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Computer modeling temperature and strength characteristics of an ultra-supercritical loop-type steam turbine rotor

Relevance. A global trends have emerged involving the continuous increase in demands for efficiency and reliability of power equipment, particularly ultra-supercritical steam turbines. Such turbines operate under challenging conditions of high temperatures and pressures, which can lead to significant thermomechanical stresses in rotor. Computer modeling is ideally suited to solving such problems. Therefore, research dedicated to strength calculations and determining the temperature characteristics of blades in high- and medium-pressure cylinders is highly relevant. These studies enable engineers to ensure the strength and durability of components in steam turbines.
Objective. To perform strength and temperature distribution calculations for the blades of high- and medium-pressure cylinders in a loop-type steam turbine with ultra-supercritical initial steam parameters.
Results. Data on the temperature field distribution in rotor blades of high- and medium-pressure cylinders were obtained. Using the results of the temperature calculations, the strength of the first rotor blades in the high-pressure cylinder was assessed under the influence of uneven temperature distribution and rotor rotation. Cooling of the first-stage turbine blades is achieved through convective heat exchange from the flow of cold steam from the last stage to the internal channels of the blades.
Conclusions. The results demonstrate the efficiency of the selected blade cooling method and the level of maximum stresses within the blades. One of the notable features of the operating conditions for loop-type turbine blades is the uneven heating that occurs both during transient and steady-state operating modes. Uneven heating leads to the formation of thermal stresses in the blades, which negatively affects their lifespan. Moreover, the high temperature of steam with ultra-supercritical parameters can significantly reduce the material's strength properties.

Keywords: mathematical modeling, finite element method, steam turbine, blade strength, blade cooling, computer simulation

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Introduction

Turbine blades are the most expensive and complex parts of a steam turbine. Operating under extreme conditions, these blades are subjected to significant forces and impulse effects, which cause blade vibrations and the possibility of resonance. Moreover, high temperatures in the working fluid negatively affect the material strength, creating thermally induced stresses [1]. Blade loads are divided into static and dynamic. Static loads remain constant during operation in steady-state mode and change slowly during transient states, allowing inertial effects to be neglected. These include gas-dynamic forces on the blade profile surface, centrifugal forces on rotor blades, and thermal fields [2]. Centrifugal forces cause stresses and deformations in the blades. They can also lead to blade bending and twisting. Gas-dynamic forces also cause deformations and stresses in the blade's profile section, affecting the object's position [3]. During the design of turbine blades, both static and dynamic strength calculations, as well as thermal loads, are essential. Static loads include gas-dynamic forces and non-uniform temperature fields, the latter leading to blade deformations and cracking [4]. Long-term exposure to static and dynamic loads leads to microscopic damage accumulation, resulting in the development and growth of defects, which may cause cracks and failure [5]. The physical mechanisms that determine damage are not fully understood, but numerous models exist to evaluate blade efficiency and durability [6]. Static failure and high-cycle fatigue differ in their damage accumulation mechanisms. Thus, blade design must include calculations and experiments to verify not only static strength but also cyclic durability under high-cycle fatigue [7]. Environmental program requirements for reducing harmful emissions and rising fuel costs demand significant improvements in the efficiency of steam turbines. This can be achieved by increasing the initial temperature and pressure of steam and using intermediate reheating.

Currently, achievable steam parameters are:

- Temperature: above 700°C;
- Pressure: more than 30 MPa.

The application of such ultra-supercritical parameters allows turbines to achieve an efficiency of up to 50% [8]. However, long-term operation of blade rows at these temperatures requires the use of expensive high-temperature alloys or blade cooling methods. Using working steam as a blade coolant reduces the overall efficiency of the turbine unit, but, according to estimates, the efficiency loss will be significantly smaller than the efficiency gain achieved by using steam with ultra-supercritical initial parameters [9]. Cooling systems are a standard feature in gas turbines and are becoming necessary for steam turbines as well. The most accessible method, which avoids mixing the working fluid and coolant, is internal or convective cooling [10]. This method is used to cool both working and nozzle blades [11]. Other methods, which involve injecting cold gas into the working area, significantly reduce the overall turbine efficiency and do not allow optimization of the cooling zone [12]. The convective method of blade cooling is quite flexible and effective, offering many degrees of freedom. By selecting the shape and number of channels and the parameters of the cooling gas, satisfactory temperature characteristics of the blade material can be achieved [13]. However, the local nature of cooling can cause significant temperature gradients, leading to premature wear of turbine components. Therefore, for new and improved cooling system designs, a thorough study of the thermal state using modern numerical methods is necessary [14].

1. Description of the object of research

Figures 1.1-1.2 show longitudinal sections of new variants of high and medium pressure cylinders and their additional elements. The flow parts of the high and medium pressure cylinders of the loop type are limited by the upper contour lines 1 and the lower contour lines 2.

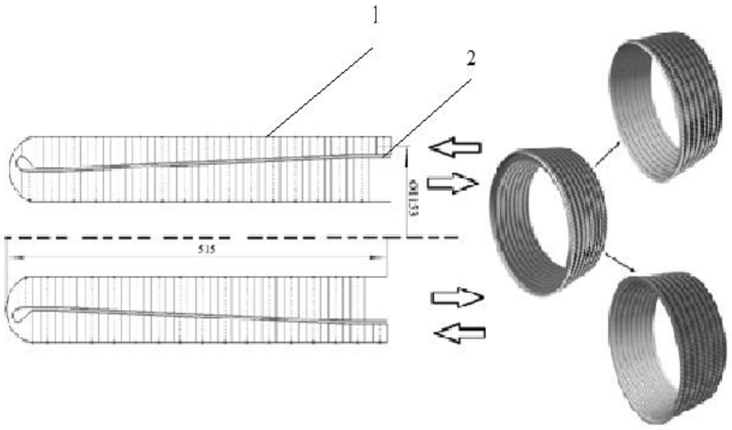


Fig. 1.1. High-pressure cylinder of a loop-type steam turbine

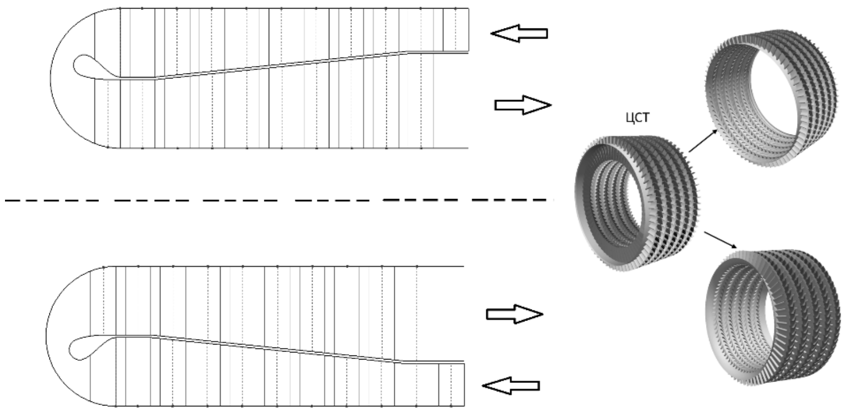


Fig. 1.2. Medium pressure cylinder of loop-type steam turbine

While the profile of the vanes is constant in all segments, the vanes of the upper and lower tiers are located asymmetrically according to the flow direction. There is no edge fastening mechanism in the blades; instead, a hard locking method is used. The geometric model of the rotor blade of the first and last stage is presented in Figure 1.3.

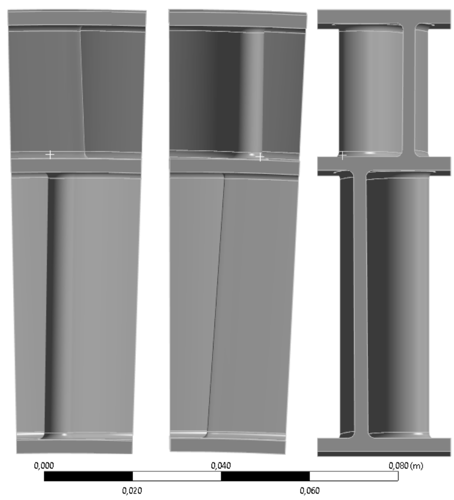


Fig. 1.3. Three-dimensional geometric model of the rotor blade of the first and last stage

2. Solution methods

2.1. Basic equations

To determine the stress-strain state of an elastic body, the system of second-order elliptic partial differential equations with respect to the displacement vector U is used

$$\mu\Delta U_j + (\lambda+\mu)\frac{\partial\vartheta}{\partial x_j} = 0, \Delta = \frac{\partial^2}{\partial x_1^2} + \frac{\partial^2}{\partial x_2^2} + \frac{\partial^2}{\partial x_3^2}, \vartheta = \mathrm{div}\mathbf{U}, \quad (1)$$

where λ and μ are the Lame coefficients.

Let us introduce the differential surface tension operator from the classical theory of elasticity as follows [22]

$$\mathbf{T}^{\mathbf{n}(\mathbf{x})}\mathbf{U} = 2\mu\frac{\partial\mathbf{u}}{\partial\mathbf{n}} + \lambda\mathbf{n}\mathrm{div}\mathbf{U} + \mu(\mathbf{n}\times\mathrm{rot}\mathbf{U}),$$

where **n** is outward unit normal to the surface.

The following boundary value problem is formulated for an unbounded three-dimensional body weakened by a system of cuts $S = \bigcup_{k=1}^{n} S_k$, to determine U:

$$\Delta \mathrm{U_j} + (\lambda+\mu)\frac{\partial\vartheta}{\partial x_j} = 0,\ j=1,2,3,\ \mathbf{T}^{\mathbf{n}(\mathbf{x})}\mathbf{U}(\mathbf{x}) = \mathbf{X}(\mathbf{x}),\ \mathbf{x}\in S. \quad (2)$$

Here **X**(**x**) is the given forces.

For the numerical analysis of the shell structure using ANSYS Workbench, homogeneous volumetric twenty-nodal finite elements of the SOLID186 type [15] were selected.

The SOLID186 element is a universal three-dimensional element (Fig. 2.1); quadratic approximation of displacements is used here. The element is determined by twenty nodes having three degrees of freedom: movement in the direction of the OX, OU, and OZ axes.

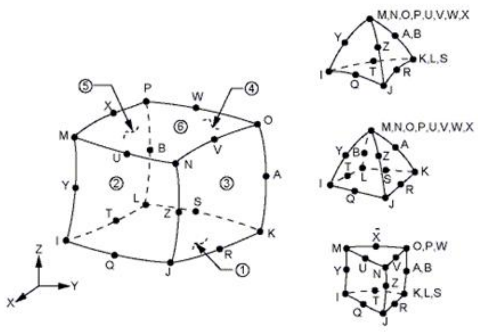


Fig. 2.1. Three-dimensional finite element SOLID186

Equations for the finite element are the equations of the theory of elasticity.

The advantages of the specified element are due to the fact that it can have an arbitrary spatial orientation, be specified in the form of a tetrahedron, a pyramid and a prism. This element allows you to model quite general geometric features of this structure, for example, sharp changes in thickness, the presence of corner points, etc., take into account plasticity, creep, hyperelasticity, strengthening, geometric nonlinearity [18]. Use of the selected element also allows you to take into account the anisotropy of the material, residual stresses and temperature loads. A finite element mathematical model of the rotor blade of the first and last stage was created, containing more than 1 million finite elements with thickenings in places of expected local stress maxima and zones of sharp changes in geometric parameters. Part of this model is shown in Fig. 2.2.

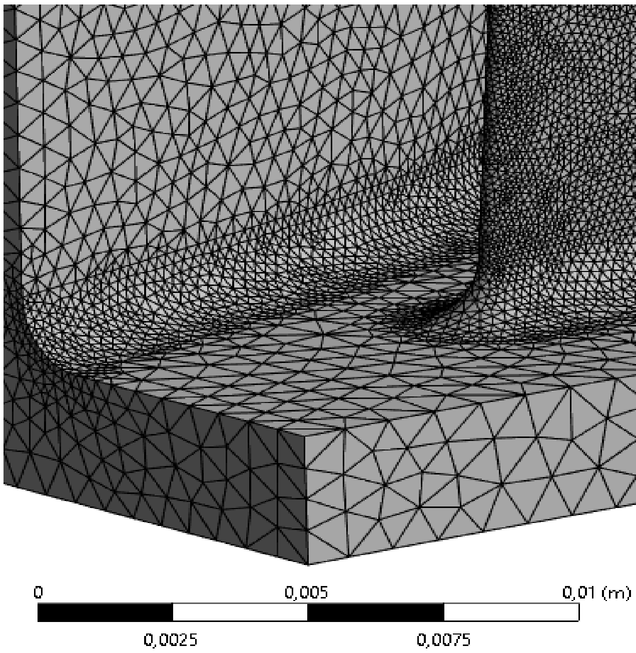


Fig. 2.2. Finite element model of the scapula in the fillet region

In general, the surfaces of the blades were thickened, especially around the leading edge, where the smallest size of the finite element was 0.075 of the base size, as well as the surface of the fillet rounding, where the size of the finite element was 0.05 of the base size.

In addition, as a result of additional test calculations, the quality of the finite element model was checked. The calculations showed that when the number of finite elements changes by 50 thousand, the change in the measurement result does not exceed 0.1%.

Thus, the choice was made precisely on the model described above, which allows for high-precision calculations and at the same time limits the requirements for the computing power of the equipment.

Modeling of the temperature state of the rotor blades of the first and last stages of the loop turbine was performed using the ANSYS Fluent Academic Edition R19.3 software package. The computational domain includes one rotor channel of the first and last stages of the high and medium pressure turbine and consists of four subdomains: the first stage rotor channel, the last stage rotor channel, the first stage blade cooling channel and the blade body. Numerical flow simulation was performed using a pressure-based algorithm that solves the Navier-Stokes equation [15], supplemented by the two-parameter Menter SST turbulence model [16], and the equation of state for water vapor according to the Peng-Robinson method [17]. The temperature distribution on the blades was simulated by solving the coupled problem of heat conduction and aerodynamics [15].

3. Numerical simulation of the temperature state of the blades

3.1 Simulation of the temperature state of the blades of the first rotor of the high-pressure cylinder

Tables 3.1 and 3.2 show the boundary conditions for the computational domains. The blade surfaces adjacent to the flow were assumed to be temperature and heat flux equal, while the others were assumed to be adiabatic. The mass flow rate for the cooling channels was determined to be 2% of the mass flow rate of the main flow in the last stage cylinder.

Table 3.1. Boundary conditions for blade channels

	1st stage rotor	16th stage rotor
Total pressure at the inlet, MPa	34,9	11,1
Total temperature at the inlet, K	975,0	760,3
Outlet pressure, MPa	32,9	9,7

Table 3.2. Boundary conditions for the cooling channel

Mass flow, kg/s	4,4
Total temperature at the inlet, K	760,3
Outlet pressure, MPa	9,7

On Figure 3.1, the temperature distribution on the blade surface is shown, while Figure 3.2 depicts the temperature isolines in the mid-cross-section of the first-stage blade, including the cooling channels. Figure 3.3 presents the isolines of the relative velocity of the main flow and the flow within the cooling channels in the mid-cross-section of the first-stage blade. Figure 3.4 illustrates the temperature distribution on the surface of the first-stage blade along the axial direction.

According to the distribution, the average surface temperature of the blade is 912.8 K, which is 62.2 K lower than the overall steam flow temperature.

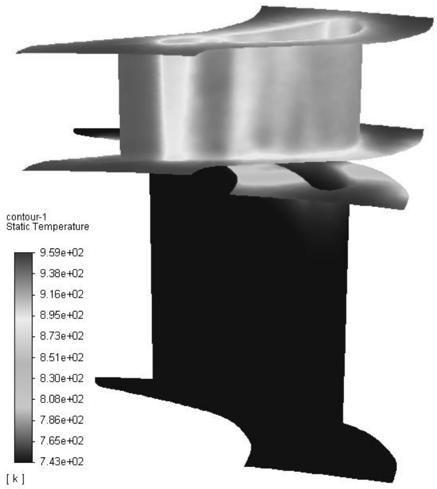


Fig. 3.1. *Temperature distribution on the surface of the blades*

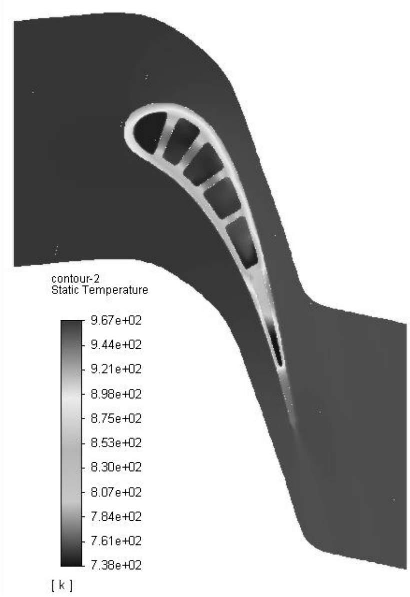


Fig. 3.2. *Temperature isolines in the middle cross-section of the blade of the first stage*

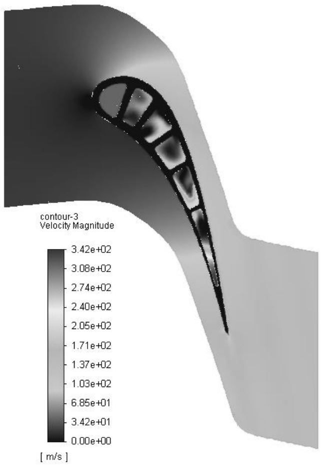


Fig. 3.3. *Relative velocity isolines in the middle cross-section of the blade of the first stage*

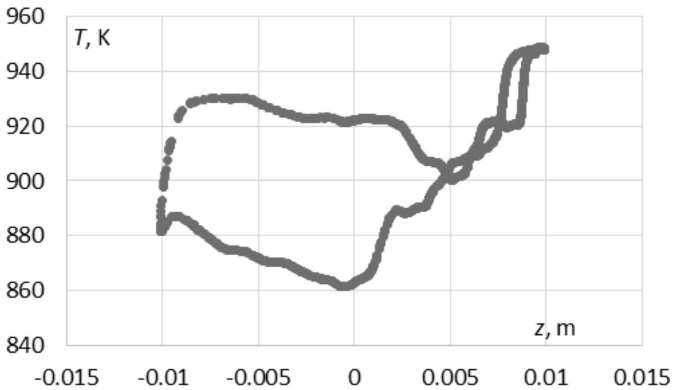


Fig. 3.4. *Temperature distribution on the blade surface of the first stage in the axial direction*

3.2 Modeling temperature state of blades of the first rotor of medium pressure cylinder

The thermal state modeling of the rotor blades in the first and last stages of the medium-pressure cylinder of the reheat turbine was performed similarly to the high-pressure cylinder [17].

Tables 3.3 and 3.4 present the boundary conditions for the calculation domains. On the surfaces of the blades in contact with the flow, conditions of temperature and heat flux equality were applied, while adiabatic conditions were set for other surfaces. The mass flow rate for the cooling channels was determined as 2% of the mass flow rate of the main flow in the last stage of the cylinder.

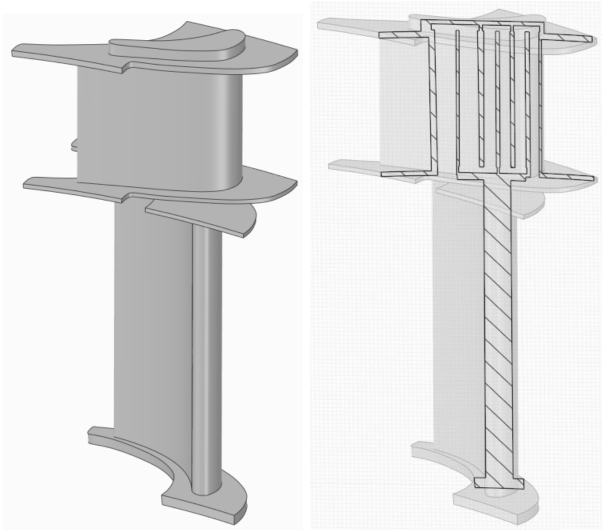
Table 3.3. Boundary conditions for blade channels

	1st stage rotor	10th stage rotor
Total pressure at the inlet, MPa	8,54	3,96
Total temperature at the inlet, K	973,2	834,0
Outlet pressure, MPa	7,88	3,53

Table 3.4. Boundary conditions for the cooling channel

Mass flow, kg/s	4,4
Total temperature at the inlet, K	834,0
Outlet pressure, MPa	3,53

Figure 3.5 shows the external appearance of the modeled blades and the configuration of the cooling channels within the first-stage blade.



a) b)

a – appearance of the blades; *b – configuration of cooling channels*

Fig. 3.5. *Appearance of blades, configuration of cooling channels*

Figure 3.6 shows the temperature distribution on the blade surface, while Figure 3.7 displays the temperature isolines in the midsection of the first-stage blade, which also includes cooling channels. Figure 3.8 presents the isolines of the relative velocity of the main flow and the flow within the cooling channels in the midsection of the first-stage blade. Figure 3.9 illustrates the temperature distribution along the axial direction of the first-stage blade.

According to the distribution, the average surface temperature of the blade is 932.8 K, which is 40.4 K lower than the total temperature of the steam flow.

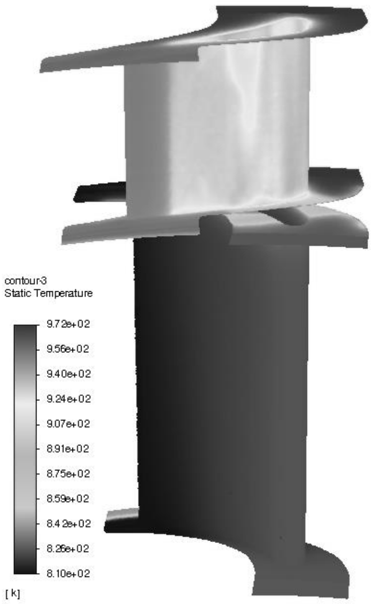


Fig. 3.6. *Temperature distribution on the surface of the blades*

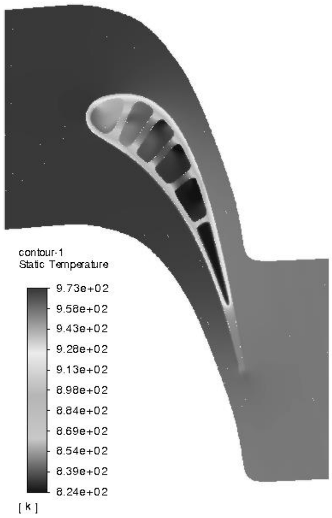


Fig. 3.7. Isolines of temperature in the middle cross-section of the blade of the first stage

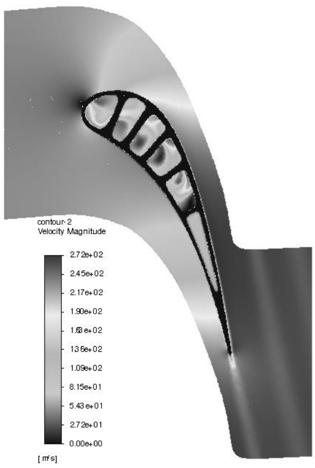


Fig. 3.8. Isolines of relative velocity in the middle section of the blade of the first stage

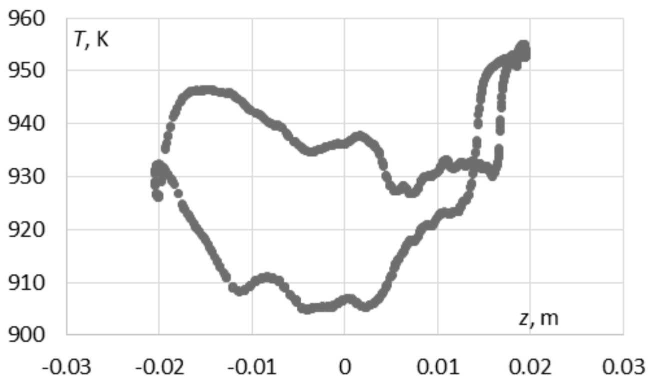


Fig. 3.9. Temperature distribution on the blade surface of the first stage in the axial direction

4. Strength Calculation of the First Rotor Blades and Analysis of Results

The dominant stresses in the rotor arise under the influence of centrifugal forces at a rotational speed of 3000 rpm. These stresses also depend on the density of the material used, with higher density causing higher stresses. As a result of preliminary calculations, a material with a density of 8160 kg/m³ was selected. The stress-strain state of the blades was determined based on calculations and is shown in Figure 4.1 as a distribution of equivalent stresses according to the von Mises criterion.

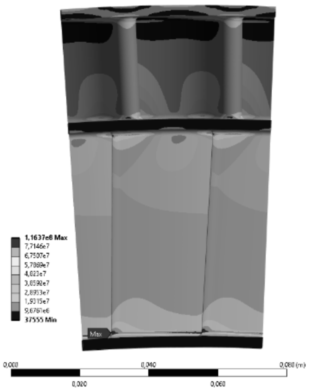


Fig. 4.1 – Distribution of equivalent stresses in the blades according to the Mises?

Global maximum stress (Global) is 116.4 MPa, it is located almost near the attachment point at the transition point of the fillet rounding into the main part of the blade at the exit from the crown, Fig. 4.2, it is the most highly loaded part of the rotor.

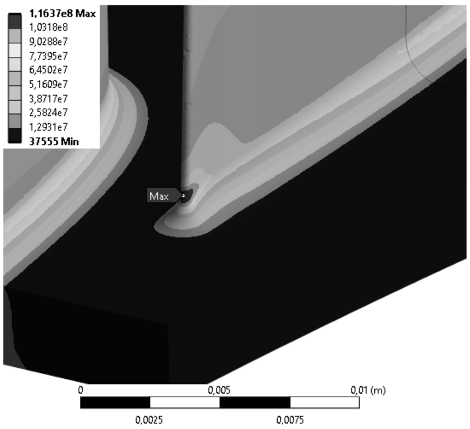


Fig. 4.2 – Localization of the global stress maximum

A similar location is observed for the next, localized stress maximum (L1), which is 98.8 MPa and is also situated near the base of the blade, but only at the crown inlet, as shown in Fig. 4.3. The Global and L1 maxima are interconnected and depend on the overall balance of the crown.

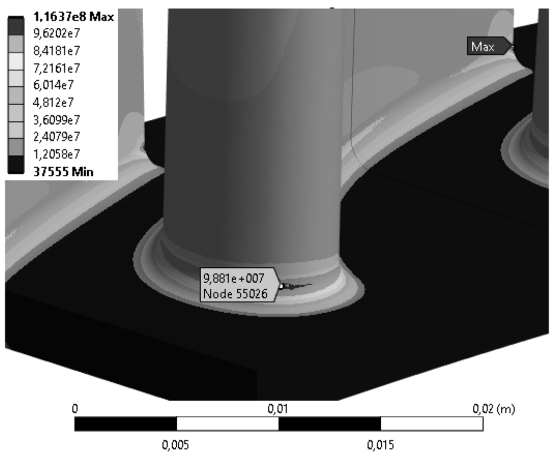


Fig. 4.3 – Localization of the L1 stress maximum

This distribution includes the main maximum stresses in the rotor as well as its local maxima. Additionally, due to the significant similarity of all the cylinder rotors, they will also contain these maxima with minor numerical differences. The global maximum stress (Global) is 116.4 MPa, located near the attachment point at the transition from the curvature of the blade fillet to the main part of the blade at the rotor outlet.

Based on the obtained data, a summary Table 4.1 was created, containing the calculation results for the identified stress maxima.

Table 4.1. Maximum stresses of the first rotor

	Global	L1	L2	L3	L4	L5	L6
Values, MPa	116,4	98,8	102,6	90,4	72,3	69,1	66,3
Temperature, °C	464	464	464	464	695	695	695

Conclusion

Promising design of a new high-pressure cylinder (HPC) and additional stages of the intermediate-pressure cylinder (IPC) for the K-300 series loop-type turbine with two-tier configuration has been presented [19]. The new flow path is designed to fit within the dimensions of the existing turbine. Number of innovative solutions, some of which are unprecedented in the global steam turbine industry, were employed in developing the new flow path, including a special meridional contour shape. The design was executed using a comprehensive methodology implemented in the IPMFlow and Ansys Workbench software packages. This methodology incorporates gas-dynamic calculations of varying complexity levels, as well as methods for constructing the spatial shape of blade paths based on a limited number of parameterized values [21].

Numerical study was conducted on the thermal and stress-strain state of the rotor blades in the high-pressure and intermediate-pressure cylinders of a state-of-the-art loop-type steam turbine with ultra-supercritical steam parameters. Strength calculations were performed to determine the stress distribution in the rotor blades, with the results indicating a sufficiently low level of maximum stresses. Numerical modeling of the thermal state of the first rotor blades in the high-pressure and intermediate-pressure cylinders of the loop turbine was also carried out. Cooling of the first-stage rotor blades was achieved by extracting steam from the last-stage channel. It was determined that using 2% of the steam mass flow for cooling reduces the average surface temperature of the first-stage blade by 62.2 K [20] for the high-pressure cylinder and by 40.4 K for the intermediate-pressure cylinder.

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Комп'ютерне моделювання температурних та міцнісних характеристик ротору ультрасуперкритичної парової турбіни петельного типу

Актуальність. У світі з'явилась тенденція, пов'язана з постійним зростанням вимог до ефективності та надійності енергетичного обладнання, зокрема ультрасуперкритичних парових турбін. Такі турбіни працюють у складних умовах високих температур і тисків, що може викликати значні термомеханічні напруження у роторі. Комп'ютерне моделювання ідеально підходить для розв'язку подібних задач. Тому є актуальним дослідження, присвячено розрахункам на міцність та визначення температурних характеристик лопаток у циліндрах високого та середнього тиску. За допомогою даних задач інженери можуть гарантувати міцність та довговічність вузлів у парових турбінах.
Мета. Провести розрахунки на міцність та розподіл температури лопаток для циліндрів високого та середнього тиску парової турбіни петельного типу із ультра-суперкритичними початковими параметрами пари.
Результати. Були отримані дані по розподіленню температурних полів у лопатках ротора циліндру високого та середнього тиску. Використовуючи результати температурного розрахунку, було отримано оцінку міцності лопаток першого ротору циліндру високого тиску під впливом нерівномірності розподілу температури і обертання ротору. Охолодження лопаток першого ступеня турбіни досягається завдяки конвективному теплообміну від потоку холодної пари з останнього ступеня до внутрішніх каналів лопаток.
исновки. За результатами зроблено висновок, про ефективність обраного способу охолодження лопаток та рівень максимальних напружень всередині лопаток. Однією з особливостей умов роботи лопаток турбіни петельного типу є нерівномірне нагрівання, яке відбувається як у перехідних, так і у стаціонарних режимах роботи лопаток. Нерівномірне нагрівання призводить до виникнення температурних напружень в лопатках, що негативно впливає на їхній ресурс. Крім того, висока температура пари з ультра-суперкритичними параметрами може значно знизити міцнісні властивості матеріалу.

Ключові слова: *математичне моделювання, метод скінченних елементів, парова турбіна, міцність лопаток, охолодження лопаток*

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