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# EXPERIMENTAL STUDY OF THE MODEL COMPARTMENT OF THE LOW-PRESSURE CYLINDER OF K-320-240 TURBINES OF JSC "TURBOATOM"

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*A reduction in the efficiency of the last stages of powerful steam turbines, as well as their reliability due to an increase in the disturbing forces acting on the blades, is an urgent study subject, which requires a comprehensive analysis of gas-dynamic processes in these parts of the turbines. To determine the integral and gas-dynamic characteristics of the last LP stages of K-320-240 turbines under variable operating conditions, and develop recommendations for their further improvement, experimental studies of the model flow path of the cylinder compartment were conducted using the gas-dynamic test stand 0102 of Turboatom JSC. As a result, it was established that closing the moisture removal slot increases the efficiency of the last LP stage by 2–7% in the range of volumetric steam flow rate changes from 900 to 2900 m$^3$/s. With a nominal operating mode, the efficiency of the two penultimate LP stages is 0.786. With a significant change in the volumetric steam flow rate downstream of the last stage in the previously specified limits, the compartment efficiency changes by less than 0.9%. As a result of the experimental studies, there was determined the possibility of increasing the efficiency of the flow path of the K-320-23, 5-4 turbine LPC. The results of the above studies were used to modernize the K-320-240 turbines, which made it possible to increase the efficiency of the compartments of the last three LPC stages by 1.3%, of which 0.9% were due to the modernization of the third stage and 0.4%, due to the sealing of the fourth stage radial clearance, and use the results of the study to improve the LPC of the K-325-23.5 turbine.*

***Keywords:*** *volumetric steam flow rate, low pressure cylinder, steam turbine, efficiency.*

## Introduction

The last stages of the low-pressure cylinders (LPC) of powerful steam turbines, unlike other stages, have a number of significant peculiarities that are manifested both in nominal and variable turbine operating modes [1].

In the nominal mode, the last stages are characterized by large thermal differences, which leads to a sharp opening of the stage meridional contour. This circumstance leads to the flow around the profiles on oblique surfaces in a variable thickness layer and possibility of a diffuser flow in the peripheral zone of nozzle channels, i.e. to a significant increase in losses in the nozzle diaphragm. An increase in the opening angle of the outer contour leads to an increase in losses not only in the peripheral regions, but also at the LP stage root. The increase in the opening angles of the LP flow path and Mach numbers $M_{c1}$ is associated with a significant increase in the gradient of the degree of reactivity in comparison with the stages in which the subsonic and streamline surface flows are close to cylindrical ones. If this factor is not taken into account, then in the stages with a small hub-tip ratio, a significant negative degree of reactivity in the root zone is observed. The result of the diffuser nature of the flow in the passages of the working blades in this zone is a decrease in efficiency, exacerbated by the loss of stability and flow separation in some cases, even in the nominal mode. With a decrease in the relative volumetric steam flow rate at the rotor outlet, $\overline{Gv_2}$, a further decrease in the root degree of reactivity occurs, accompanied by a rise in streamlines to the rotor outlet, which causes an off-design oblique flow around the profiles, a decrease in the throughput of rotor root zones, and, in the long run, an increase in losses both with turbine exit losses and flow separation in the hub [2].

The increase in energy losses at the periphery and, especially, the separation of the flow in the hub under partial conditions is the cause of not only efficiency deterioration, but also an increase in the disturbing forces acting on the blades, and therefore, a decrease in their reliability [3].

The analysis of the results of computational and experimental studies of well-known publications made it possible to develop constructive recommendations, the combination of which eliminates a number of disadvantages inherent in the last stages, and increases their efficiency and reliability in a wide range of loads (when the relative volumetric steam flow rate downstream of the last stage changes from 1 to 0.5). This made it possible to design full-scale turbine stages with a small ratio $D/l$=2.43; 2.87; 4.5, and install them in the LPCs of high-power turbines produced by JSC "Turboatom" [4].

**Problem Statement and Solution Method**

To determine the integral and gas-dynamic characteristics of the last LP stages of a K-320-240 turbine under variable operating conditions, and develop recommendations for their further improvement, at the gas-dynamic test stand 0102 of JSC Turboatom, there have been performed experimental studies of the model flow path of the compartment that consists of the third, fourth, and fifth LP stages, with the last-stage blade having a length of $l$=1030 mm ($D/l$=2.46), and is installed in the case of the experimental steam turbine with a divided shaft. All the flow path dimensions correspond to full-scale ones, based on the accepted modeling scale of 1:3, with the exception of the thicknesses of the trailing edges of the working and nozzle blades. The experimental steam turbine test stand 0102 is designed to study and test the compartments of LP stages operating in superheated and wet steam.

The research program provided for the determination of the integral characteristics of a three-stage compartment, compartments of the two penultimate stages and, separately, the last stage compartment in a wide $Gv_2$ change range covering all the main operating modes of K-320-240-4, K-310-240-3, K-750- 65/3000 turbines.

**Research Results**

As a nominal mode, we adopted the operating mode of the K-320-240-4 turbine with the 1977 m$^3$/s volumetric steam flow through its last stage.

$Gv_2$ values varied both due to changes in the steam mass flow rate and (or) the specific volume by changing both the initial pressure upstream of the LPC and the final pressure downstream of the compartment. The value $Gv_2$ was determined by the total steam flow rate $G$ through the stage, including leakage into the moisture removal slot upstream of the rotor.

Fig. 1 shows data on the last LP stage efficiency. Curve 1 corresponds to the standard version of the flow path with the moisture removal slot upstream of the working blade (the slot upstream of the last stage working blade is open). In this version, the stage reaches its maximum efficiency $\eta_{oi}$=0.693 at the volumetric steam flow rate $Gv_2$=1750 m$^3$/s.

As can be seen in Fig. 1, this stage version has an efficiency close to the maximum one, with the values of volumetric steam flow rate $Gv_2$ from 1,500 to 1,900 m$^3$/s. With an overload in the flow rate of about 1.5 times ($Gv_2$=2,500 m$^3$/s), due to an increase in turbine exit losses, the stage efficiency decreases to ~0.61.

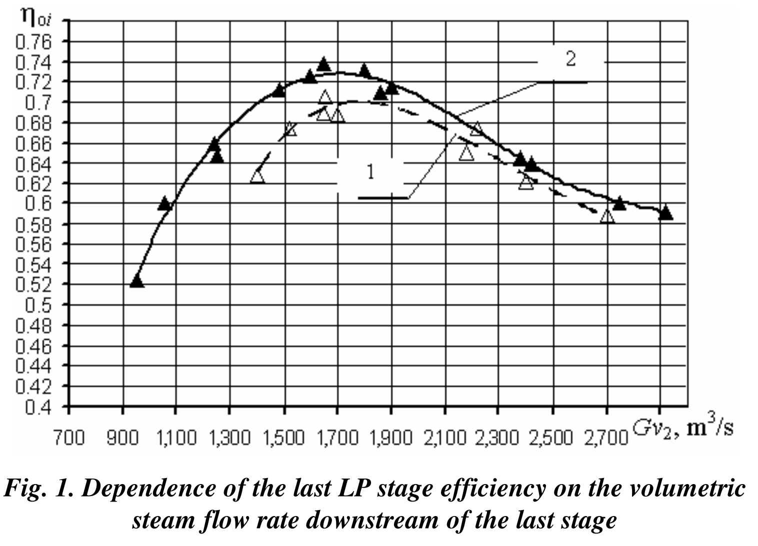


***Fig. 1. Dependence of the last LP stage efficiency on the volumetric steam flow rate downstream of the last stage***

Curve 2 corresponds to the flow path version without the moisture removal slot upstream of the working blades (the slot is closed). The organization of the moisture removal slot is an effective way of preventing the erosion of the working blades, however, at the same time, the flow rate of the steam-water mixture coming from the flow path into the moisture removal slot is significant. So, for the model last LP stage, the calculated leakage value is 8–10% of the steam flow through the stage, which should lead to a significant decrease in the stage efficiency and, consequently, the entire LPC efficiency.

As can be seen from the comparison of curves 1 and 2, closing the moisture removal slot increases the last stage efficiency by 2–7% in the above range of the volumetric steam flow rate at the maximum standard stage efficiency $\eta_{oi}$=0.7.

The data on the efficiency of the compartments of the two penultimate LP stages are shown in Fig. 2.

As can be seen from Fig. 2, in the nominal operating mode ($Gv_2$=1977 m$^3$/s), the efficiency of the two penultimate stages $\eta_{oi}$=0.786. With a significant change in the volumetric steam flow rate downstream of the last stage from $Gv_2$=900 to $Gv_2$=2900 m$^3$/s, the compartment efficiency changes by less than 0.9%.

Fig. 3 presents data on the efficiency of the compartments of the last three LP stages. Curve 1 corresponds to the standard version with the moisture removal slot upstream of the last stage working blade (the slot is open). In this case, the compartment efficiency in the range of the volumetric steam flow rate $Gv_2$ 1400–2700 m$^3$/s is from $\eta_{oi}$=0.746 to $\eta_{oi}$=0.695. The maximum compartment efficiency $\eta_{oi}$=0.755 lies in the range of $Gv_2$ regimes of 1500–1800 m$^3$/ s.

Closing the moisture removal slot leads to an increase in the compartment efficiency $\eta_{oi}$ in the considered range of volumetric steam flow rate changes by 1–2.5%, depending on the load (see Fig. 3, curve 2). At $Gv_2$=1,900 m$^3$/s, the compartment efficiency $\eta_{oi}$=0.77.

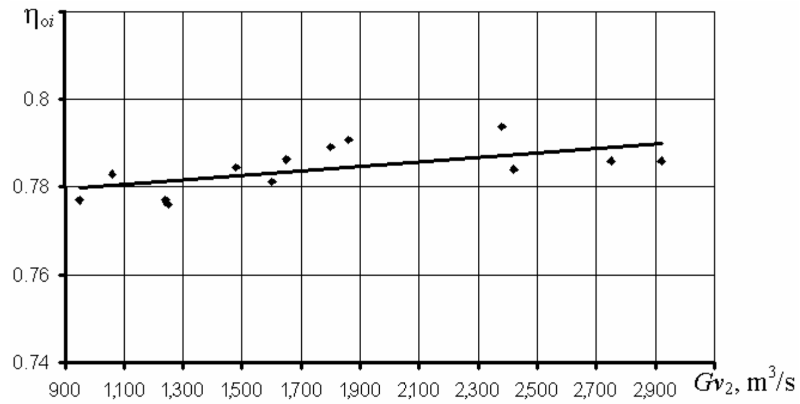


***Fig. 2. Dependence of the efficiency of the compartment of the third and fourth LPC stages from the volumetric steam flow rate downstream of the last stage***

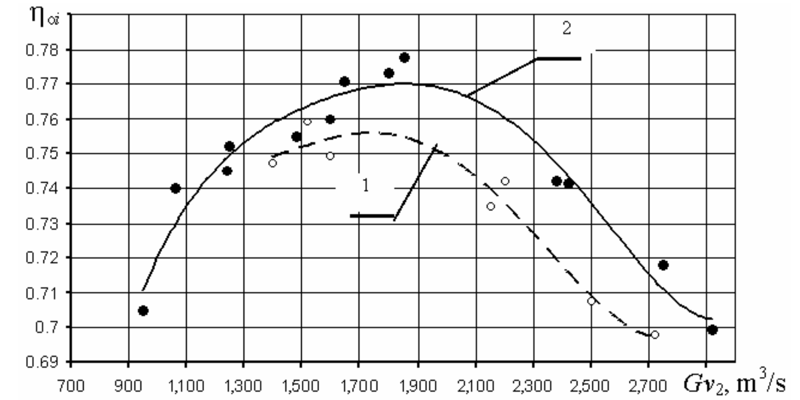


***Fig. 3. Dependence of the efficiency of the compartments of the last three LP stages on the volumetric steam flow rate downstream of the last stage***

With an increase or decrease in volumetric steam flow rate, the compartment efficiency decreases mainly due to a decrease in the last stage efficiency (see Fig. 1).

To determine the gas-dynamic characteristics of the last three LP stages with the last stage working blade length $l$=1030 mm, the flow was traversed by five-channel gas-dynamic probes at the third stage inlet, downstream of the rotors of the third, fourth and fifth stages. It showed that with a change in pressure and temperature along the radius downstream of the third-stage working blades, starting from the height $\bar{l} = 0.2$, there is a high radial gradient of both static $p_{2st}$ and total pressure $p_2^*$. In the region where $l$=0–0.2, the values $p_{2st}$ and $p_2^*$ are practically constant. The increase in static pressure at the blade root is due to the presence of blade root leakages in the stage rim clearance.

An increase in the dynamic pressure to the peripheral boundary indicates an uneven distribution of turbine exit losses. The drop in the total pressure $p_2^*$ at the height $l$=0.97–1.0 is a consequence of the formation of an extensive stagnation region with an increased static pressure downstream of the thickened blade shroud.

The presence of zones of increased losses both in the root and on the stage periphery is indicated by the radial distribution of the braking temperature $t^*$. The flow core with the minimum braking temperature $t^*$ occupies the height of $l$=0.3–0.8 of the rotor.

The experiments made it possible to identify both the third stage design defects and methodology drawbacks that were taken into account when designing the LPC of the K-325-23.5 turbine.

The modes, at which the fourth stage operation is considered, are characterized by the volumetric steam flow rate downstream of the last stage. In all study modes ($\overline{Gv_2}$=0.836; 0.59; 0.43), the static pressure $p_{2st}$ is almost constant over the entire stage height, with the exception of the radial clearance region, where $p_{2st}$ drops sharply due to leakage through the radial clearance. With decreasing load, the static pressure level downstream of the fourth stage decreases, which is related with a significant drop in the available difference in the last LP stage and a decrease in the initial pressure.

The total pressure $p_2^*$ in all modes is constant in height, except for the shroud wire region, where the value of $p_2^*$ decreases. The stage has increased losses due to leakage through the radial clearance. Thus, in the leakage jet core, the total pressure $p_2^*$ increases by 40–60% compared with the one in the main stream downstream of the stage.

The traverse results downstream of the fifth stage showed that in the whole range of test modes ($\overline{Gv_2}$=1.08; 0.88; 0.622; 0.47), the static pressure $p_{2st}$ downstream of the stage at the blade height $l$=0–0.65 remains constant, increasing slightly to the periphery. This is due to the off-design flow around the working blades due to leakage into the moisture removal slot, as well as the presence of a guiding visor above the working blades.

In the modes close to the nominal one ($\overline{Gv_2}$=0.88–1.08), the total pressure $p_2^*$ is almost constant over the entire height, with the exception of the shroud wire and radial clearance regions. In the operating mode $\overline{Gv_2}$=0.622, an increase in $p_2^*$ to the periphery is observed, while in the operating mode $\overline{Gv_2}$=0.74, a positive gradient of the total pressure occurs already along the entire height. The characteristics of the flow from the stage are close to axial ones in the mode $\overline{Gv_2}$=0.47, which is due to the swirling of the working blades at the selected level and gradient of the reactivity degree. At higher loads, the $\alpha_2$ angles in the lower half of the stage decrease, becoming $\alpha_2$<90°.

Steam leakage into the moisture removal slot and the resulting decrease in steam flow through the upper half of the stage ($\bar{l}$=0.7–1.0) leads both to the appearance of undetermined design-related losses on the working blades and an increase in $\alpha_2$ angles.

With a decrease in load ($\overline{Gv_2}$=0.47 mode), the overall level of angles $\alpha_2$ (from 120 to 155°) increases. In the blade root zone, which primarily responds to the changes in load, the $\alpha_2$ angles approach 180°.

In the stage, there is a significantly reduced gradient of the reactivity degree. The Mach number $\mathrm{M}w_2$ at large $\overline{Gv_2}$ is almost constant in height, which is a favorable condition for the design of working blades. With a decrease in steam consumption, the radial gradient of $\mathrm{M}w_2$ is observed only in the root zone due to a decrease in the reactivity degree.

The analysis of the stage operation in variable modes makes it possible to state that the combination of the selected value $\Delta\alpha_1=\alpha_{1p}-\alpha_{1r}$= -8.6° of the reverse swirl of the guide vanes with specially adopted measures for compressing the flow to the hub in the nominal mode neutralizes the formation of the flow separation in the blade root into the stage up to the mode $\overline{Gv_2}$=0.47. The boundary of the separated flow in the stage can be inferred by the change in $p_2^*/p_{2st}$. As $p_2^*/p_{2st}$ approaches unity, signs of the ventilation of root sections begin to appear.

The approximate calculation of the stage transition mode to the power absorption mode was performed by the method [2], showing that for the last LP stages of K-320-240 turbines this mode occurs at $\overline{Gv_2}$=0.29–0.32. Thus, the actual distribution of parameters downstream of the stage shows, on the one hand, correspondence of the stage gas-dynamic characteristics with the design ones, and, on the other, the significant influence of both the moisture removal slot and visor above the working blades, which was not taken into account during design.

## Conclusions

It follows from the foregoing that it is advisable to install the stage under study in the LPCs of turbines operating under conditions of increased pressure in the condenser or at variable loads. However, it should be understood that at low pressure in a heavily loaded condenser, the last stage is overloaded, which leads to a decrease in efficiency.

As a result of the experimental studies, we determined the possibility of increasing the efficiency of the flow path of the LPC of the K-320-23.5-4 turbine with the last stage working blade length of $l$=1,030 mm due to:

– the changes in the shape of the shroud of the third stage working blades;

– the alignment of the third stage working blades with the relative pitch $\bar{t}$ and angles $\beta_1$ and $\beta_2$;

– the sealing of the fourth stage radial clearance;

– the installation of the exit cone immediately downstream of the last stage;

– the reduction in the size of the moisture removal slot upstream of the last stage rotor or its elimination while ensuring the anti-erosion reliability of the blades by other structural methods.

The results of the studies performed were used to modernize the K-320-240 type turbines, which made it possible to increase the efficiency of the compartments of the last three stages by 1.3%, of which 0.9% is due to the modernization of the third stage and 0.4% due to the sealing of the fourth stage radial clearance, and use the results of the study to improve the LPC of the K-325-23.5 turbine.

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## Експериментальне дослідження модельного відсіку циліндра низького тиску турбіни К-320-240 АТ «Турбоатом»

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*Зниження коефіцієнта корисної дії (ККД) останніх ступенів потужних парових турбін, а також їх надійності внаслідок збільшення збурюючих сил, які діють на лопатки, є актуальним предметом дослідження, що потребує всебічного аналізу газодинамічних процесів в цих частинах турбін. Для визначення інтегральних та газодинамічних характеристик останніх ступенів циліндра низького тиску (ЦНТ) турбіни К-320-240 за змінних режимів роботи та розробки рекомендацій щодо їх подальшого удосконалення на газодинамічному стенді 0102 АТ «Турбоатом» були проведені експериментальні дослідження модельної проточної частини відсіку циліндра. В результаті* *встановлено, що закриття щілини вологовидалення підвищує ККД останнього ступеня на 2–7% в інтервалі зміни об'ємної витрати від 900 до 2900 м³/с. За номінального режиму роботи ККД двох передостанніх ступенів складає 0,786. За суттєвої зміни об'ємної витрати за останнім ступенем у вказаних раніше границях ККД відсіку змінюється менше ніж на 0,9 %. В результаті проведених експериментальних досліджень була визначена можливість підвищення економічності проточної частини ЦНТ турбіни К-320-23,5-4. Результати проведених досліджень використані під час модернізації турбін типу К-320-240, що дозволило підвищити ККД відсіку трьох останніх ступенів на 1,3 %, з яких 0,9 % – за рахунок модернізації 3-го ступеня та 0,4 % – за рахунок ущільнення радіального зазору 4-го ступеня, та використовувати результати дослідження для удосконалення ЦНТ турбіни К-325,23,5.*

***Ключові слова:*** *об'ємна витрата пари, циліндр низького тиску, парова турбіна, коефіцієнт корисної дії.*

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