Development of a program code for calculating the flow paths of steam turbines with long blades

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Abstract. Paper is devoted to the development of software code for calculating low-pressure cylinders of steam turbines. The program code divides the steam turbine compartment with a long blades into separate stages, and also calculates these stages by sections in height. The code is based on a 2D thermal design calculation technique, supplemented by refined coefficients of turbine cascade velocities. This approach makes it possible to significantly speed up the design and selection of optimal parameters for flow paths with long blades by reducing resource intensity. Thus, the developed code makes it possible to obtain the parameters necessary for profiling the blade apparatus of the flow path of the low-pressure cylinder of a steam turbine. The paper discusses the method of dividing the compartment by heat drops and the method of calculating the stage by cross-section. The flow path of the low-pressure cylinder is profiled. A calculation fluid dynamics (CFD) modelling of the designed last stage was performed. The results of the CFD calculation showed the absence of separations in the stage profiled according to the parameters obtained in the developed program code. And a comparison of efficiency based on the results of these two calculations showed a good agreement.

1 Introduction

One of the highest priority tasks of modern power energy is increasing the efficiency of production of electrical and thermal energy. At the same time, the increase in energy consumption leads to the creation of high-power power units. The design of new steam turbines with increased power and efficiency most fully meets these challenges.

Steam moving through the flow path of the turbine increases its specific volume. A steam turbine can be divided into high-medium and low-pressure parts [1]. In the high-pressure part, a weak specific volume occurs, close to a linear law. Since the steam in this part of the turbine has a high density and a small specific volume, this makes it possible to make turbine blades of short length and a profile that is almost constant in height. In the medium pressure part, the increase in specific volume from one stage to another is more intense. This leads to a fairly strong increase in the length of the blades towards the end of the expansion process in this part. Elongation of the blades after a certain value introduces new corrections to the calculation, since we can no longer assume that the flow is uniform along the height of the blade [2]. The blade has to be twisted and changed in height in order not to lose performance

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and efficiency, since if the cross-section is left the same in height, the flow will not optimally flow onto the blade at the root and at the periphery, which ultimately leads to a decrease in the efficiency of the turbine cascade [3]. This problem is especially important for the low-pressure parts and last stages of steam turbines, since there are the largest changes in specific volume. In addition, as the length of the blade increases, the ratio of the length of the blade to the average diameter of the stage, called fanning coefficient θ, increases. This parameter has a significant impact on the flow structure in the stage. Steps with a large fanning coefficient θ are the most difficult to design [1].

When designing steam turbines, special attention is paid to low pressure cylinders (LPC) and the last stages [4]. In this area there is a large reserve for increasing the efficiency and reliability of turbine units. And the unit power of a turbine directly depends on the flow rate that its last stage is capable of passing through [5]. The main problem is the extreme complexity of the processes occurring in the LPC. Therefore, when designing a low-pressure blade apparatus, a combination of several methods is currently used to obtain the gas-dynamic characteristics of the flow in the flow path of the turbine [6]: 2D thermal calculation, CFD calculation and experiment on a full-scale test rig. The most complex and expensive method is a full-scale experiment. At the same time, it allows one to obtain reliable information about the steam flow in the last stages [7]. Carrying out CFD calculations is less resource-intensive than a full-scale experiment. This method provides a very detailed view of the steam flow. However, it requires significant computational power [8] and subsequent verification of the results [9]. Unlike experiment and CFD, 2D thermal calculation is much faster and does not require significant resources. This allows us to consider this method as the main one at the initial stages of design. With its help, it is advisable to solve the problems of determining the design of the flow path and optimizing its parameters.

Currently, 2D thermal calculations are actively used in the design of steam turbines. However, most existing software products are designed for high and medium pressure stages, which can be designed for one section and it is enough to construct a blade. For long blades of medium pressure cylinders (MPC) and LPC, the calculation of several sections in height is required for their good profiling [10]. In addition, some available software products do not allow solving the problem of designing a flow path in a complex and are a set of separate algorithms that are poorly integrated with each other. It is also worth noting that an important component of the thermal calculation, which largely determines its accuracy, is the energy loss assessment model [11].

This article discusses the development of a program code for calculating the design of the flow path of a steam turbine with long blades. The program being developed will determine the number of stages, their geometry and calculate the kinematic parameters of steam movement (angles and velocities) along sections along the height of the blades. The results of the developed program code will be verified using CFD.

2 Methodology for thermal calculation of the turbine flow path

The operation of the developed algorithm will be presented using the example of designing one low pressure flow of a steam turbine of a combined cycle plant [12]. The available heat drop at the LPC is 597 kJ/kg, steam consumption is 59 kg/s. Inlet pressure 0.3 MPa, inlet temperature 140°C. Outlet pressure 5 kPa.

2.1 Dividing of the turbine compartment by heat drops

In order to calculate the height of the steps and determine their parameters, you must first distribute the heat drop of the compartment between the steps and determine the number of steps in the most rational and optimal way.

Dividing into stages is based on the specified heat drop per compartment. It is important to correctly distribute the amount of energy to each stage. So as not to get a huge amount of them. And also to get maximum efficiency of each stage, taking into account changes in the height of the blades. Fig. 1 shows the change in steam specific volume in the LPC along the relative axial coordinate $\bar{a}$ ($\bar{a} = a_i / a_{max}$, where a_i – is the current coordinate along the LPC axis, a_{max} – is the maximum coordinate along the LPC axis). This change is associated with the rapid increase in the length of the blades and the need for their profiling and careful calculation along the high.

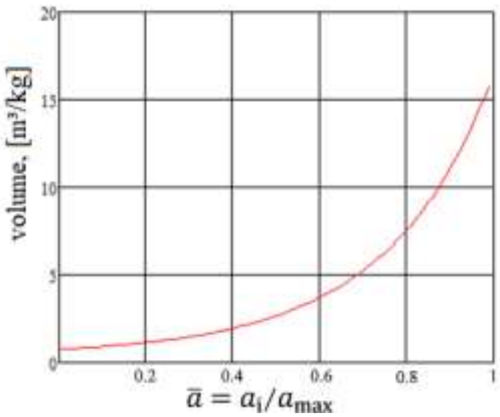

Fig. 1. Growth of specific steam volume in LPC along the axial direction.

For each stage, the heat drop is determined by the velocity coefficient, equal to the ratio of the rotor rotation velocity at the average diameter u to the isentropic exit velocity of the stage c_{is}. The optimal ratio of these velocities corresponds to the maximum efficiency of the stage blade apparatus. To determine the optimal heat drop per stage, the equation for the optimal velocity ratio is used [1]:

$$\left(\frac{u}{c_{is}}\right)_{opt} = \frac{\varphi \cdot \cos(\alpha_1)}{2 \cdot \sqrt{1-\rho_1}}, \tag{1}$$

where φ – stator blades cascade speed coefficient; α_1 – exit angle of stator blades cascade; ρ_1 – degree of reaction.

Equation (1) is valid for small and medium-sized blades. The definition of the value of this quantity for the last stages is different and looks like [1]:

$$\left(\frac{u}{c_{is}}\right)_{opt} = \frac{\varphi(1-(c_2/c_{is})^2)}{2 \cdot \cos(\alpha_1) \cdot \sqrt{1-\rho_1}}, \tag{2}$$

where c_2 – actual exit velocity of the stage in absolute motion.

Often, for the last stages, heat drop values are selected that are less than those given by (2). This is due to the need to ensure stable operation in variable modes [13].

First, the parameters of the first stage of the low-pressure compartment are determined. This can be done based on the initial parameters specified for the breakdown of heat drops. Next, according to (1), the speed ratio is calculated, according to which the heat drop for a given stage will be selected. It is calculated using equation [1]:

$$H_0 = \frac{u^2}{2 \cdot c_{is}^2}. \tag{3}$$

After that, the parameters are determined in front of the stator blades cascade and behind it to determine the stator and rotor blades height. Next, it is necessary to accept the law of change in the hub diameter of the flow path. It directly affects the streamlines being built and the final results of the heat drop dividing. Then it is necessary to estimate the height of the last blade of the compartment. This is done by using the continuity equation, where the velocity is set at the exit from the last stage. This exit speed entails losses, so it is taken to be the minimum possible based on design considerations. The specific volume for the continuity equation is taken from the end of the steam expansion process, taking into account the fact that the remaining quantities are known, the formula takes the following form:

$$l_{2z} = \frac{-d_h + \sqrt{{d_h}^2 - 4\frac{G*v}{\pi * c_{2z}}}}{2}, \tag{4}$$

where d_h – hub diameter of the flow path; c_{2z} – actual exit velocity of the last stage in absolute motion.

After determining the length of the last stage blade. It is also calculated using formulas (2) and (3) to determine the optimal heat drop. The increase in specific volume is so great that it changes exponentially, so the change in heat drop from the first to the last stage should also be made exponential for the most rational distribution and most efficient operation of all low-pressure stages:

$$H_{0_i} = H_{0_1} + e^{\frac{i}{z-1} \cdot ln(H_{0_z} - H_{0_1})}. \tag{5}$$

In this relationship, you can determine the heat drop of each stage knowing their number, which is selected as a first approximation. The choice of the number of stages can be done either in an automatic format or by allowing the engineer to choose the number of stages himself.

Fig. 2 shows a graph of what the change in heat drop obtained by this method will look like when choosing the number of stages equal to 5.

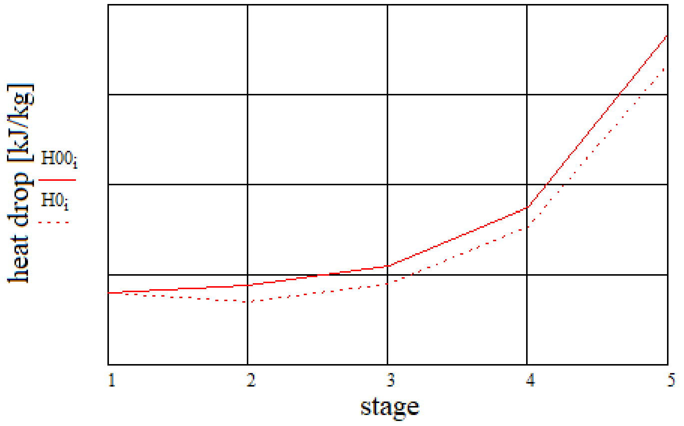

Fig. 2. Distribution of the heat drop by stages.

Next, the resulting sections are summed up, thereby determining the resulting total heat drop across all stages. This is necessary to take into account the difference between the data obtained at the LPC stage and the specified amount of heat transfer. After that, a discrepancy is determined, which is added to the obtained data to clarify the heat drop so that it exactly matches the data for the LPC. The value of the discrepancy can be calculated using formula presented in [1]:

$$\Delta H = H_{0\mathrm{LPC}}(1 + q_{\mathrm{h}}) - H_{0\mathrm{av}} \cdot z, \tag{6}$$

where $H_{0\mathrm{LPC}}$ – heat drop of the LPC; q_{h} – heat recovery coefficient; $H_{0\mathrm{av}}$ – average heat drop of the stage in the LPC.

Optimization paths for such a method lie through varying the design parameters that can be specified. Here are some implemented methods:

- Changing the average degree of reaction ρ_{av} allows you to change the optimal heat drop, thereby distributing its amount. By increasing or decreasing it, you can obtain the optimal value of the residual or the number of steps. At the same time, it is important not to forget that it cannot be made small, since this can cause a negative degree of reactivity at the root, which will cause huge losses and non-optimal flow around the working lattice.
- Changing the angle α1 also affects the parameters and directly the length of the blade. Changing it allows for a smoother flow path, which is very important when assembling the cylinder, since too strong changes in the length of the blades will lead to additional vortex losses and poor flow.
- We should also not forget about such a parameter as the output speed from the last stage, which must satisfy many requirements, for example, the exit angle, Mach. Moreover, this parameter directly affects the length of the blade of the last stage, which can make it shorter or longer depending on the task, and also affects losses with the output speed.
- The hub diameter allows you to select the heat drop per stage, the lengths of the blades and other parameters. While varying the law of change in the hub and shroud diameter, it is possible to obtain various versions of the LPC, some of which are presented in Fig. 3. This parameter and the law of its change should also be taken into account in the process of automating calculations.

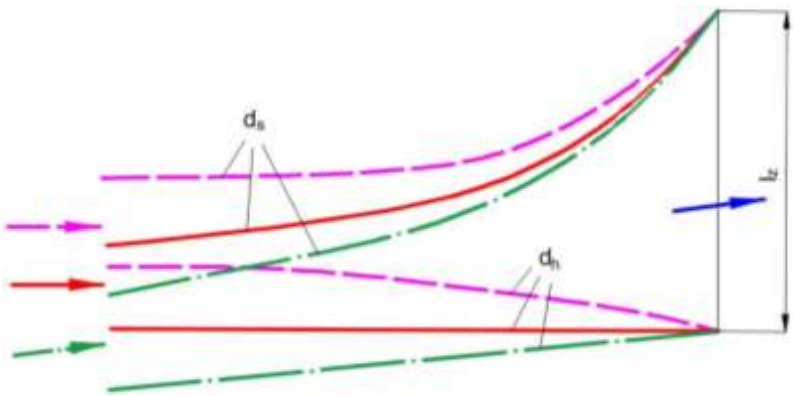

Fig. 3. Distribution of the heat drop by stages.

All of the above parameters can be set constructively, but then the resulting compartment may have a number of disadvantages. The code being developed will allow you to independently calculate them, including the heat differences described above, in order to give the most favorable and correct LPC for further design.

2.2 Calculation of turbine stages

In general, in the annular cascades of turbomachines, steam particles move along complex trajectories. The flow velocity vector is decomposed into three components: circumferential, axial and radial. Fig. 4 shows the appearance of these components in cylindrical coordinate systems [10].

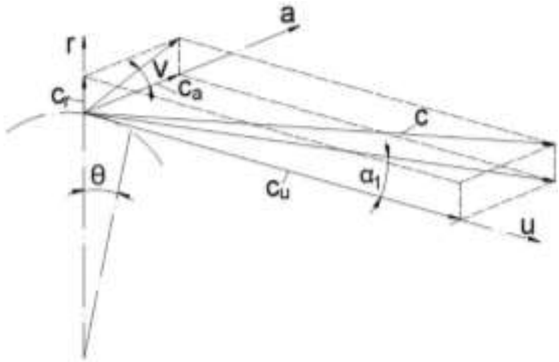

Fig. 4. Decomposition of the absolute steam velocity vector in cylindrical coordinates.

Since the flow in the blade sections is twisted and directed at an angle towards the working blades, it is affected by centrifugal components. Fig. 5 shows the streamlines of the working fluid [10]. As well as the calculated sections, in places where the program will determine the parameters.

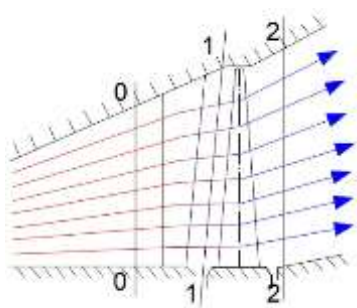

Fig. 5. Streamlines in a turbine stage with long blades.

As can be seen from the Fig. 5 the stream lines move with an inclination to the axis and are acted upon by centrifugal forces. So there will be a force in the radial direction that counteracts them in order to maintain the equilibrium of the system and the flow. This is an opposing force - a pressure gradient; in the general case, it increases along the height of the blade and is not the same for all sections of the blade. The greatest change in pressure corresponds to section I-I, between the working and nozzle blades, since in this section the flow is most twisted and has the smallest angles α. Solving the continuity equations and momentum equations in three-dimensional coordinates, in the general case we obtain the radial equilibrium equation [10]:

$$v\frac{dp}{dr} = \frac{c_{\mathrm{u}}^2}{r}. \tag{7}$$

Equation (7) can be written for each section, and with the help of which analytical problems can be solved for the turbine cascades under consideration.

Since when calculating the stages there is a strong change in the steam parameters along the height, the stage should be calculated on several sections so that in the future it is possible to recreate the profile more accurately and obtain the least losses. The program being developed allows to calculate stages for any given number of sections, which makes it very flexible.

The basis is two laws of blade twist, each of which is derived from the dependence $\alpha_1 = \alpha(r)$ and the radial equilibrium equation [1]. The most common are the laws of constant circulation:

$$c_u \cdot r = const \tag{8}$$

and constant specific flow rate:

$$\Delta G = const. \tag{9}$$

Constant specific flow law (9) works better for the last stages, and constant circulation law (8) for the first and intermediate ones.

There are a number of fundamental differences between the calculation algorithms according to this law. First of all, the calculation of the degree of reactivity differs.

For the law of constant circulation, the degree of reactivity is determined by the formula:

$$\rho_i = 1 - (1 - \rho_{av}) * (1 + \cos(\alpha_{1av})^2 (\frac{r_{av}^2}{r_i^2} - 1)), \tag{10}$$

where ρ_{av} – degree of reaction on the average diameter of the turbine stage; α_{1av} – exit angle of stator blades cascade on the average diameter of the turbine stage.

And for the law of constant specific flow rate, the degree of reactivity is defined as:

$$\rho_i = 1 - (1 - \rho_h) * \left(\frac{r_i}{\frac{d_h}{2}}\right)^{-(2 \cdot \varphi^2 * \cos^2(\alpha_1))}. \tag{11}$$

where ρ_h – degree of reaction on the hub diameter of the turbine stage.

The flow in both cases, first expands to pressure downstream the stator blades cascade P_1, increasing its velocity. And then expands to pressure downstream the rotor blades cascade P_2, converting the resulting kinetic energy of the steam into the energy of rotation of the rotor. Then the further calculation will practically not differ in any way from the choice of the swirl law. But for the law of constant circulation, it is also important to take into account the flow rate that the stator blade cascade can pass through [1]:

$$G_1 = 2\pi \int_{r_h}^{r_s} \mu_i \cdot r \cdot \frac{c_1}{v_1} \sin\left(\alpha_{1_i}\right) dr. \tag{12}$$

where μ_i – coefficient of flow rate in the section i.

The final result of the calculation is the internal efficiency, blade efficiency, specific work and various geometric characteristics of the stage in each section.

One of the most important parameters for calculating stages are the flow and velocity coefficients for the stator and rotor cascade. In modern calculations, it is customary to take these parameters to certain values and refine them only after aerodynamic calculations of the stage. Often these coefficients are assumed to be constant in height, which is neither correct nor accurate.

The code being developed uses statistical data on the distribution of these parameters over height [2]. This allows to quite accurately find the correct values of actual velocities and flow rates in stages with long blades. Fig. 6 shows the distribution of flow and velocity coefficients for the stator and rotor cascade along the high of the blade. The maximum value of these coefficients will be in the middle of the blade flow. These parameters decrease towards the ends of the blade due to the occurrence of end losses, which are greater the closer we are to the shroud or hub diameter.

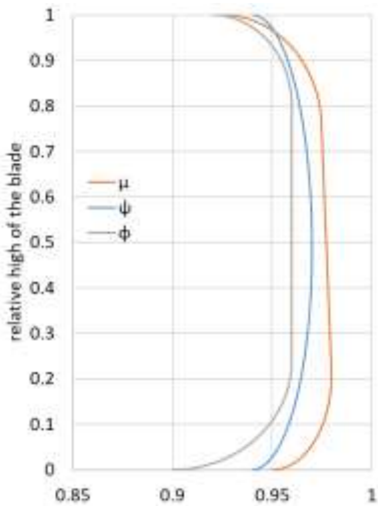

Fig. 6. Distribution of flow and velocity coefficients for the stator and rotor cascade along the high of the blade.

The dependences presented in Fig. 6 are directly related to the effects of viscosity and energy losses in the turbine stage. With their help, the quality of calculations increases significantly. The use of such functions is an exclusive feature of this code, which allows very accurate determination of all thermodynamic parameters.

3 Calculation results

After determining the main methods and formulas for the program, we will calculate the flow path of the LPC and analyze the results obtained.

According to the selected law of change in the root diameter d_h = const and formulas (1) – (6), the heat drops and main parameters of the stage was found, presented in Fig. 7.

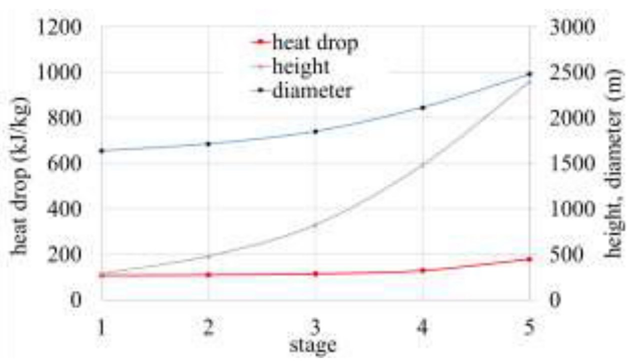

Fig. 7. Distribution of parameters by stages.

Also, during the dividing, the final value of the number of stages was determined to be 5. After determining the parameters of the stages, they were calculated using formulas (8) - (12) and all the necessary data for profiling and further aerodynamic calculations were obtained for all 5 stages of LPC. Fig. 8 shows the main required flow angle parameters for profiling a last stage. The Numeca software package was used to profile the blades in height.

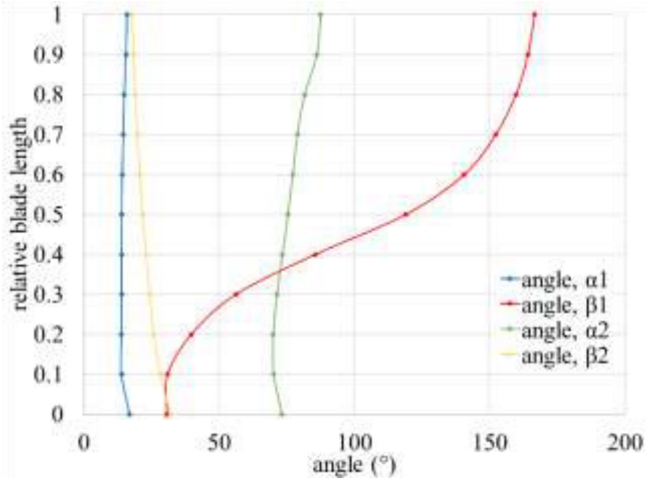

Fig. 8. Angles of flow on the last stage blades of a steam turbine.

This stage has the most unfavorable conditions for the flow, the largest heat drop and the longest blades CFD calculation of the entire profiled LPC is quite resource-intensive. In this regard, to assess the quality of the 2D thermal calculation of the flow path, CFD modelling of only the last stage of steam turbine will be carried out. Boundary conditions for the calculation are given in Table I.

Table 1. Boundary conditions.

Boundary	Boundary condition		
	Name	***Dimension***	***Value***
Inlet	Flow regime	-	Subsonic
Inlet	Total pressure	Pa	19246.2
Inlet	Total temperature	K	332.381
Inlet	Steam gas Mass Fraction	-	0.9019
Outlet	Average Static Pressure	Pa	5000
Outlet	Pres. Profile Blend	-	0

CFD calculation of the last stage was carried out in the Ansys CFX software package with an equilibrium condensation steam model and sst turbulence model. A three-dimensional picture of the steam flow was obtained, data on all sections with all parameters, which can be compared with the data obtained during the calculation using the program. Fig. 9 shows a stage with obtained stream lines. Analysis of the flow pattern allows us to say that there are no separations in the profiled stage. This indicates the possibility of using the developed software code to determine the design and calculation of the flow parts of steam turbines with long blades. Including LPC and last stages.

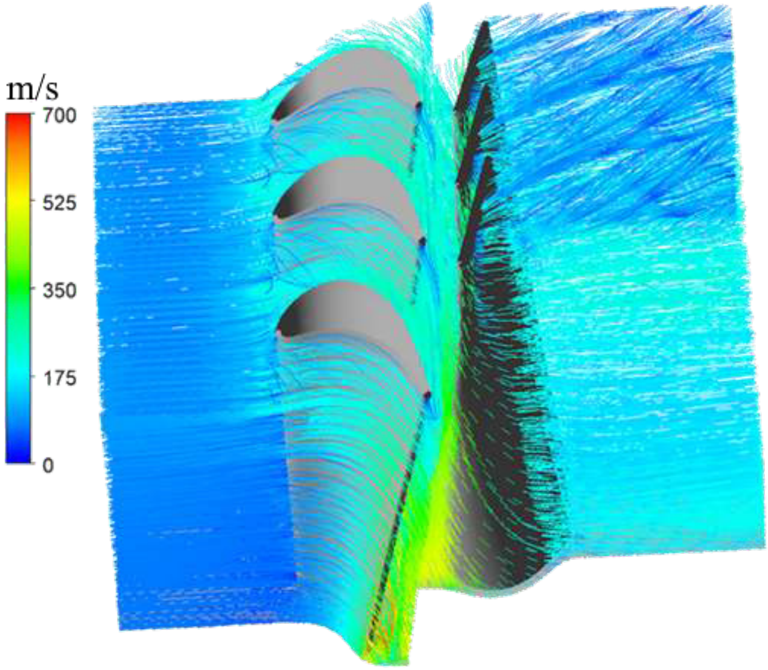

Fig. 9. Streamlines in the profiled last stage of a steam turbine.

The results of the CFD calculation were processed and compared with the results of a 2D thermal calculation of the stage performed in the developed program.

The angles of motion (Fig. 10) and velocity (Fig. 11) of the steam flow have similar values. The largest deviations are observed in the near-wall zones in the area of the hub and shroud. This is due to the presence of developed tip vortices in this region. From the overall picture, the angle of exit of the absolute velocity from the stage α_2 stands out. Here the discrepancies are somewhat greater. Especially in the hub area. This discrepancy can be explained by the fact that when profiling the stage, the supersonic nature of the flow in some sections was not taken into account [13].

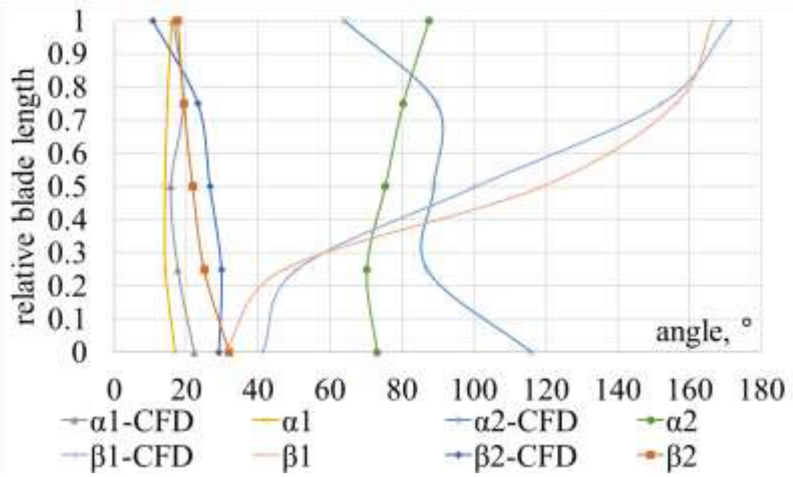

Fig. 10. Comparison of CFD and 2D thermal calculation flow angles in the last stage of a steam turbine.

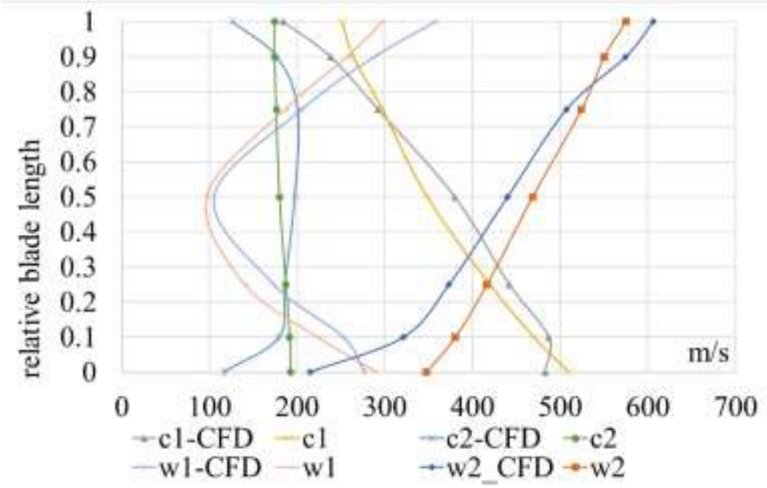

Fig. 11. Comparison of CFD and 2D thermal calculation velocities in the last stage of a steam turbine.

The relative blade efficiency values obtained in the CFD calculation and in the 2D thermal calculation have some discrepancies (Fig. 12). Especially in the bushing area. As mentioned above, this is due to the presence of a developed vortex flow in this area. And also, at supersonic speeds. Despite the discrepancies, the overall nature of the results obtained is similar.

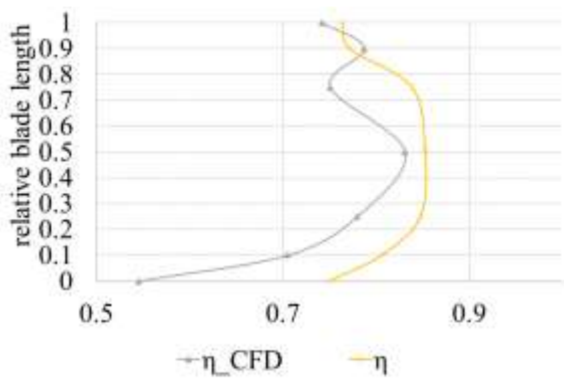

Fig. 12. Comparison of CFD and 2D thermal calculation relative blade efficiency of the last stage of a steam turbine.

4 Conclusions

A methodology for designing the flow paths of steam turbines with long blades based on 2D thermal calculations is proposed. The program code developed on its basis makes it possible to determine the design of the flow path of the steam turbine compartment. The speed of the code makes it a convenient tool for solving problems of optimizing the geometry of turbine blades.

Using the developed code, the steam turbine LPC was profiled. To assess the correctness of the calculation results, a CFD calculation of the last stage of the profiled LPC was carried out. The calculation results showed the absence of flow separations when flowing around blades profiled based on the developed methodology. The obtained values of angles and velocities gave a good agreement.

A comparison was made of the relative blade efficiency obtained as a result of CFD calculations and calculations in the developed software code. The comparison results allow to conclude that it is possible to use the developed code when assessing the efficiency of flow paths with introducing a correction factor.

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