DEVELOPMENT AND RESEARCH OF EFFICIENCY OF A NEW LOW-PRESSURE TURBINE WITH ONE AND A HALF EXHAUST BASED ON FORKED TWO-TIER BLADES

A. Zariankin – V. Krutitskii

National Research University "Moscow Power Engineering Institute"
14 Krasnokazarmennaya Street, 111250, Moscow, Russia
Department of Steam and Gas Turbines
Zariankinay@mpei.ru - Krutitskiivi@gmail.com

ABSTRACT

The maximum power of single-flow steam turbines is determined by the amount of steam that can be passed through its last stage. With a fixed length of the blades of this stage, the passage of steam into the condenser can be increased by increasing the number of steam flows. Structurally, this problem is solved by using double-flow low-pressure cylinders (LPC) and increasing the number of LPCs.

It is this path that is currently being intensively used in the world turbomachinery industry. As a result, over the past decades, the blade lengths of the last stages of condensing steam turbines have increased from 1200 mm to 1500 mm.

The presented materials consider an alternative solution based on the Bauman stage. This method was used in steam turbines until the middle of the twentieth century and was rejected due to the very low efficiency of such LPCs.

It is shown that such a decision was made without a proper analysis of the reasons for the low efficiency of cylinders with Bauman stages. Elimination of these reasons will allow creating a low-pressure cylinder with one and a half exhaust steam, the efficiency of which may be higher than the efficiency of a modern low-pressure cylinder made on the basis of the last stage rotor blades with a length of 1400-1500 mm.

When developing a new low-pressure cylinder with one and a half steam exhaust, two-tier stages were considered as stages made on the basis of two-tier fork blades, which made it possible to sharply reduce losses from the fan, and nozzle diaphragm of these stages were equipped with upstream distribution grids, which ensured a uniform distribution of steam flow rates over all sections of the two-tier stages.

KEYWORDS

LOW-PRESSURE CYLINDER, BAUMANN STAGE, FORKED TWO-TIER BLADES.

NOMENCLATURE

l	blade length, [mm]
Δ	absolute overlap of the vane and blade airfoil grids, [mm]
Δ/l	relative overlap, [-]
D/l	relative diameter, [-]
φ	velocity coefficients at vane assembly, [-]
α	diffuser opening angle, [-]
n	diffuser expansion ratio, [-]
ζ	local coefficients of energy losses, [-]
c	velocity streamline, [m/s]

INTRODUCTION

Condensing steam turbines are the only energy machines, where specific volumes of working fluid (in this case steam) varies from the inlet section to the outlet in the 1000-1500 time. Accordingly, their ultimate power is determined primarily by the technical possibility of draining huge volumetric steam flows from their last stages to the subsequent condenser.

In this regard, a large clear practical interest in developing highly efficient low-pressure cylinders condensing turbines providing a pass large volumes of steam with a minimal increase their specific metal consumption. This problem is still far from being solved, since at present it is being solved by simply increasing the lengths of the blades of the last turbine stages with a simultaneous increase to two flows in one cylinder and an increase in the number of such cylinders.

Naturally, such solutions inevitably lead to a sharp increase in metal consumption, a decrease in the reliability of the blades of the LPC and the appearance of great problems with ensuring the high efficiency of the last stages with a blade length of 1400-1700 mm.

In fact, at present the possibilities of increasing the power of single-shaft steam turbines by increasing the steam flow rate into the condenser have exhausted themselves. In this regard, the problem of preserving and even increasing the achieved results in the efficiency of modern LPCs with a decrease in the values of the specific metal consumption of such cylinders becomes very urgent.

The solution to this problem is contained in the patents of Bauman (1918), Schelens (1918), Clark (1921), where a LPC with one-and-a-half steam exhaust and two-tier LPC (Bauman) is considered, as well as the shape of rotor blades for two-tier LPC (Schelens, Clark).

A fairly wide practical implementation in the first half of the 20th century was received by the two-tier Bauman stage, which provides an increase in steam flow into the condenser by 50% without increasing the length of the blades of the last stage. In Russia, the specified stage was used in turbines AK-25, AK-50, AK-100, VK-50, VK-100, VK-150, as well as in the most famous and mass turbine K-200-130 LMZ (over 150 turbines), the first version of which was produced in 1958.

Thanks to the use of one and a half exhaust in the LPC, it was possible with the length of the last stage blades equal to only 765 mm, was able to create a turbine with a capacity of 200 MW, three-cylinder design, which received the mass distribution, both in Russia and in several foreign countries.

However, as a result of subsequent industrial tests, it was found that the efficiency of the LPC with the Bauman stage turned out to be significantly lower than the LPC with a traditional blade apparatus. Since in the 60s of the 20th century rotor blades exceeding 900 mm were already created, in order to increase the efficiency of the LPC of the K-200-130 turbine, a massive modernization of the specified cylinder began with the elimination of the Bauman stage without analyzing of the reasons for their low efficiency and the possibility of eliminating these reasons.

According to the data obtained in [1], the modernization of the K-200-130 turbine with the removal of the Bauman stage from the flow path of the LPC made it possible to increase the efficiency of this turbine by 10%. Such an impressive result excluded from consideration the question of the practical use of the Bauman stage in the LPC of powerful condensing steam turbines.

As a result, the increase in the throughput of single-flow LPC is still carried out by simply increasing the length of the blades of the last stages. If in the 50s of the last century the length of the blades in the last stage of the LPC of the K-200-130 LMZ turbine was only 765 mm, now a turbine with a blade length of 1500 mm is already in operation [2].

However, despite the unconditional technological and metallurgical success, which made it possible to ensure the calculated operational reliability of such blades, there is no data in the literature on the real (and not calculated) efficiency of stages with such rotor blades.

In purely aerodynamic terms, a number of factors can be pointed out that lead to a significant increase in energy losses in the stages as the length of the rotor blades increases. Naturally, the question arises about the economic feasibility of using stages with very long blades.

If we take into account that the refusal to use Bauman stages in the LPC went without analyzing the reasons for the low efficiency of this stage in the SVK-150 LMZ and PVK-200-130 LMZ turbines, then it is quite possible, when these reasons are eliminated, to obtain an alternative solution for increasing the steam flow through the LPC without increasing the length of the blades of the last stage of a single-flow turbine.

Let us consider just such an alternative version of the LPC based on a well-developed rotor blade 1200 m long, having previously considered the disadvantages of the Bauman stage inscribed in the flow path of the K-200-130 LMZ turbine and possible ways to.

1. ANALYSIS OF THE REASONS FOR TNE LOW EFFICIENCY OF THE LPC TURBINE K-200-130 LMZ

The low-pressure cylinder of the K-200-130 LMZ turbine is double-flow with four pressure stages in one flow. A fragment of a longitudinal section of its flow path, shown in Figure 1, indicates that its flow path, together with the flow path of a diagonal radial diffuser, is a smoothly expanding channel, which provides a good opportunity for a smooth turn of the steam flow by 90 degrees with its subsequent direction into the condenser.

According to the data of [1], due to this solution, the pressure behind the upper tier of the Bauman stage was reduced to a pressure equal to the pressure in the condenser. At the same time, in the system for removing steam from the last stage, due to the implementation of the diffuser effect, the pressure was 10% lower than the pressure in the condenser. (In most turbines this pressure is 20-40% higher than the condenser pressure [3].)

Despite this circumstance, the efficiency of the cylinder under consideration with the Bauman stage turned out to be significantly lower than the efficiency of the LPC without these stages. It was this circumstance that for many years excluded the Bauman stage from the flow paths of all subsequent power turbines.

However, back in 1964 in [4] Professor M.E. Deich made an attempt to rehabilitate the Bauman stage, where the following is said about this. «The low efficiency of Bauman's two-tier stages in existing turbines is caused not by a specific feature of this stage, but by its incorrect design. So the stage does not have a partition in the nozzle apparatus, there is no proper compaction between the tiers, the profile in the upper tier is not optimal from the point of view of high speeds.

With the elimination of the noted shortcomings already now (1965), we can talk about the competitive ability of the option with a two-tier stage. This option will significantly simplify the design of turbines, reduce its cost, switch to a deeper vacuum and, ultimately, increase the efficiency of the plant».

The question of the extent to which these disadvantages reduce the efficiency of the upper tier of the Bauman stage and whether they can significantly reduce the efficiency of the entire low-pressure cylinder was discussed in detail in [5], where it was shown that all the noted disadvantages reduce the efficiency of the upper tier of the stage under consideration no more than 10%. Applied to the entire cylinder, such a decrease in the efficiency of one of its stages reduces the efficiency of the LPC by only 2-3%.

At the same time, upon careful examination of the flow path of the LPC shown in Figure 1, a number of more serious disadvantages can be added, which sharply reduce the efficiency of the entire cylinder. These disadvantages include:

1. huge overlaps (up to 40%) when steam enters the upper tier of the Bauman stage;
2. a sharp increase in the overlap when steam enters the last stage of the LPC;
3. the presence between the lower tier of the Bauman stage and the last stage of the turbine of an annular diffuser with an opening angle $\alpha = 30$ and a degree of expansion $n = 2$;
4. a sharp increase (in comparison with the lower tier) of the vapor enthalpy difference in the upper tier;
5. high energy losses from fanning in the upper tier and the last stage, which cannot be compensated for by simply increasing the chords of the profiles.

Most of these disadvantages no longer concern the Bauman stage, but the entire cylinder as a whole, and their contribution to the efficiency of the LPC is significantly greater than the contribution of the disadvantages noted in [4].

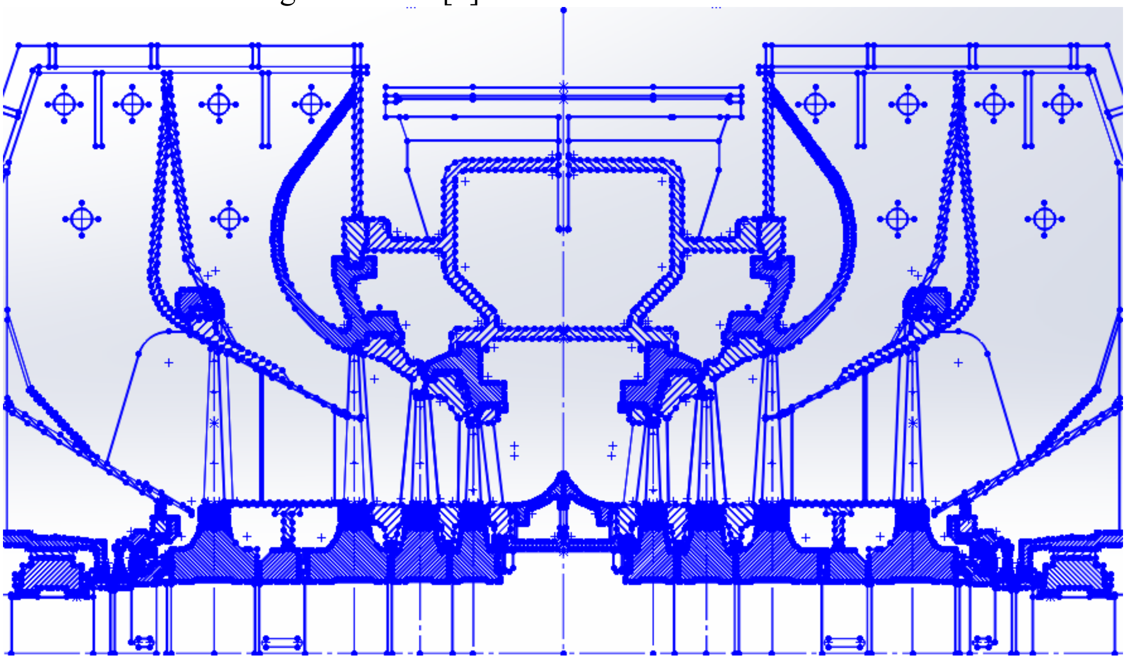

Figure 1. A fragment of the flow path of the LPC of the K-200-130 turbine.

Indeed, in the upper tier of the Bauman stage, the difference between the periphery of the 2nd stage rotor blades of the LPC under consideration and the periphery of the nozzle apparatus of the two-stage stage with respect to the length of the 2nd stage rotor blades reaches 50% with a very small axial distance between these stages.

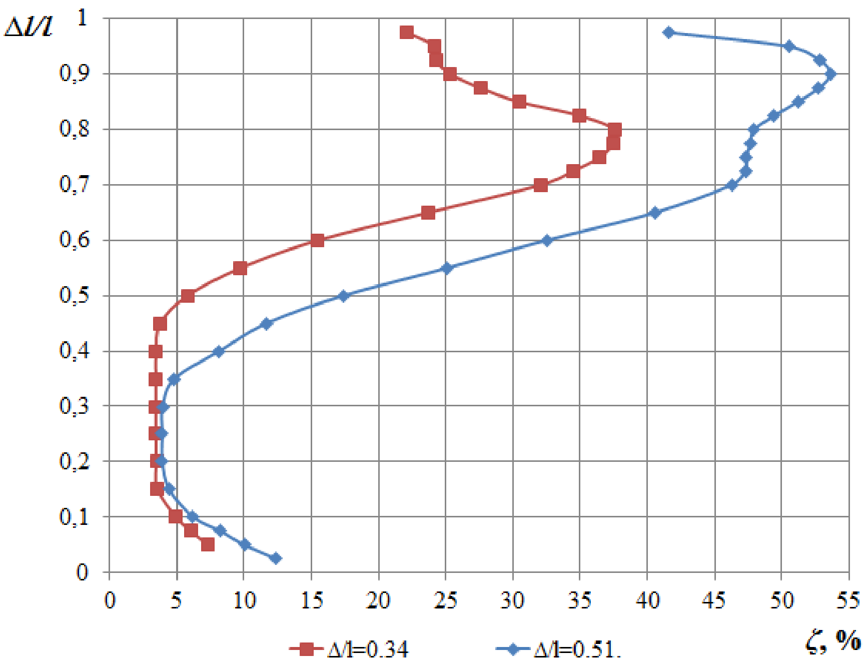

Figure 2. Distribution of local coefficients of energy losses along the height of the annular lattice of profiles with a relative diameter D / l = 2.59 [6].

The extent to which this circumstance affects the distribution of local energy loss coefficients along the height of the nozzle apparatus receiving the working fluid with such an overlap is clearly seen from the corresponding distribution of the loss coefficient ζ shown in Figure. The figure shows the distribution of this loss factor along the length of the blades at the outlet from the nozzle apparatus at 2 overlap values equal to 34% and 51%, which were obtained empirically by NTU "KhPI [6].

If in the center of the nozzle apparatus the loss coefficient is 5 - 6%, then in the periphery with an overlap of Δ = 34% it grows to 38%, and with Δ = 51% it reaches 50-53%. The result is quite natural, since when the working fluid is supplied to the lower half of the nozzle apparatus, its upper half is filled only as a result of intense radial flows arising in the interblade channels of the nozzle apparatus under consideration.

As a result, with such a supply of the working fluid, energy losses increase in all sections of the nozzle apparatus and grow catastrophically in the direction of its peripheral sections.

Compared to the flow path of the LPC without the Bauman stage, but with the same length of the last stage blades, the introduction of the specified stage into its flow path significantly increases the overlap when steam enters the last stage of the LPC.

In the case under consideration, to ensure the supply of steam along the entire length of the blades of the nozzle apparatus of the last stage, an annular diffuser was installed between the penultimate (two-tier) and last stages with an opening angle of the peripheral bypass α = 35 °. With such an opening angle of the flow path of the diffuser in its peripheral part, flow separation from its peripheral bypass occurs with the formation of an intense vortex flow generating low-frequency pulsation of flow velocities.

However, when additional resistance is installed in the outlet part of the diffuser in the form of a converging nozzle lattice, the area not occupied by the active flow is sharply reduced. The foregoing is well confirmed by the velocity field in front of the nozzle apparatus when the diffuser under consideration is installed in front of it, which we obtained as a result of mathematical modeling of the flow in this compartment of the LPC (Figure 3).

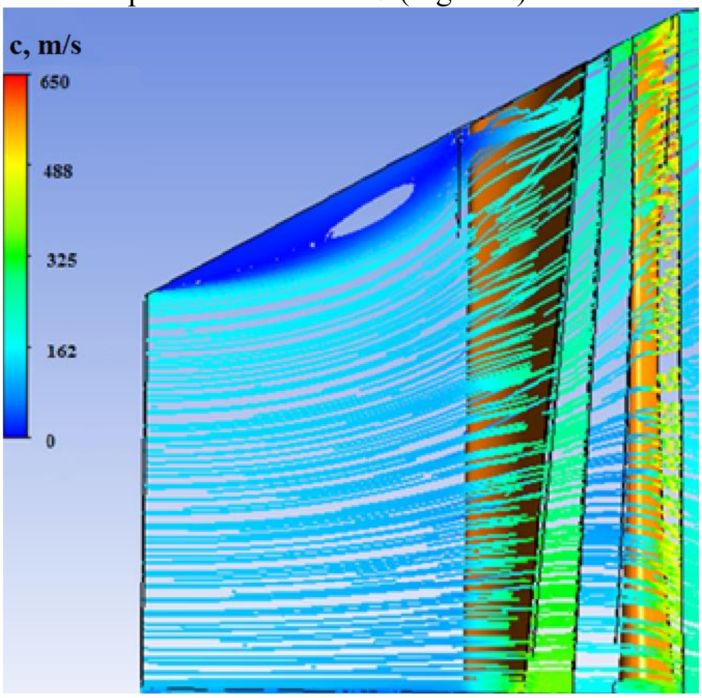

Figure 3. Estimated picture of streamlines in the last stage of the LPC.

The simulation was carried out using the ANSYS CFX software package using the k-e turbulence model. The simulation is based on numerical solution of the RANS equations.

The computational model is made in the form of a periodic segment, since the flow in the flow path of the turbine is axisymmetric. The computational model consists of a nozzle channel with an upstream diffuser, an impeller channel, and a part of an outlet diffuser.

The construction of the grid model was carried out using a specialized grid generator for turbomachines CFX - TurboGrid. The computational grid for one blade channel of the last stage includes 1.4 million hexahedral elements.

The main boundary conditions are the following: total pressure equal to 12.7 kPa and total temperature equal to 50 °C was input boundary condition, static pressure equal to 4 kPa was outlet boundary condition. Turbine rotor speed is 3000 rpm. The wet steam flow was modeled by a simplified model of ideal gas with an individual gas constant and specific heat ratio.

According to the simulation results, it was not possible to completely eliminate the separation of the flow from the external bypass of the diffuser, but the separation region decreased and remains only near the periphery of the nozzle apparatus, which actually blocks the contribution of the active flow to the nozzle apparatus. That is, the considered method failed to completely eliminate the negative influence of large overlaps on the efficiency of the last stage.

It should be borne in mind that, in contrast to the upper tier, in the last stage, the losses from fanning sharply increase. The physical essence of these losses is that at a constant (along the length of the blades) angle of the flow exit from the nozzle apparatus α, an inevitable increase in the flow area from the root to the periphery occurs. Accordingly, if, when steam is supplied to the nozzle of the last stage in the peripheral region, the flow is separated from the outer surface of the supply channel and its flow rate decreases in this area, then intense radial flows inevitably arise in the interblade channels, causing a root separation of the flow with the formation of a reverse flow of steam.

Experimentally, the presence of radial flows in stages with long rotor blades was discovered during a detailed study of steam flows in the LPC of a K-100-5 LMZ turbine back in 1967 [7], where even at full load the radial slope of the steam velocity vector behind such a stage in the root area reached 11 °.

Thus, the low efficiency of the LPC with a Bauman stage, noted in the literature, is largely due to the fact that this stage was inscribed in a low-pressure cylinder without due regard to those additional difficulties that affect essentially the entire flow path of the cylinder under consideration.

As applied to the K-200-130 LMZ turbine, work [8] shows how, with relatively low material costs, it is possible to eliminate a number of distinctive disadvantages in its LPC with a blade length in the last stage l = 765 mm without changing the blade apparatus of this turbine. This will ensure an increase in the efficiency of the LPC with one and a half exhaust up to the efficiency of traditional LPC.

Based on these decisions, we will further consider a new five-stage LPC with one-and-a-half exhaust based on the last stage rotor blade, l = 1200 mm long.

2. DESIGN FEATURES AND RESULTS OF MATHEMATICAL MODELING OF A NEW LPC WITH ONE-AND-A-HALF STEAM EXHAUST INTO THE CONDENSER

As noted in [4], the use of one-and-a-half steam exhaust in the LPC makes it possible to significantly simplify the design, reduce its cost, switch to a deeper vacuum and, ultimately, increase the efficiency of the installation.

At present, these words, spoken in 1964, become relevant in connection with the creation of an ultra-high-throughput LPC based on new rotor blades for the last stages, 1400-1700 mm long.

This way of increasing the throughput capacity of the LPC based on super-long rotor blades requires solving very complex problems associated with ensuring the reliable operation of stages with such blades and with their efficiency. Since these blades are already in operation, the problem of ensuring their reliable operation was successfully solved. As for the efficiency, there is no publicly available data on industrial tests of the LPC with the indicated blades.

In this regard, it seems expedient to return to an alternative solution of increasing the throughput of the last stages of condensing turbines by organizing one and a half steam exhaust on the basis of two-tier stages.

As shown in [8], almost all design flaws and shortcomings caused by aerodynamic factors inherent in low-pressure cylinders with one-and-a-half exhaust based on the Bauman stage can now be successfully eliminated.

Accordingly, in this case, such LPCs, both in terms of efficiency and especially in terms of specific metal consumption, can not only successfully compete with LPCs with the same steam capacity, made without a two-tier stage, but significantly surpass them in terms of the indicated indicators.

To confirm what has been said, let us consider a version of the LPC with one and a half exhaust for the K-1200-240 LMZ turbine with the last stage blade length equal to 1200 m, while eliminating the disadvantages inherent in the LPC of the K-200-130 LMZ turbine. The design of the flow path for one flow of such a cylinder is shown in Figure 4.

In one flow of the cylinder there are 5 pressure stages, of which, traditionally for a LPC with one-and-a-half exhaust, the fourth stage is two-tier. With a total length of the rotor blade equal to 1200 mm, the length of the blades in the lower tier was 745 mm, and in the upper tier, 430 mm. The thickness of the partition on the blades is 25 mm. This design will provide an exhaust area of one LPC flow equal to 16.3 m^2.

Steam removal to the condenser is carried out according to the same design scheme as for the K-200-130 LMZ turbine. This is where the similarity of the considered flow path of the new LPC with the flow path of the K-200-130 LMZ turbine ends.

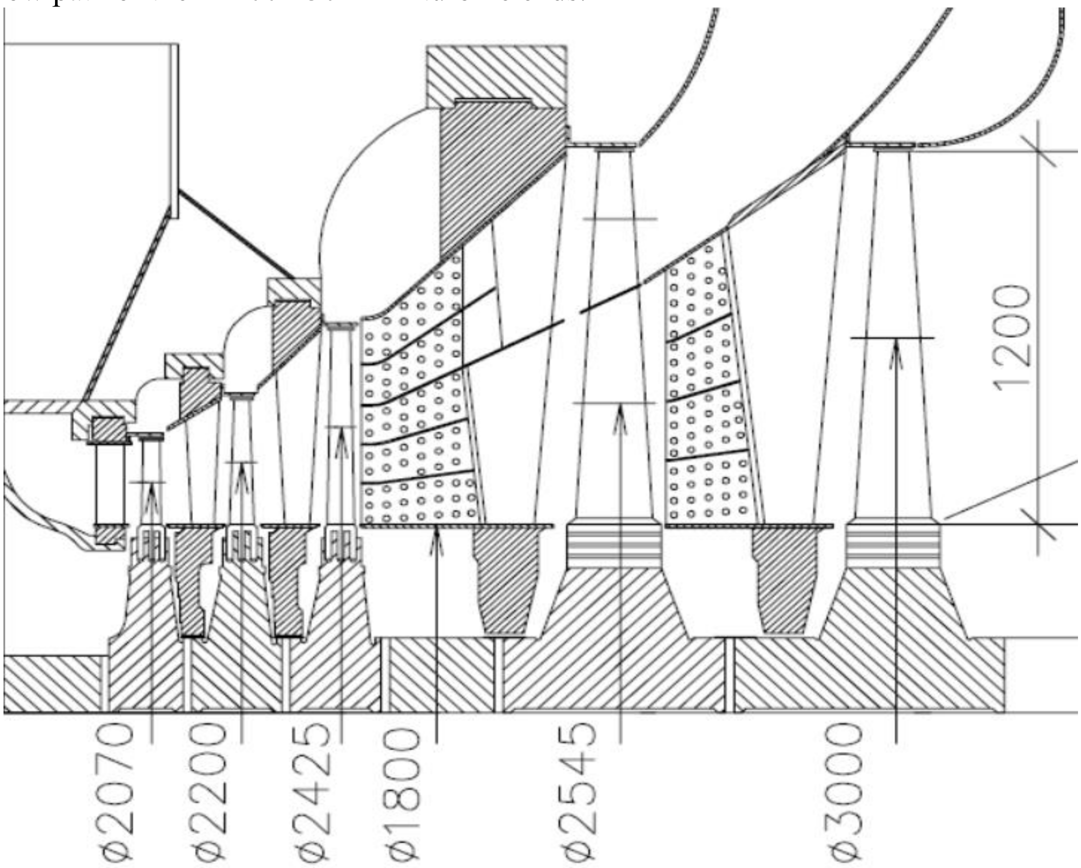

Figure 4. A fragment of the flow path of the LPC with last stage blade length of 1200 mm.

As already noted, the main disadvantage of all low-pressure cylinders of condensing steam turbines is a very high overlap between the stages, which, due to a sharp change in specific volumes of steam, continuously increase along the steam path and reach maximum values when steam enters the last stage.

When installing a two-tier stage, the overlap at the entrance to the upper tier of a two-tier stage increases by 50%, and upon entering the last stage it reaches 40%. As a result, as follows from the above figure 2, the coefficients of local losses in the periphery and at the root of the nozzle apparatus of these stages increase to 30-50%. With such losses, energy conversion in the end zones of these stages occurs with an unacceptably low efficiency.

Physically, such a cumbersome increase in the energy loss coefficients in the root zones is due to the fact that with large overlaps of the steam supply to the nozzle apparatus, their peripheral zones are outside the active flow. Since there is a sharp decrease in the flow area in the interblade channels of the nozzle apparatus in the axial direction, intense radial flows from the root to the periphery inevitably arise. At the same time, naturally, the losses at the root increase sharply.

The longer the length of the blades, the more intense the arising radial flows, since with an increase in the length of the blades at a constant angle of exit of the flow from the nozzle apparatus, the flow area of the nozzle grids increases intensively from root to top, which leads to an even greater increase in losses from large overlaps.

The extent to which large overlaps affect the velocity field when steam enters the subsequent nozzle apparatus is clearly seen from [8], which shows the results of modeling the flow in a nozzle apparatus of a two-tier stage with an overlap of 50% (Figure 5). It is clearly seen here that a developed vortex flow takes place in the peripheral region, blocking the supply of steam to the peripheral zone.

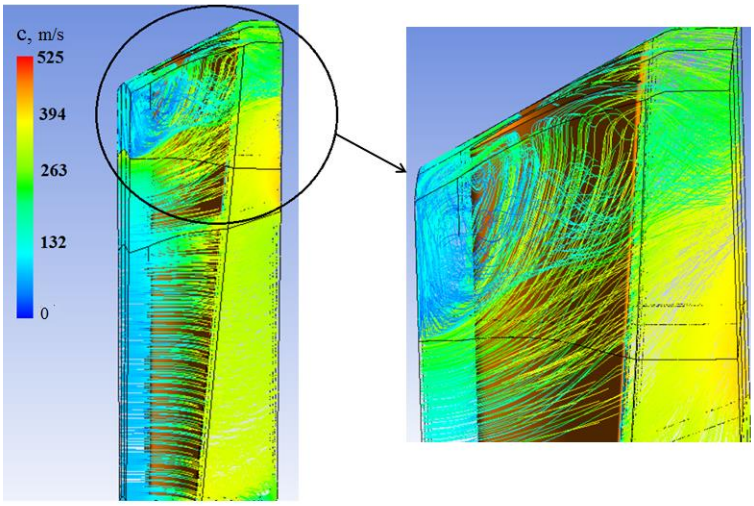

Figure 5. Calculated picture of streamlines in the nozzle apparatus of the Bauman stage of the LPC turbine K-200-130 [8].

Accordingly, the sharp drop in the local values of the velocity coefficients φ in the periphery of the nozzle apparatus and in the root region, shown in Figure 6, obtained as a result of mathematical modeling, also becomes clear.

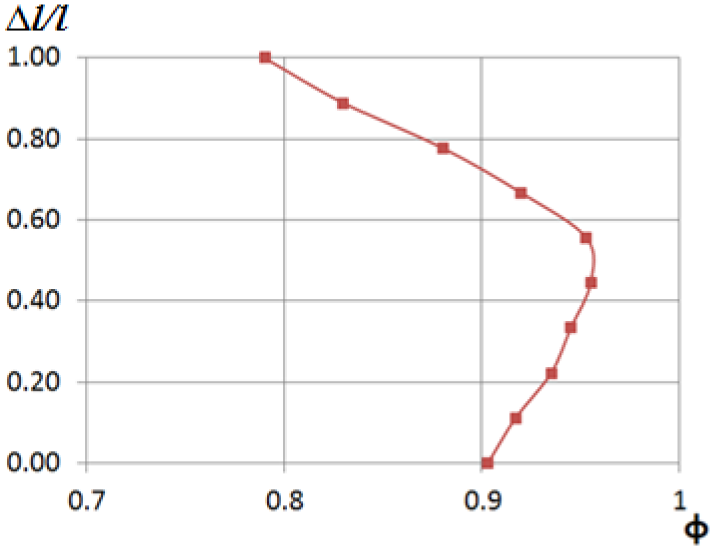

Figure 6. Distribution of the velocity coefficient φ along the height of the nozzle apparatus in the Bauman stage.

Thus, the main aerodynamic task of almost all LPCs, and especially for LPCs with one and a half exhaust, is to ensure a guaranteed supply of steam to the nozzles of the upper tier of the two-tier stage and the last stage of the turbine.

To solve this problem, the axial clearance (compared to the LPC of the K-200-130 LMZ turbine) between the third and fourth stages of the flow path, shown in Figure 4, was significantly increased. In addition, in comparison with the LPC without a two-tier stage, the axial clearance between the fourth and fifth stages was also increased.

In the axial space obtained in this way, annular distribution grids were inscribed, ensuring complete filling with steam of the nozzle apparatus of problem stages. Radial perforated ribs were used to secure the annular distribution grids.

The effectiveness of such a solution is clearly illustrated by the result obtained as a result of mathematical modeling of the velocity field in a nozzle apparatus with an upstream distribution grid of a two-tier stage with a long blade equal to 1200 mm. (Figure 7).

The simulation was carried out using the ANSYS CFX software package using the k-e turbulence model. The computational model is made in the form of a periodic segment and consists of interblade channels of the upper and lower tiers of a two-tier stage, a part of the outlet diffuser, as well as an upstream diffuser with annular distribution grids. Perforated stiffeners were not taken into account when modeling the distribution grids. The grid model was built in TurboGrid. The computational mesh for one segment of a two-tier step includes 2.7 million hexahedral elements.

The input boundary condition was a total pressure of 35 kPa and a total temperature of 90 ° C. The outlet was set to a static pressure of 4 kPa for the upper tier and 12.7 kPa for the lower tier. Turbine rotor speed is 3000 rpm

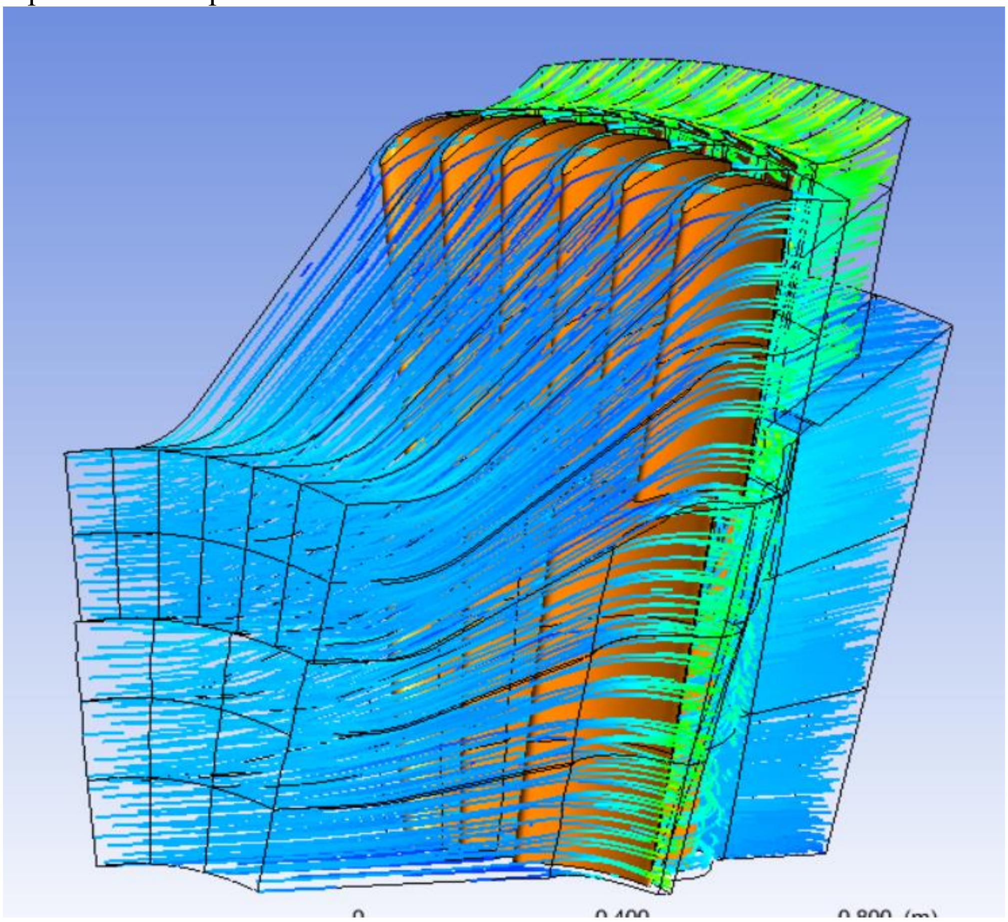

Figure 7. Streamlines in a two-tier fork stage.

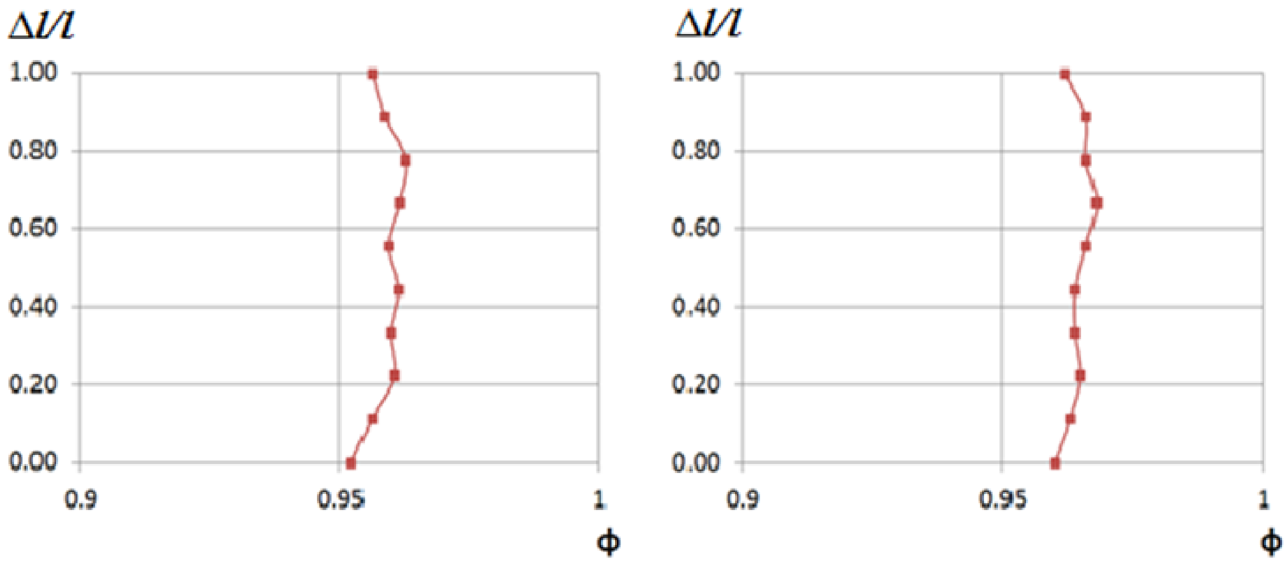

Figure 8. Distribution of velocity coefficients along the height of the nozzle apparatus in the lower (a) and upper (b) tiers.

It is clearly seen that in this case there is a complete filling along the radius of the inlet section of the nozzle apparatus in the complete absence of zones of separation of the flow from the limiting surfaces. The additional energy losses caused by the distribution grids are rather small, due to low velocities in the axial gap, and are more than compensated for by a sharp decrease in losses in the nozzle apparatus.

The distribution of local velocity coefficients shown in Figure 8 along the radius behind the nozzle apparatus indicates that in the presence of an upstream distribution grid, this coefficient hardly changes, remaining at the level of 0.96-0.965.

Thus, the proposed solution makes it possible to increase the efficiency of stages with large overlaps to the level of losses typical for stages of medium-pressure cylinders.

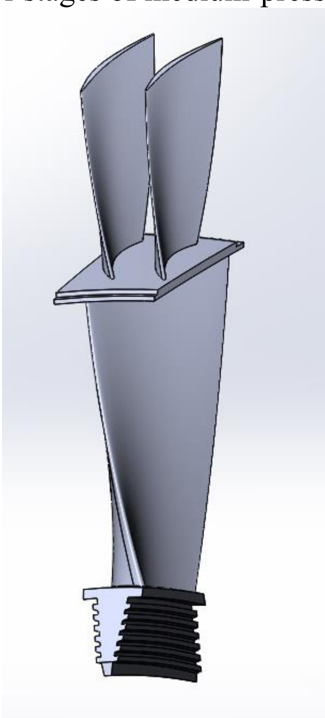

Figure 9. Model of a forked rotor blade.

In the cylinder shown in Figure 4, the entire design of the two-tier stage was fundamentally changed. Instead of the Bauman stage, a two-tier stage was used, made on the basis of a forked blade [9], shown in Figure 9.

In this case, on one feather of the lower tier there are 2 feathers of the upper tier, which, provided that the structure is non-separable, required to increase the partition between the tiers to 25 mm. Detailed strength and vibration calculations performed in [10] confirmed the possibility of reliable operation of such a blade.

A feature of this blade, in addition to a thick partition, is the possibility of making blade profiles in the upper tier with significantly smaller chords than in the lower tier. As a result, the load on the feather of the lower tier is sharply reduced and the relative length of the blade in the upper tier significantly increases.

The latter circumstance made it possible to reduce the terminal energy losses, thereby eliminating one of the distinctive disadvantages of the Bauman stage. In addition, with a total length of the fork blade of 1200 mm in the cylinder shown in Figure 4, the relative diameter was 8.65. In this case, the magnitude of fan-like losses can be neglected, and there is no need to twist the blades in the upper tier of a two-tier stage, thereby significantly simplifying the technology for manufacturing such a blade.

In addition, it should be borne in mind that slightly superheated or dry saturated steam enters the stage blade, and the enthalpy drop in the upper tier is almost 35% higher than the enthalpy drop at the last stage of the cylinder. Accordingly, in the upper tier, the main moisture loss occurs already in the interblade channels of the impeller. As a result, moisture loss in a two-tier stage is significantly less than similar losses in the last stage [11].

This circumstance not only completely compensates for the losses from steam leaks in a two-tier stage, but also allows, when using fork blades, to exceed the efficiency of the last stages of condensing turbines.

All the above considerations and the proposed solutions allow us to return to the use of classical one-and-a-half steam exhaust in low-pressure cylinders of condensing turbines in powerful and super-powerful steam turbines.

CONCLUSIONS

1. The analysis of the reasons for the refusal to use one-and-a-half exhaust on the basis of a two-tier stage in powerful condensing turbines showed that such a decision was made without a proper analysis of these reasons and, accordingly, without any attempts to eliminate them.
2. It is noted that the decisive factors that reduce the efficiency of LPC with a Bauman stage are unacceptably large overlaps between the stages and a very large fan of the last stages of condensing turbines.
3. For stages with long blades and stages with large overlaps, nozzle devices with upstream annular grids have been developed and investigated on the basis of mathematical modeling, which provide guaranteed steam supply along the entire inlet section of the nozzle devices, which made it possible to drastically reduce end losses. The distribution of the steam flow rate over the inlet section of the nozzle apparatus made it possible to drastically reduce the end energies and bring the efficiency of the LPC stage with large overlaps closer to the efficiency of stages in medium-pressure cylinder.
4. As a possible option, the flow path of the new LPC with one and a half exhaust on the basis of a new forked blade, 1200 mm long, is structurally presented, and a new nozzle unit with upstream distribution grids providing steam distribution over the entire inlet section of the nozzle unit, allowing to raise the efficiency of the LPC to a level typical of the best steam turbines with single-stage flow parts.

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