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THE STRUCTURE OF THE FLOW BEHIND THE LAST STAGE OF STEAM TURBINE AT THE LOW FLOW RATE OPERATING CONDITIONS

Introduction. *The results of experimental studies, which given in various scholarly research sources, have shown that changes in the flow structure and the formation of a breakaway near a bushing (with a decrease in power) begins in the outlet nozzle and gradually spreads towards the last stage of low-pressure cylinder. As a result, the efficiency of the low-pressure cylinder and the power unit decreases.*

Problem Statement. *The data analysis has shown that recently powerful turbines often operate in off-design modes. This leads to changes in the flow structure (the appearance of the breakaway near a bushing and the vortex rotating in the inter-row clearance) and additional energy losses, especially in the flow path of low-pressure cylinder, leading to erosive wear of trailing edges of the working wheel due to the suction of wet steam from the condenser.*

Purpose. *The purpose of this research is to consider the movement of the working medium behind the working wheel in the outlet nozzle of the low-pressure cylinder and to evaluate the development of the breakaway near a bushing in the axisymmetric setting under the low-flow conditions to obtain dependences that allow analyzing the operation of turbine stages with a large fanning and preventing the turbine operation under the low-flow rate conditions.*

Materials and Methods. *The methods for mathematical modeling of the processes in the flowing part of the turbine under low-flow rate conditions have been chosen based on the experimental research of V.A. Khaimov.*

Results. *The characteristics of the interaction of rotor blades with the breakaway near a bushing have been given. The dependences that enable determining the characteristics of the flow behind the working wheel when it flows out into the outlet nozzle have been obtained.*

Conclusions. *The proposed analytical methodology allows the use of a rational approach to the operation under the low-flow rate conditions and the prevention of erosion wear of working blades trailing edges of the last stages of low-pressure cylinder.*

Keywords: breakaway near a bushing, outlet nozzle, low-flow rate mode, flow structure, and low-pressure cylinder.

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The results of experimental studies of turbine stage models have shown that in the presence of an annular space behind the working wheel with a cylindrical outer bypass and bushing, an axisymmetric flow with its breakaway from the bushing is formed in low-flow rate modes.

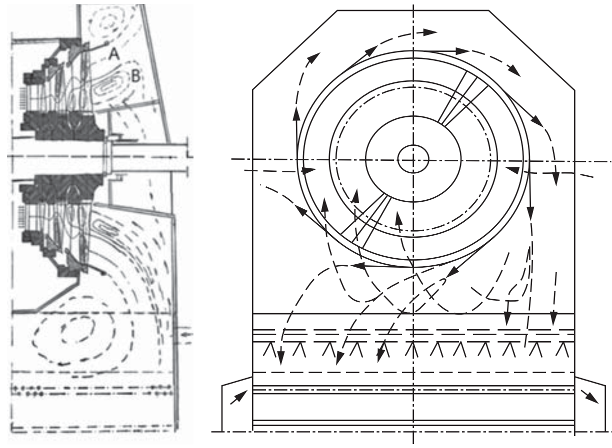
The steam outlet in turbines is made through an outlet nozzle connected to the condenser. The outlet nozzle has a vertical plane of symmetry passing through the axis of the turbine and provides a uniform steam load of the condenser at the axial flow exit from the working wheel at the nominal mode. In this case, the stiffening ribs of the outlet nozzle do not create large flow breakaway when steam moves from the working wheel to the surface of the condenser tube bundle (Fig. 1).

The operating turbine mode decreases because a drop in the steam rate, including through the working wheel of the last stage. In this case, a circumferential component of the outlet velocity C_{2u} arises behind the working wheel and the flow in relation to the plane of working blades trailing edges exits at an angle $a_2 \neq 90°$, i.e. it receives a swirl in the circumferential direction.

Investigation of the flow rotating in the annular space behind the guiding apparatus at small angles a_1 showed that the stability criterion is tga_1 on the midline of the swirling flow behind the guiding apparatus [1]. The analysis of the flow behind the working wheel at low-flow modes showed that the "swirling" of the flow is the working wheel, which does not have time to unwind the rotating flow created by the guiding apparatus [2 — 6] and the angle a_2, which taken on the streamline $\overline{G} = 0.5$ can be taken as a characteristic of its swirl.

The presence of the outlet nozzle with a one-way outlet of the steam flow to the condenser with a decrease in the flow rate breaks the symmetry of the flow leaving the working wheel, including breakaway near a bushing outside the working wheel.

Investigation of the temperature state of the last stage carried out by E. V. Uriev [7], showed that the symmetry of the interacting flows: main



a *b*

Fig. 1. Structure of the flow behind the working wheel of the last stage with a large fanning: *a* — longitudinal section; *b* — cross section; *A* — main flow; *B* — breakaway near a bushing

and reverse from the area near a bushing breakaway along the bushing surface is observed within the working wheel.

The movement of the working medium behind the working wheel under low-flow rate modes and the development of breakaway near a bushing in an axisymmetric setting are considered in [8]. In this case, the steam pressure near working blades trailing edges becomes lower than the pressure in the condenser, which ensures the formation and maintenance of reverse steam currents from the condenser to the area near a bushing breakaway. In real conditions, the rotating flow runs in the stiffening ribs and reverse currents are created in this flow (in areas near a bushing breakaway), along which steam is sucked from the condenser into the area near a bushing breakaway (Fig. 1). The driving force for the reverse currents is the reduced pressure in the area near a bushing breakaway, created by the rotation of the main flow.

Let us further consider the change in the structure of the rotating steam flow and its effect on the breakaway near a bushing.

Breakaway near a bushing. The energy consumption by the working wheel to maintain the breakaway near a bushing begins with the steam flow rate $\overline{Gv}_{2_{z=0}}$, i. e. from the steam flow rate cor-

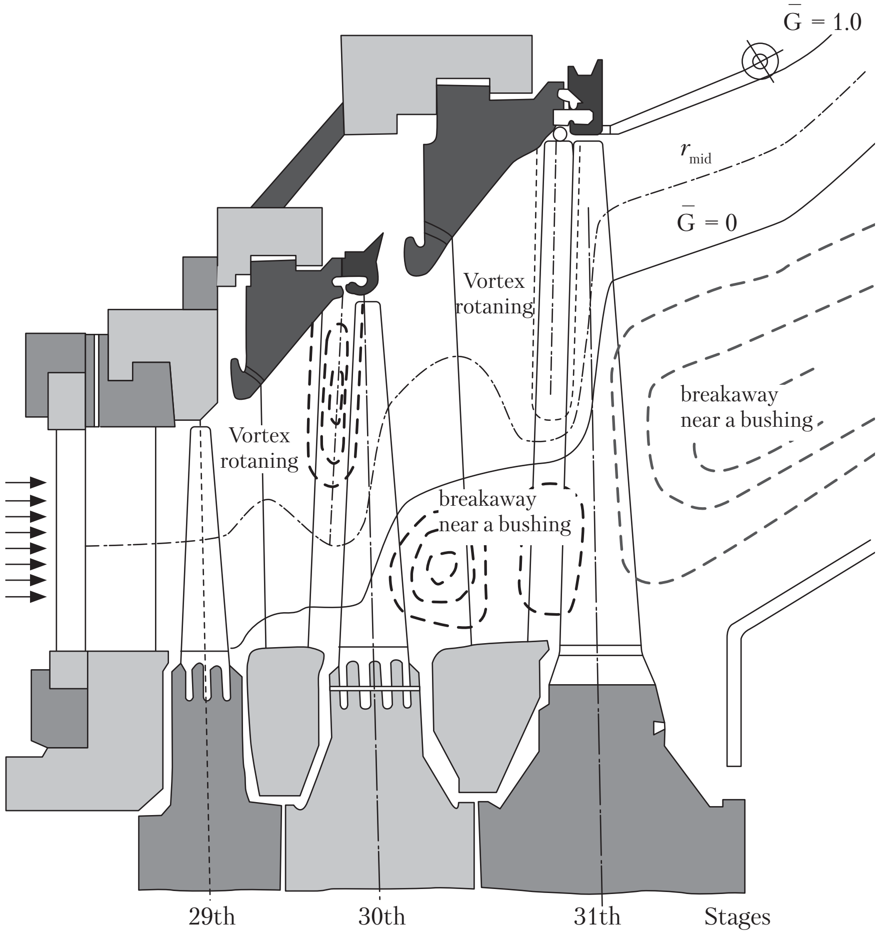


Fig. 2. Areas of vortex flows in the flow path of cogeneration turbine low-pressure cylinder at low-flow rate modes

responding to the connection of the breakaway near a bushing to working blades trailing edges ($\overline{Gv}_2 < \overline{Gv}_{2_{z=0}}$). Before joining the line $\overline{G} = 0$ of the main flow to working blades edges, the breakaway near a bushing develops outside the channels of the working wheel. However, areas near a bushing breakaway on the convex side of blades are formed inside the interblade channels, which do not connect with the area near a bushing breakaway behind the working wheel [8—10]. The main flow passes along the channel, partially rising in the radial direction. In this case, the position of streamlines within the channels is not considered.

With a decrease in $\overline{Gv}_2$ inside the channels the movement of the working medium in the radial direction increases. Behind working blades trailing edges, narrow jets of the main flow are mixed, a part of the working medium enters the upper area near a bushing breakaway (Fig. 2).

The flow inside the working wheel channels, at the first stage, moving along the conical surfaces, passing with a decrease in $\overline{Gv}_2$ to the radial direction, comes out at a radius larger than at the inlet.

With the formation of a vortex rotating in the inter-row clearance the working medium movement in the radial direction is enhanced. The flow, leaving the working wheel into the radial clearance is decelerated against the cylindrical surface above the working blades and is divided into two branches in the periphery. One branch is directed towards the inter-row clearance and forms a ro-

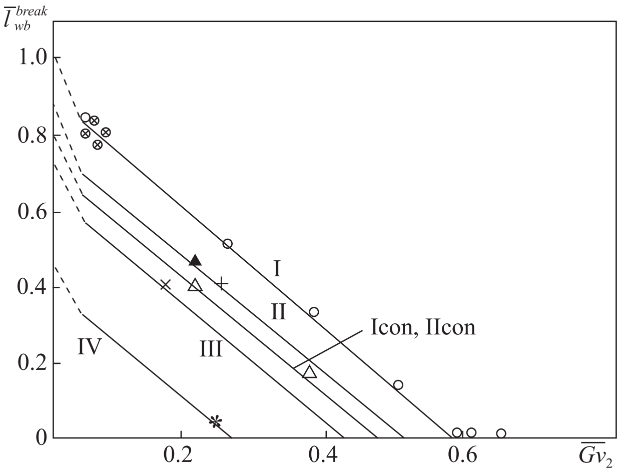


Fig. 3. Formation of the breakaway near a bushing depending on the height of the working blade and the volumetric steam flow: I—IV are experimental model stages [8]; Icon, IIcon are stages with a conical meridian bypass in the guiding apparatus

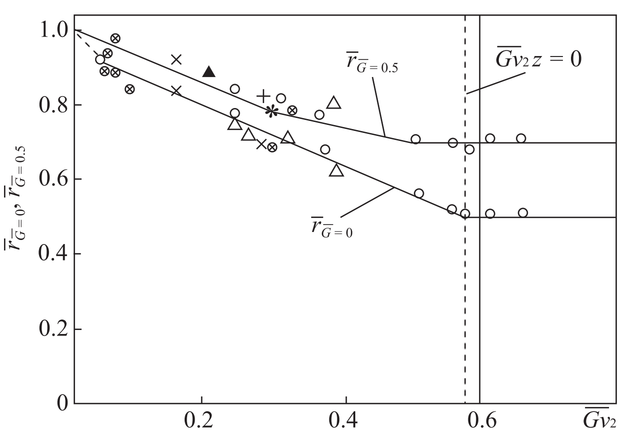


Fig. 4. Change in the position on the point attachment radius of the line $\bar{G} = 0$ and $\bar{G} = 0.5$ to working blades trailing edges

tating vortex and the other branch towards working blades trailing edges.

The structure of the flow during its outflow into the outlet nozzle and the movement in it differs from its movement in the annular channel of the simulating experimental object.

The rotating flow behind the working wheel moves in the outlet nozzle along the ruled surfaces and the breakaway near a bushing from the bushing surface occurs at a relatively small circumferential velocity component $\bar{C}_{2u}$ behind the working wheel.

Interaction of working blades with the breakaway near a bushing. As characteristics of the interaction of working blades with the breakaway near a bushing are considered:

- the relative volumetric steam flow rate $\overline{Gv}_{2_{z=0}} = \overline{Gv}_{2_{z=0}}/(\overline{Gv}_2)_{nom}$ corresponding to the moment of the boundary line $\bar{G} = 0$ connection to working blades trailing edges $\overline{Gv}_{2_{z=0}}$ at the radius of the bushing r_{bush}, where $\overline{Gv}_{2_{z=0}}$ is the volumetric steam flow rate at the considered mode, $(\overline{Gv}_2)_{nom}$ is the volumetric steam flow rate at the nominal mode;
- the position of the boundary line $\bar{G} = 0$ at working blades trailing edges $r_{\bar{G}=0}$ in different modes, characterizing the development of interaction of the breakaway near a bushing with the working wheel;
- the position of the line $\bar{G} = 0.5$ on working blades edges $r_{\bar{G}=0}$;
- the angle a_2 as a factor of flow swirling by the working wheel;
- the influence of the angle β_{2eff}^{mid} on the stage idle mode.

Processing the results of experimental studies [8, 11, 12] made it possible to obtain dependencies for determining the named characteristics.

The point position of the line attachment $\bar{G} = 0$ to working blades trailing edges for stages of different fanning, depending on the mode of their operation is considered in Fig. 3. The value $\overline{Gv}_{2_{z=0}}$ at the bushing level (zero length of the working blade $\bar{l} = 0$) is determined. This value is the relative volumetric flow rate corresponding to the attachment of breakaway near a bushing to the working wheel, which characterizes the moment when the working wheel starts transmitting power to the breakaway near a bushing, in addition to the transmission of energy by the main flow behind the working wheel.

The attachment of the line $\bar{G} = 0$ to working blades trailing edges is characterized by a decrease in $\overline{Gv}_2$ by the movement of the attachment point up to $\overline{Gv}_2 < 0.05$. In the range of variation $0 < \overline{Gv}_2 \leq 0.05$ along working blades trailing edges there is a significant change in parameters and characteristics of the flow, which is also accompanied by a change in angle a_1 at the outlet from the guiding apparatus.

The movement of the position on the radius of the attachment point of the line $\overline{G} = 0$ ($\overline{r}_{\overline{G}=0}$) to working blades trailing edges is considered in Fig. 4. It has a general character for the studied stages. A similar character is also observed for the line $\overline{G} = 0.5$ ($\overline{r}_{\overline{G}=0}$).

The breakaway near a bushing at $\overline{Gv}_2 > \overline{Gv}_{2_{z=0}}$ is deleted from working blades and does not affect the structure of the working medium movement inside the interblade channels of the working wheel.

Results of the study. The behavior analysis of the circumferential velocity component $\overline{C}_{2u}$ (Fig. 5) showed that in the entire area of its variation (including the turbine mode for stage II) it is linear and increases when the mode decreases.

The main influence on the change in the angle a_2 on the line $\overline{G} = 0.5$ is exerted by the axial component of the velocity $\overline{C}_{2z}$ which has three areas: $0 < \overline{Gv}_2 \le 0.3$; $0.3 < \overline{Gv}_2 \le 0.45$ in which this change corresponds to the condition $\overline{C}_{2u} = \text{const}$ and the area $0.45 < \overline{Gv}_2 \le 1.0$ in which $\overline{C}_{2z}$ decreases with a drop in $\overline{Gv}_2$; $\mathrm{tg}a_2$ is defined as the ratio of the output velocity components on the line $\overline{G} = 0.5$

$$\mathrm{tg}a_2 = \overline{C}_{2z} / \overline{C}_{2u}. \qquad (1)$$

Angle a_2 is determined by the direct measurement.

The character of change in the ratio $\overline{C}_{2z} / \overline{C}_{2u}$ on the line $\overline{G} = 0.5$ makes it possible to interpret mode as *the turbine mode*, up to close to the lower boundary of the transition area of working blades sections when $\overline{Gv}_2 \approx 0.46 \div 0.47$, *the transitional mode* when $\overline{Gv}_2 = 0.3 \div 0.46$ and *the mode of ventilation losses* development when $0 < \overline{Gv}_2 < 0.3$ [13—15]. This mode is also accompanied by an increase in the maximum temperature of the working medium (steam) at the outlet of the working wheel [7].

The analysis of changes in $\overline{Gv}_{2id}$ and $\overline{Gv}_{2_{z=0}}$ with decreasing the mode showed that the function $\mathrm{tg}^2\beta_{2eff}^{mid}$ can use as their characteristics when swirling the flow behind the working wheel (Fig. 6).

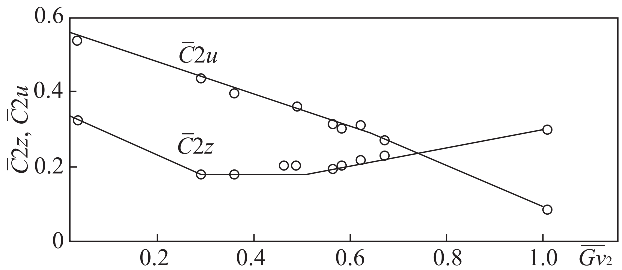


Fig. 5. Change in the projection of the velocity $\overline{C}$ behind the working wheel with a decrease in the mode on the line $\overline{G} = 0.5$

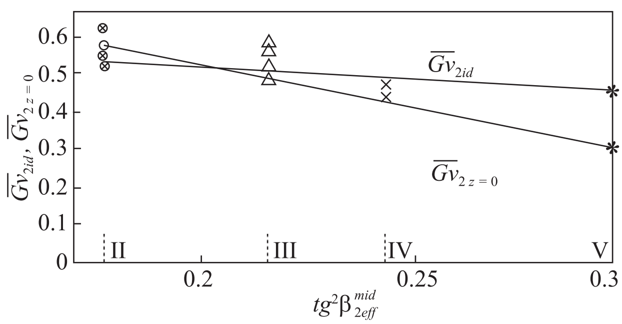


Fig. 6. Dependence of relative volumetric flow values on $tg^2\beta_{2\,eff}^{\,mid}$ (at $\gamma_m = 0°$): II—V are experimental model stages [8]

The character of these characteristics for an incompressible working medium can be represented by dependences

$$\overline{Gv}_{2id} = 0.665\,(1 - 1.048\mathrm{tg}^2\beta_{2eff}^{mid} - \sqrt{0.1\mathrm{tg}\gamma_{\mathrm{m}}}), \qquad (2)$$

$$\overline{Gv}_{2_{z=0}} = 1.094\,(1 - 2.587\mathrm{tg}^2\beta_{2eff}^{mid} - \sqrt{0.029\mathrm{tg}\gamma_{\mathrm{m}}}), \qquad (3)$$

and their ratio for the studied stages with a fanning from 0.219 to 0.358 at change in the angle $\beta_{2eff}^{mid} = 23° \div 28.5°$ is determined as follows

$$\overline{Gv}_{2id} / \overline{Gv}_{2_{z=0}} = 1.491\,(1 - 2.245\mathrm{tg}^2\beta_{2eff}^{mid} - \sqrt{0.0215\mathrm{tg}\gamma_{\mathrm{m}}}). \qquad (4)$$

These dependences allow determining the flow characteristics behind the working wheel when its flows out into the outlet nozzle.

The calculation of characteristics at the inlet to the outlet nozzle was completed with the use of the example of the last (31st) stage of T-250/300—240 turbine. The geometrical characteristics of the stage are as follows: the effective angle of the flow exit from the working wheel $\beta_{2eff}^{mid} = 27.8°$ and the angle of meridional bypass inclination in the guiding apparatus $\gamma_m = 47°$.

Computational studies were carried out with the use of the above characteristic dependences (2), (3), which made it possible to obtain the following values $\overline{Gv}_{2id} = 0.40$ and $\overline{Gv}_{2_{z=0}} = 0.34$.

Comparison of the experimentally obtained value on a full-scale object, which was performed by V. A. Khaimov [11] (turbine operated at the combined cycle power plant) $\overline{Gv}_{2id} \approx 0.31$ and values calculated by the proposed method show satisfactory coincidence.

Comparing the calculated values with the experimentally obtained one has shown that they differ by 15%, on average. This is admissible for problems of such complexity when passing from a multivariate problem to its solution in a one-dimensional formulation.

The proposed method allows for calculated value $\overline{Gv}_{2id}$ to evaluate the range of turbine operation regulation, if the last stage does not consume power.

Conclusions. The analysis of scholarly research literature sources based on theoretical research and operating experience of powerful co-generation steam turbines has shown that at present they are working mainly in off-design (partial), so-called the low-flow rate modes. Such modes are characterized by complex vortex structures of flow movement, especially behind the working wheel in the outlet nozzle of low-pressure cylinder.

The complex vortex structures lead to the emergence of additional energy losses associated with the breakaway near a bushing, the vortex rotating in the inter-row clearance, as well as with erosive wear of trailing edges of the working wheel due to the suction of wet steam from the condenser directly to the working blades trailing edges of the last stage. The obtained dependences make it possible to analyze the operation of turbine stages with a large fanning and, if possible, prevent the operation of the turbine in these modes.

The experimental value $\overline{Gv}_{2id}$ at the turbine T-250/300-240, which was obtained by V. A. Khaimov, has been compared with the values calculated by the proposed method. The comparison has shown a satisfactory coincidence (the average discrepancy is 15%) that is allowable for problems of such complexity.

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СТРУКТУРА ПОТОКУ ЗА ОСТАННІМ СТУПЕНЕМ ПАРОВОЇ ТУРБІНИ ПРИ МАЛОВИТРАТНИХ РЕЖИМАХ ЕКСПЛУАТАЦІЇ

Вступ. Результати експериментальних досліджень, наведених в різних наукових джерелах, показали, що зміна структури потоку й формування привтулкового відриву (при зниженні потужності) починається у вихідному патрубку та поступово поширюється в бік останнього ступеня циліндра низького тиску. Це спричиняє зниження ефективності роботи циліндра низького тиску та усього енергоблоку в цілому.

Проблематика. Аналіз даних показав, що останнім часом потужні турбіни часто працюють на нерозрахункових режимах, що призводить до зміни структури потоку (поява привтулкового відриву та вихору, що обертається у міжвінцевому зазорі) та виникнення додаткових втрат енергії, зокрема в проточній частині циліндра низького тиску. Це спричиняє ерозійне зношення вихідних кромок робочого колеса за рахунок підсмоктування вологої пари з конденсатора.

Мета. Розгляд руху робочого середовища за робочим колесом у вихідному патрубку циліндра низького тиску та оцінювання щодо розвитку привтулкового відриву в осьосиметричній постановці при маловитратних режимах для отримання залежностей, які дозволять аналізувати роботу турбінних ступенів великої віяловості та не допускати експлуатацію турбіни на маловитратних режимах.

Матеріали й методи. За основу взято експериментальні дослідження Хаімова В.А., до яких застосовано методи математичного моделювання процесів, що протікають у проточній частині турбіни на маловитратних режимах експлуатації.

Результати. Наведено характеристики взаємодії робочих лопаток з привтулковим відривом. Отримано залежності, які дозволяють визначити характеристики потоку за робочим колесом при його течії до вихідного патрубку.

Висновки. Запропонований аналітичний підхід дозволить раціонально підходити до експлуатації на маловитратних режимах і не допускати ерозійного зносу вихідних кромок робочих лопаток останніх ступенів циліндра низького тиску.

Ключові слова: циліндр низького тиску, привтулковий відрив, вихідний патрубок, маловитратний режим, рух робочого середовища.