Computational and experimental research of a steam turbine system for an intra-cycle compression of fuel

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Abstract. The paper deals with the design of highly efficient small-size turbines. A method for designing a steam gas turbine using numerical methods for calculating the working processes in the flow section is described. The results of the calculated and theoretical research, which allowed determining the main characteristics of the turbine, are given. The estimated efficiency of the resulting turbine was not less than 85%. The results of tests in the model conditions of the model turbine, manufactured using additive technology, are presented. A comparison of experimental and calculated data has led to the conclusion about the adequacy of the developed mathematical model used in the calculated and theoretical research.

1. Introduction
Modern energy is characterized by large-scale introduction of combined-cycle gas turbines (CCGT). Until recently, the most common way to increase the efficiency of such plants was to increase the temperature and pressure of the working fluid before the turbine of a gas turbine unit (GTU) [1,2]. However, a further increase in these parameters is associated with a significant increase in costs (including energy) of auxiliary equipment. In this regard, authors of [3] proposed a variant to improve the method of supplying fuel gas to the combustion chamber (CC) of the GTU, which is an intra-cycle compression of fuel system (ICCF). One of the elements of this setup is a steam turbine, the parameters of which significantly affect the efficiency of the ICCF and, accordingly, the efficiency of the CCGT, taking into account the proper energy consumption.

As an example, in this paper, the setup GTU-2.5P, which has a small capacity (2.7 MW), was considered. For its operation in the nominal mode, it is necessary to supply the fuel gas (methane) to the CS with a pressure of 0.8 MPa and a flow rate of 0.246 kg/s with the help of a compressor-regulator. The power source for driving such a compressor can be a steam turbine with a rated power of 60 kW.

The purpose of this work is to design a steam turbine for the setup of the ICCF, with an efficiency of at least 85%. It is necessary to solve the following tasks: to design the nozzle and rotating wheel of the steam turbine; using the mathematical model developed by the authors, to estimate the parameters of the developed turbine and make the necessary improvements based on the results of a three-dimensional calculation; to experimentally confirm the reliability of the characteristics of the model turbine obtained at the stage of computational and theoretical studies.

2. Turbine design technique
The determination of the geometric parameters of the turbine elements was carried out in several stages. At the first stage, a primary calculation of the characteristics of the nozzle, specifying its degree of
reactivity and the number of blades, was made [4]. The geometrical angle of the flow from the nozzle, as well as the flow area of the flow part corresponding to the specified flow rate was obtained. One of the main results of the calculation of the nozzle parameters is the static pressure at the outlet, i.e. the “inter-stage” pressure (between the nozzle and the turbine). On the basis of the existing prototypes, taking into account the obtained geometric characteristics, the nozzle was profiled [5]. Next, a numerical simulation of the processes occurring in the flow part of the designed nozzle was carried out and the average flow angle at the outlet was obtained.

In the second stage, the turbine blade was profiled. Profiling of the turbine blade leading edge was carried out taking into account the velocity triangle at the entrance corresponding to the selected nominal speed of rotation of the turbine wheel 87000 rpm. The geometric angle of entry should correspond to the unaccented hit of the gas flow on the blade. The geometrical exit angle was chosen taking into account the maximum rotation of the flow in the direction of the radial exit from the rotating wheel, at the same time with the minimum separation at the trailing edge of the blade. Rotation of the flow inside the wheel is limited. To stabilize the flow between the blades, the cross section of the flow section in the meridional section was cut. As a result of a number of measures to change the geometric characteristics of the blade and the flow part, taking into account the limitations on the strength properties and manufacturing technology, as well as a consistent series of calculations based on numerical methods of computational fluid dynamics, the blade profile was developed and a turbine wheel was developed.

In the third stage, a joint calculation of the turbine stage was carried out, after which the entry and exit angles, the number of blades of the turbine wheel were adjusted, since it was necessary to minimize the speed at the exit of the turbine in the circumferential direction. The adjusted flow section at the exit from the turbine ensured better flow and rotation of the flow in the axial direction. To obtain the most efficient parameters of the turbine stage, the geometrical models of the blades and the flow part were adjusted using the following changes: reducing the blade profile thickness, changing the number of blades, changing the spacing of the flow section, changing the spacing between the blades, cutting the blades at the exit, changing the gap between the nozzle and the turbine wheel. As a result of a set of design measures and the subsequent numerical study of gas flow in the flow part of the turbine stage, its calculated efficiency was more than 87%.

3. CFD analysis

The calculated and theoretical study of the gas flow in the working channel of the turbine stage was carried out using computational complexes of hydro-gas dynamics, based on the use of numerical methods [6-8]. The calculated area of the object of study (the flow part of the turbine stage) is presented in Figure 1a and has several subdomains: a segment of the nozzle 1, a turbine wheel 2 and a segment of the output section 3. The selected subregions are closed in the circumferential direction. The number of nozzle and turbine segments with a segment of the output section corresponds to the number of their blades (11 and 17, respectively). The basis of the mathematical model describing the flow of a gas flow in the considered area is the basic equations: the Navier-Stokes system of equations, the continuity and energy equation, supplemented by the ideal gas state equation and the SST-Menter turbulence model equations. The boundary conditions are set: on the surface S₁, the mass flow rate G and the total temperature of the flow \( T_{0}^* \); on the surface S₂ - the averaged static pressure \( P_{2} \); The surfaces S₃ and S₄ are adiabatic walls, the surfaces S₅ are closed with opposite corresponding wall segments. For elements in region 2, the peripheral velocities corresponding to the turbine rotation frequency \( n \) are specified. Each subdomain is transformed into a calculated “grid” based on hexagonal finite elements with a decrease in the thickness of elements in the boundary areas of the blades and walls. The number of elements for regions 1, 2, and 3 was 240, 980, and 625 thousand cells, respectively. The density and quality of the calculated “grid” is presented in Figure 1b. The presence of the gap between the turbine wheel and the housing was not taken into account in the calculation.
Figure 1. The calculated area of the turbine stage (a) and the “grid” of finite elements (b)
1 – nozzle area, 2 – turbine area, 3 – output area

The compiled mathematical model is used to obtain the gas-dynamic parameters of the flow part for two cases: physical - with real parameters of the working medium flowing in the designed units, and model - with parameters brought to the conditions achieved during the experiment. In the first case, the working medium is superheated steam, in the second - air. The criteria for the convergence of the computational model were: the equality of flow at the S₁ and S₂ boundaries within the computational error, the stabilization of the gas temperature at the S₂ boundary, and the average static pressure at the S₁ boundary. The boundary conditions and the data obtained after the calculation are presented in Table 1, where p₀* is the total gas pressure on the surface S₁; p₁* is the total gas pressure on the surface of the junction of regions 1 and 2; π is the degree of expansion in the turbine, η is the efficiency of the turbine stage, taking into account losses in region 3, N is the turbine power.

Table 1. Boundary conditions and calculated data of the mathematical model.

| Case       | Boundary, initial conditions | Calculated data |
|------------|-----------------------------|-----------------|
|            | G, g/s                      | T₀*, K          | p₂*, MPa | n, RPM | p₀*, MPa | p₁*, MPa | η, %  | π    | N, kW |
| Physical   | 394                         | 623             | 0.95     | 87000  | 2.022   | 1.962   | 88.3  | 1.96 | 63.7  |
| Model      | 27                          | 320             | 0.057    | 47300  | 0.104   | 0.101   | 86.1  | 1.88 | 2.1   |

Further, the analysis of the obtained data and comparison of the flow patterns of the flow section of the turbine in the middle section for two cases were carried out. Figures 2 and 3 show the distribution of Mach numbers and velocities in the middle section of the flow part of the turbine stage.
The results obtained indicate that the correspondence of Mach and Reynolds numbers and the chosen ratio of gas temperatures at the entrance to the turbine stage lead to similar results in the flow in physical and model cases and, accordingly, to similar efficiency values. This allows in practice to implement the testing of blade wheels at lower speeds, other working environments and temperature conditions.

### 4. Experimental research

To confirm the results of the theoretical calculations, an experiment was conducted using a physical-gas-dynamics experiment stand (PGDES) [9]. The stand (Fig. 4) operates in a closed loop in which a centrifugal vacuum compressor 1 is used to organize a directional air flow. The turbine is located in the high-pressure compartment 2, the turbine wheel is mounted on the transmission shaft. To achieve the required rotational speed of the turbine wheel in the medium pressure compartment 3, the rotating wheel of the technological compressor is fixed on the transmission shaft. A schematic diagram of the stand is shown in Figure 5. The vacuum compressor (VC) provides vacuum gas behind the turbine model. Under the action of pressure drop in the turbine stage, air flow is organized, which causes it to rotate. To achieve the required mass flow rate through the turbine and to protect the vacuum compressor from surging in the stand, a bypass line is arranged, allowing smoothly to change the amount of gas transferred from the discharge to the suction of the vacuum compressor. The stand allows for the measurement of air pressure and temperature, as well as the frequency of rotation of the transmission shaft. Maintaining...
the required temperature at the entrance to the model turbine is carried out by regulating the flow of water through the air cooler.

![Appearance of the PGDES.](image)

**Figure 4.** Appearance of the PGDES.

![Schematic diagram of the PGDES](image)

**Figure 5.** Schematic diagram of the PGDES

The tests were carried out as follows: in the initial position, the total pressure and temperature values were measured in sections before and after the turbine model, before the narrowing device, before the vacuum compressor, and the difference between static and total pressure in the narrowing device. To increase the accuracy of temperature measurements in each section, 3 thermocouples were installed. It was confirmed that all sensors showed the same value. With the valve fully open, installed on the bypass line, the vacuum compressor was turned on at 9000 rpm. Gradually closing the valve, the operation of the vacuum compressor corresponding to the nominal one (according to the well-known characteristic) was achieved. After that, the stationary mode of operation of the turbine model was achieved at which the values of the measured parameters did not change for 5 minutes. Further, the rotational speed of the rotating wheel of the vacuum compressor was consistently increased to 35000 rpm.
As a result of the experiment, the turbine wheel speeds, total turbine input and output air pressure and temperature, as well as the total and static air pressure and temperature in constricting nozzle were determined. The processing of the obtained data was carried out as a result of which the sample of values was determined by the stationary modes of the turbine model. Values were removed from each sample, the deviations of which from the mathematical expectation exceeded twice the value of the standard deviation. Further, the parameters of the air flow in the narrowing device were recalculated with the mass flow rate ($G_a$). The degree of expansion in the turbine was also determined. ($\pi_T = \frac{P_2^*}{P_3^*}$) and adiabatic efficiency ($\eta = \frac{(T_1^a - T_2^a)}{[T_1^a \cdot (1 - 1 / \pi_T^{(k-1)/k})]}$). After processing the obtained results, the dependences of the air flow through the turbine and adiabatic efficiency on the degree of expansion were constructed (Fig. 6-7). It should be noted that the maximum error in determining the parameters of the turbine corresponds to small values $\pi_T$. Wherein the relative error when working with $\pi_T$ higher than 1,4 does not exceed 10%.

![Figure 6](image.png)

**Figure 6.** Dependence of mass air flow $G_a$ on the degree of expansion $\pi_T$ (left) Dependence of adiabatic efficiency $\eta$ on the degree of expansion $\pi_T$ (right)

Comparison of the results of theoretical and experimental studies show a good match for the model parameters. These results confirm the reliability of the results obtained using mathematical modeling.

**Conclusion**

As a result of the work, a steam turbine was designed for an intra-cycle fuel compression system. With the help of the developed mathematical model, its parameters were determined under the nominal operation mode in the setup. When calculating the process of air flow in a designed turbine with identical model conditions within the boundary conditions, a change in the flow conditions in the turbine stage was revealed. This is caused by the impossibility of achieving unchanged Reynolds numbers when changing the working fluid. However, the good agreement between the results of the calculated and theoretical research of the process of air flow in the turbine with the experimental data allows us to conclude that the developed mathematical model is adequate. It should also be noted that tests carried out under model conditions allow us to make conclusions about turbine operability and efficiency and could be used to construct and improve turbines.

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