The beginning of the flow separation area along the radius is located near the bushing area. To determine the area of separation behind the impeller, the braking pressure and the static pressure were measured and compared. The difficulty in this case was to determine the angle of the flow exit. Near the area of the flow separation, turning the measuring device in the range from -10 ° to +10° did not lead to a change in the braking pressure. As a result, the final judgment on the appearance of the separation area was recorded when the braking pressure and the static pressure were equal. The experimentally measured pressures along the height of the outlet section of the flow section are shown in Fig. 4. As shown in Figure 4, a flow separation appears in the root zone. The length of the flow separation zone along the output section of the impeller was set when the braking pressure was equal to the static pressure [8].



a b

Figure 3: Results of the calculation of the flow in the flow part of the standard turbine: a) the separation flow mode; b) the maximum efficiency mode

A comparison of the calculated (Fig. 3a) and experimental data (Fig. 4) on the flow separation mode allows us to conclude that the assumption that there is a flow separation in the turbine has been experimentally confirmed.

The results of this experiment can be extended to the experimental turbines № 1, 2 and 3, in which a reverse current zone occurs, since these turbines and the standard turbine belong to the same type size.

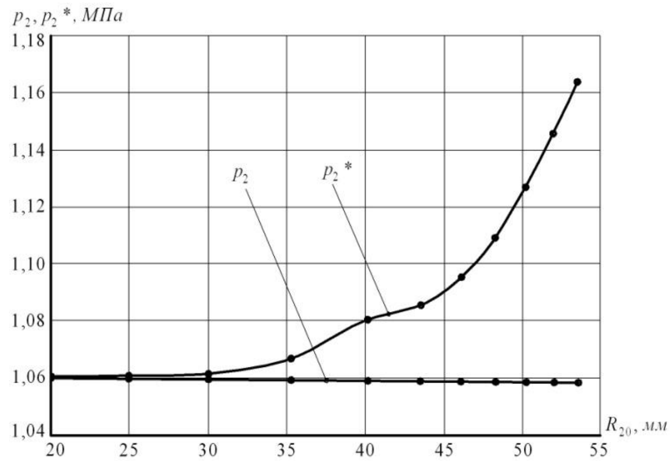


Figure 4: Change in the braking pressure and static pressure along the height of the output section

6. Conclusions

1. A decrease in the design pressure in front of the turbine leads to an increase in the height of the impeller blade at the inlet and outlet, an increase in the efficiency of the turbine, a drop in the effective power, and an increase in the backflow zone in the flow part of the turbine. Based on this, when designing a radial-axial turbine, we recommend taking the value of the total gas pressure at the turbine inlet $p_0^* = 0{,}24$ МПа. This will avoid the area of return currents.

2. With an increase in the design gas pressure at the turbine inlet, the current lines are somewhat aligned. For example, in the impeller of turbine № 4, the current lines are smoother than in the impellers of turbines № 1, 2 and 3.

3. A comparison of the calculated and experimental data on the flow separation mode allows us to conclude that the assumption that there is a flow separation in the turbine has been experimentally confirmed.

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