5.2 Operation of the last stage of the cogeneration turbine in the mode of heat dissipation

Cogeneration turbines of high power (from 100 MW and more) are made with two heat extractions and condenser, which ensures its operation in the condensing mode with increased electrical power [71, 72]. As a rule, two network water heaters provide its heating in the range from 50 to 110°C when the steam flow rate changes from 1000 to 7500 ÷ 8000 m^3/h. In the designs of cogeneration turbines, a low-pressure cylinder (LPC) consists of 2 stages (turbine T-100/120-130) and 3 stages (turbines T-180/220-12.8, T-185/230-12.8, T-250/300-240, T-260/300-240). LPCs are performed according to a 2-line scheme [71]. Each flow is equipped with a control stage with a rotary diaphragm. Full opening of the swivel ring and shutdown of the network water heaters ensures the condensing mode of the turbine.

Cogeneration modes are created when the network heaters are on the steam condensation pressure in which depends on the network water flow rate and its temperature at the outlet of the top extraction heater and is provided by covering the swivel ring of the control stage diaphragm from full to zero position, corresponding to full closure. The minimum steam flow passes through the stages in LPC flow at the zero position of the swivel ring, which is necessary to remove heat from the ventilation losses of stages operating at different steam pressures in the condenser and a minimum steam mass flow rate of 2 ÷ 5% of the flow rate at the nominal operating mode of the last stage from which the steam is directed to the condenser.

The operation of LPC at different levels of pressure in the condenser leads to a change in the specific volume of steam v. Therefore, at a constant flow rate G through the last stage for different operating modes, the characteristic of the low-flow rate mode will be the volume flow rate of steam through the last stage $V = Gv$ [73]. In this case, it is advisable to represent the operation of stages depending on the relative volume flow rate of steam as

$$\overline{Gv_2} = Gv_2/(Gv_2)_{\mathrm{nom}}, \quad (1)$$

where $(Gv_2)_{\mathrm{nom}}$ is the volume flow rate of steam at the outlet of the rotor wheel at the nominal mode.

An analysis of the last stage operation of cogeneration turbines showed that it operates in a wide range of changing modes, corresponding to both the generation of mechanical energy (turbine mode) and its consumption (low-flow rate mode).

If considerable attention is given to the study of the operation of large fanning stages in nominal and close to it modes when creating high-power turbines [71, 73 – 75], then insufficient attention is currently given to certain issues of their operation in the region of modes below idle.

The special attention is given to the determination of power losses due to friction and ventilation, a method and algorithm for calculating the temperature state of the flow path of a steam turbine are given in [76]. It is noted that when modeling the flow path of LPC at the low-flow rate modes, it is necessary to consider the hydrodynamic processes occurring in the turbine stages. Non-stationary aerodynamic characteristics in a low-pressure steam turbine were numerically studied under off-design operating conditions using commercial software ANSYS-CFX [77]. This paper discusses the issues of changing the torque on the steam turbine blade, the distribution of Mach number, the determination of the flow line and the distribution of static pressure over the surface of the blade at different heights. The calculation results indicate the formation of vortices in the interblade channel.

An assessment of modern approaches to describing the flow and energy characteristics of high-power turbine stages [78] makes it possible to integrate them into computer simulation of the calculation of processes occurring in high-power turbines for variable modes, including low-flow rate modes.

The most complete description of the operating parameters for the low-flow rate modes of T-250/300-240 cogeneration turbine based on the results of experimental data is given in the monograph by A. V. Khaimov [79].

However, not enough attention has been given to the issues of the flow structure of the working medium in the stages with a large fanning. This may be one of the reasons why the calculated data obtained during mathematical modeling of the flow in the large fanning stage are contradictory by various authors.

Knowledge of the flow formation conditions in the large fanning stage in the last stage of LPC will make it possible to evaluate the main energy characteristics of the stage at the low-flow rate modes.

At modes close to the nominal flow in the stage is close to cylindrical and the conditions for energy transfer from the flow to the sections of the working blades for such modes are considered in detail based on Zhukovsky's theorem on the circulation of velocity around the profile of the elementary blade of the stage [73], when the projection of the resultant force R_u on the circumferential direction

$$R_u = \rho w_z \Gamma, \quad (2)$$

where ρ is the working environment density; w_z is the axial (consumable) component of the oncoming flow velocity; Γ is the velocity circulation around the profile of the rotor blade element, equal to

$$\Gamma = \oint w_s ds = w_{1u} d_1 - w_{2u} d_2, \quad (3)$$

where w_{1u} , w_{2u} are projections of the relative flow velocity onto the circumferential direction in front of and behind the profile blade, respectively; d_1 and d_2 are the step of the blade profile at the entrance and exit from it (Fig. 1).

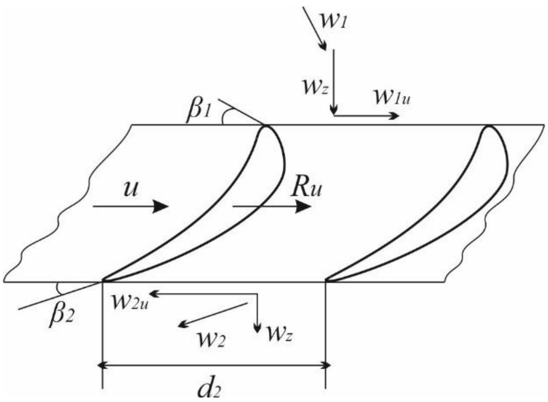

Figure 1. Determination of the flow circulation in the blade profiles

As follows from (2) and (3), the energy transfer, depending on the ratios w_{2u}/w_{1u} and d_2/d_1 occurs both from the flow to the rotor blade elements ($+R_u$) and from the blade elements to the flow ($-R_u$).

It follows from the analysis of the flow structure [80] that, depending on the relative volumetric flow rate $\overline{Gv_2}$, these ratios change and the rotor wheel both receives energy from the flow and gives it away flow, receiving energy from an external source [81]. Since the stage under consideration has a large fanning, then, according to equation (3), the transition from transferring energy to the working blades from the flow and vice versa occurs along elementary streams, starting from the peripheral sections of the blade, ending in the root ones, i.e., the idle for each stream (Γ = 0) varies along the length of the blade and the idle mode of the stage is an integral value determined by the summation of the consumed and produced energy along the length of the working blade.

Taking into account the changes in the flow structure and the impossibility of determining the energy losses in the stream filaments, in order to determine its characteristics at the low-flow rate modes, it is advisable to use the results of an experimental study of models of large fanning stages, processing them according to the approach based on the use of the Lagrange method [73], dividing the flow into its components and each of them into elementary streams.

The power ΔN developed by the element of the rotor wheel of the stage, referred to a layer of small thickness ΔG, can be represented as

$$\Delta N = \Delta G \cdot h_u, \tag{4}$$

according to the Euler equation, the specific work h_u of the selected stream of the working medium is for the streams passing through the rotor wheel [71]

$$h_u = u_1 \cdot C_{1u} + u_2 \cdot C_{2u}, \tag{5}$$

where u_1, u_2 are circumferential velocities at the inlet "1" to the rotor wheel and the outlet "2" of the elementary stream from it; C_{1u} , C_{2u} are circumferential components of the stream velocity at the rotor wheel inlet and outlet (the + sign corresponds to the circumferential velocity direction).

To determine the performance characteristics of the stage, because the actual measured values include the power costs for moving the working medium into the elementary stream and the energy losses in it, when processing the results of experiments, one can use the generalized Bernoulli equation written for section 1 at the inlet of the stream into the rotor wheel and for section 2 at the exit from it in the form [71]

$$\frac{P_1^*}{\rho^*} = \frac{P_2^*}{\rho^*} + h_u + h_r, \tag{6}$$

where h_u is the specific work (5) spent on moving the stream from radius r_1 in the section 1 to radius r_2 in the section 2; h_r is the irreversible (thermal) energy losses in the stream when moving from the section 1 at the rotor wheel inlet to the section 2 at its outlet (Fig. 2).

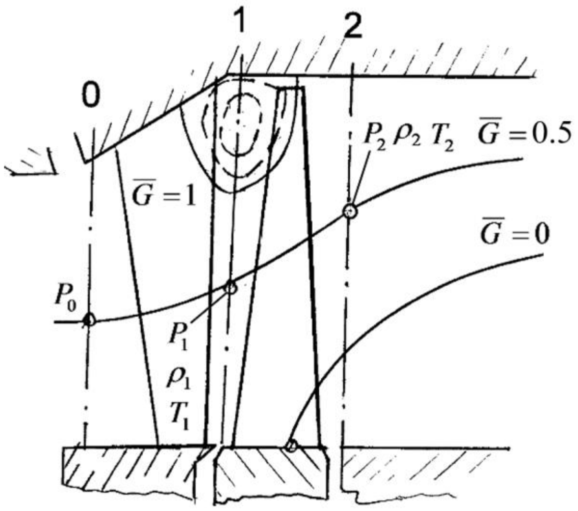

Figure 2. Flow parameters in the stage

The stagnant flow pressure (total pressure) P^* is equal to

$$P_1^* = \rho^* \cdot \frac{C_1^2}{2} + P_1, \qquad P_2^* = \rho^* \cdot \frac{C_2^2}{2} + P_2, \tag{7}$$

where P_1 and P_2 are the static pressure in the stream at the inlet "1" to the rotor wheel and exit "2" from it, the difference of which $\Delta P_{1-2} = P_1 - P_2$ characterizes the compression of the working medium in the stream; C_1, C_2 are velocities of the working medium in the stream in front of the rotor wheel "1" and behind it "2".

When generalizing equation (6), the density of the working medium is taken as an average value determined from the parameters of the stagnant flow using the equation of state

$$\rho_1^* = \frac{P_1^*}{RT_1^*}, \qquad \rho_2^* = \frac{P_2^*}{RT_2^*}, \tag{8}$$

$$\rho^* = \frac{1}{2}(\rho_1^* - \rho_2^*). \tag{9}$$

The difference $\frac{\Delta P^*_{1-2}}{\rho^*}$ is the specific work for an elementary stream expended during its movement in the interblade channels of the rotor wheel.

Having experimentally obtained values P^*, P, angles α_1, $\gamma_м$ and their distribution along the radius for the sections in front of the rotor wheel (section 1) and behind it (section 2), after constructing streamlines and separating elementary streams in the flow ΔG, the specific work h_u can be determined from the equation (5). In this case, the energy losses (heat losses) are defined as

$$h_r = \frac{P_1^* - P_2^*}{\rho^*} - h_u = \frac{\Delta P^*_{1-2}}{\rho^*} - h_u, \tag{10}$$

where the average density of the working medium is determined according to the dependences (8) and (9) at the measured temperatures t_1^* and t_2^*.

According to the accepted flow structure model, a similar approach is applicable to the main flow. In this case, the mass flow rate G_0 is determined at the entrance to the stage and the relative mass flow rate is found from the dependence $\bar{G} = G/G_0$, where the flow rate G is the fractional part of the total flow rate when taking the flow rate in an elementary stream

$$G_i = \sum_{i=0}^{n} \Delta G_i = i\, \Delta G_i,$$

when dividing the flow through the stage into n equal parts.

The parameters of the working medium are taken on the middle line of the selected stream ΔG_i.

A similar approach is used to determine the characteristics of a rotating vortex and the breakaway near a bushing. In this case, the mass flow rate in the vortex is found by the value of the velocity component C_{1z} of the flow that enters the rim clearance from the interblade channels of the rotor wheel. The boundaries of the area of the flow entering the clearance are the value C_{1z} = 0 and the surface of the outside bypass above the rim clearance.

For the breakaway near a bushing, the mass flow rate in the vortex adjacent to the trailing edges of the rotor blades is determined by the reverse flow in the bushing area and the boundaries of the area occupied by the inflowing flow to the trailing edges are the surface of the bushing and the line $\bar{G}$ = 0 (Fig. 2) [82].

For the large fanning stage, the lines of equal flow rates were constructed based on the results of traversing [80], which made it possible to distinguish the flow of the working medium in the flow structure, limited by the lines $\bar{G}$ = 0 and $\bar{G}$ = 1 for all considered modes of the stage operation. At the same time, the main attention is given to the stage operation under the low-flow rate modes [83], which occur when the relative volumetric flow rate $\overline{Gv_2}$ decreases to the value of the idle $\overline{Gv_2}_{\mathrm{id}}$.

To analyze the change in flow characteristics depending on the value $\overline{Gv_2}$, the components of equation (6) can be represented as a pressure difference for the total energy as ΔP^*_{1-2}, the specific work $\rho^* h_u$ and the compression ΔP_{1-2} [84]. This makes it possible to estimate the difference between static and total pressure in the streams of the main flow. In the mode below idle, in which the peripheral part and a large proportion of the central part of the blade operate under the conditions of energy transfer from the sections of the blade to the flow, the difference in static pressures ΔP_{1-2} (except for the bushing area) is close to zero. A further decrease in the mode leads to some compression of the working medium in the streams, the level of which is determined by their position relative to the average value.

The nature of the change in the specific work $\rho^* h_u$ in the streams of the main flow largely repeats the change in the total energy. The development of a rotating vortex and the breakaway near a bushing with a decrease in the mode ($\overline{Gv_2}$ = 0.254 and $\overline{Gv_2}$ = 0.04) leads to an intensification of mass transfer at the boundaries of the main flow, an increase in the specific work spent on changes in the main flow, and an increase in irretrievable energy losses.

The share of energy losses determined according to dependence (10) at $\overline{Gv_2}$ = 0.367 for streamlines $\bar{G}_i$ = 0.95 is ~27%, for $\bar{G}_i$ = 0.8 is ~38.5%. The share of the specific work attributable to the compression of the flow in contact with the rotating vortex is 22% and 25% for the corresponding streams [84].

To obtain quantitative characteristics when the stage operates at the low-flow rate modes, it is necessary to have integral characteristics that determine the level of energy spent to maintain the flow structure and irreversible losses.

An analysis of the change in the specific energy along the stream filaments showed that when the specific work is distributed along the lines of the stream $\bar{G}_i$, in most of the experimentally found dependences $h_{u_i} \sim \bar{G}_i$, the equality is

$$\sum_{i=0.5}^{n} h_{u_i} \Delta G_i = h_u \Delta G,$$

which allows us to take the value $h_{u\bar{G}=0.5}$ on the average streamline as a value for the integral characteristic $\bar{G}_i$ = 0.5

$$\bar{h}_{u\bar{G}=0.5} = \frac{h_{u\bar{G}=0.5}}{h_{u\bar{G}=1.0}} \sim \overline{Gv}_2. \tag{11}$$

The values $h_{u\bar{G}=0.5}$ and $h_{u\bar{G}=1.0}$ pass through zero at the integrally determined value $\overline{Gv}_{2\mathrm{id}}$, which corresponds to the idle mode, in this case dependence (11) will take the following form

$$\bar{h}_{u\bar{G}=0.5} = \frac{\overline{Gv}_2 - \overline{Gv}_{2\mathrm{id}}}{1 - \overline{Gv}_{2\mathrm{id}}}. \tag{12}$$

Considering equation (4), the integral nature of the value $\bar{h}_{u\bar{G}=0.5}$ and $h_{u\mathrm{nom}} = H_{0\mathrm{nom}} \cdot \eta_{u\mathrm{nom}}$, the power consumption attributable to the advancement of the main flow of the working medium through the stage can be represented as

$$\Delta N = \frac{\overline{Gv}_2 - \overline{Gv}_{2\mathrm{id}}}{1 - \overline{Gv}_{2\mathrm{id}}} \cdot \overline{Gv_2} \cdot G_{\mathrm{nom}} \frac{v_{2\mathrm{nom}}}{v_2} \cdot H_{0\mathrm{nom}} \cdot \eta_{u\mathrm{nom}}, \tag{13}$$

where $H_{0\mathrm{nom}}$ is the heat drop triggered in the stage at the nominal mode.

A similar approach can be applied to determine the power consumption required to maintain a rotating vortex and the breakaway near a bushing.

As an example, let us determine the energy costs attributable to the promotion of the main flow through the stage during its operation at the low-flow rate modes in the last stage of T-250/300-240 cogeneration turbine, which has the following geometric characteristics:

– $\alpha_{1\mathrm{eff}}$ = 17.4° is the effective exit angle of the flow from the guide vanes at the average diameter;

– $\beta_{2\mathrm{eff}}$ = 27.84° is the effective exit angle of the flow from the channels of the rotor wheel at the average diameter;

– γ_{m} = 47° is the inclination angle of the peripheral contour of the guide vanes;

– $v_{2\mathrm{nom}}$ = 25 m^3/kg is the specific volume;

– φ = 0.97 and ψ = 0.93 are the speed coefficients for guide vanes and rotor wheel respectively.

For these initial data, the idle of the stage is equal to $\overline{Gv_2}_{\mathrm{id}}$ = 0.30 [83].

To determine the power costs, according to dependence (13), the heat drop triggered in the stage is equal to $H_{0\mathrm{nom}}$ = 175.4 kJ/kg, the efficiency at nominal mode is $\eta_{u\mathrm{nom}}$ = 0.748, the steam consumption at the nominal mode, considering the degree of steam dryness x_2 = 0.925 is G_{nom} = 69.5 kg/s. The steam density $\rho_2 = \frac{1}{v_2}$ is determined considering the steam temperature at the outlet of the main flow from the rotor wheel due to thermal (irreversible) losses [80].

The rotary diaphragm is closed (the swivel ring is on the “stop”), through the leaks between the ring and the diaphragm, only the “ventilation” steam flow enters the flow path of LPC to cool the flow path of the turbine.

The values of the relative volume flow rate of steam are obtained by calculation for a sealed diaphragm equals $\overline{Gv_2}$ = 0.0493 and for an unsealed diaphragm is $\overline{Gv_2}$ = 0.0143 [80].

In this case, based on formula (13), the power consumption is ΔN = 26.8 kW for a sealed diaphragm and ΔN = 110.4 kW for an unsealed one.

Comparison of the power costs for moving through the main flow stage indicates the advantage of using a sealed diaphragm. However, the power costs for the drive of the last stage at the low-flow rate modes also include the costs of maintaining the vortex rotating in the rim clearance and the breakaway near a bushing behind the rotor wheel, the value of which, based on the analysis of the experimentally established values [79], significantly exceeds the power costs for promoting the main flow, especially when the rotary diaphragm is fully closed. Therefore, to determine the total power costs for the operation of the last stage, it is necessary to have a balanced power of all cost components.