DOI: https://doi.org/10.15407/pmach2023.03.015

UDC 621.165.62

CHANGES IN THE THERMAL AND STRESS-STRAIN STATE OF THE HPC ROTOR OF A POWERFUL NPP TURBINE AFTER THE BLADES DAMAGE

[1] Olha Yu. Chernousenko
chernousenko20a@gmail.com
ORCID: 0000-0002-1427-8068

[1] Vitalii A. Peshko
vapeshko@gmail.com
ORCID: 0000-0003-0610-1403

[2] Oleksandr P. Usatyi
alpaus@ukr.net
ORCID: 0000-0002-8568-5007

[1] National Technical University of Ukraine "Igor Sikorsky Kyiv Polytechnic Institute",
37, Peremohy ave., Kyiv, 03056, Ukraine

[2] National Technical University "Kharkiv Polytechnic Institute"
2, Kyrpychova str., Kharkiv, 61002, Ukraine

In practice, during the operation of steam turbines, accidental damage to the blades of the rotors and stators of powerful steam turbines occurs. The main causes of emergency stops of steam turbines were vibration fatigue of the blades material, erosive damage to the blades body, and resonance problems during the power equipment operation. Based on this study, the assessment of changes in the thermal and stress-strain state of power equipment elements, which at nuclear power plants significantly affect the continued operation of the turbine after its damage, are quite relevant. Changes in the thermal and stress-strain state, which may occur after damage to the rotor of high-pressure cylinder (HPC rotor) of the K-1000-60/3000 turbine power unit of the LMZ in the station conditions, have been considered and analyzed and will provide an opportunity to assess the individual resource and continue the power unit operation. In the calculated assessment of changes in the thermal and stress-strain state of the HPC rotor, taking into account the data of the technical audit regarding damage, a geometric model of the rotor was created. Studies were conducted for three options of designs: the original option (five stages of the HPC rotor), the option without the blades of the last stage and the option without the fifth stage (with four first stages). For the project design, when working at the nominal parameters of the steam, the most stressed areas are the unloading holes of the 5th stage (σ_i=202.8 MPa), axial hole of the rotor in the area of the 5th stage (σ_i=195.2 MPa), as well as the 5th-degree welding fillet from the side of the end seals (σ_i=200.3 MPa) and unloading holes of the 4th and 3rd stages with a stress intensity of about 170–185 MPa. The high values of the stress intensity in the area of the 5th stage can be explained by the significant concentration of the mass of both the stage itself and its blades, which provoke significant centrifugal forces when working at the nominal rotation frequency. For a HPC rotor without blades of the 5th stage, there is a shift of the maximum stress intensity to the area of the unloading holes of the 4th and 3rd stages, as well as the axial hole of the shaft under the same stages. The maximum stress value is $\sigma_{i\,max}$=184.8 MPa. At the same time, the intensity of stresses in the area of unloading holes of the 5th degree decreased almost by half, to the level of 124 MPa.

***Keywords**: nuclear power plant, K-1000-60/3000 steam turbine, rotor of high-pressure cylinder, power, pressure, temperature, loss, equipment resource, unsteady thermal conductivity, thermal state, stress-strain state, low-cycle fatigue, long-term strength, residual resource, permissible number of start-ups.*

Introduction

During the operation of steam turbines, accidental damage to the blades of the rotors and stators of powerful steam turbines often occurs [1]. The main reasons for emergency stops of steam turbines were vibration fatigue of the blades material, erosive damage to the blades body, and resonance problems during the power equipment operation, which are thoroughly discussed in [2–4]. The given physical values of vibration loads on the blades are caused by the low load of the turbine stage, the low vacuum behind the last stage, the presence of small mass flow rates of steam at the start-up operation modes and circulation flows, zones of different pressure, etc. Because of the above-mentioned factors, vibrational stresses, excitation forces of the blade feather might occur, which will lead to destruction. Real damage can occur in the process of simultaneous action of erosive damage of the blade body from moisture, cavitation, and the interaction between Coriolis centrifugal forces on the blade surface [4].

Works related to the assessment of changes in the thermal and stress-strain state of power equipment elements, which have a significant impact in the conditions of a nuclear power plant on the continuation of turbine operation after its damage, deserve special attention. The paper examines and analyzes the changes in the thermal and stress-strain state that may occur after damage to the HPC rotor of the K-1000-60/3000 turbine power unit of the LMZ in the station conditions and will allow to assess the individual resource and prospects for continued operation of the power unit.

For each specific case of emergency damage to the rotor blades, the turbine type and its purpose, the strength and location of the blades damage, the repair capabilities of power plants and other important information must be taken into account.

Research purpose

The purpose of the paper is to analyze and evaluate the thermal and stress-strain state of the HPC rotor of the K-1000-60/3000 turbine power unit of the LMZ to extend its operation under the conditions of the stressed state of the power system. To achieve this goal, an appropriate technique was developed, a mathematical model of the thermal and stress-strain state of the HPC rotor of powerful steam turbine was improved, and relevant research was carried out.

Main material

After the accidental damage to the blades of the last stage of the HPC rotor of the K-1000-60/3000 turbine power unit of the LMZ, there was a need to assess changes in the thermal and stress-strain state of the HPC rotor. In the process of achieving the set goal, studies were carried out for three designs options: the original option (five stages of the HPC rotor), the option without the blades of the last stage and the option without the fifth stage (with four first stages).

Operating modes, technical audit and geometric model of the HPC rotor of the K-1000-60/3000 steam turbine

The start-up modes of the NPP power unit are differentiated depending on the temperature of the outer surface of the HPC flange in the steam start-up area. According to the operating instructions for the K-1000-60/3000 steam turbine of the NPP power unit, it is possible to distinguish 3 main start-up operating modes [1]:

a) start-up from a cold state at the temperature of the metal of the outer surface of the HPC flange in the steam inlet zone $T^{out}_{fl\ HPC}$<100 °C;

b) start-up from uncooled state at temperature $T^{out}_{fl\ HPC}$=100–150 °C;

c) start-up from hot state at temperature $T^{out}_{fl\ HPC}$>150 °C.

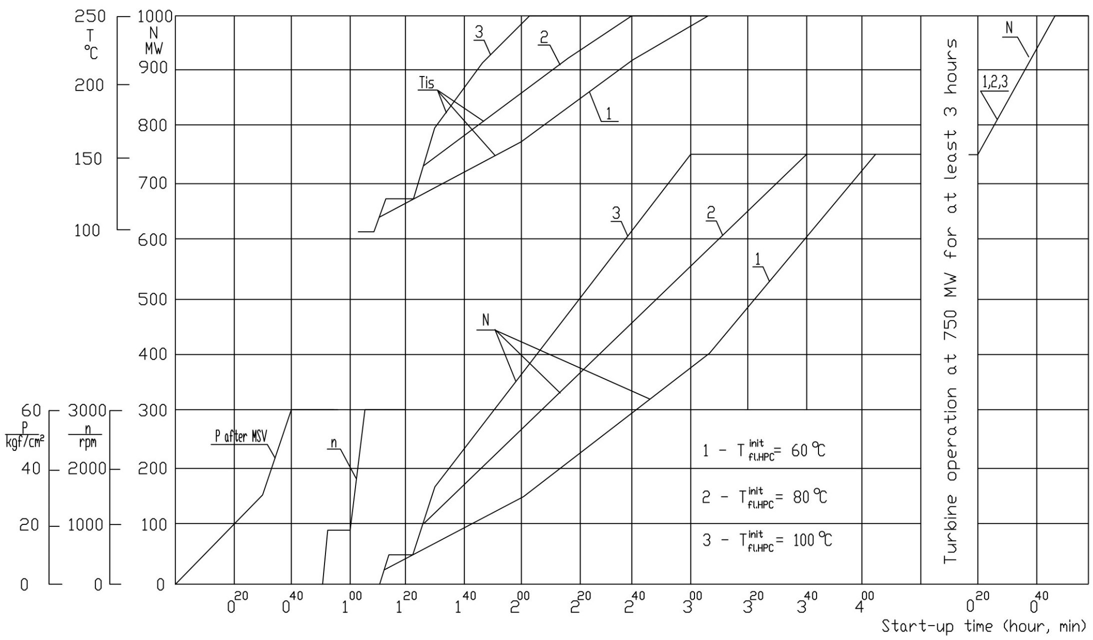


Fig. 1. Task graph of starting the K-1000-60/3000 turbine from a cold state:
T_{is} – intermediate superheating; MSV – main steam valve

When calculating the thermal state of the HPC rotor during start-up modes, the non-stationary problem of thermal conductivity, which requires the establishment of the boundary conditions of heat exchange of the I-IV type, which must necessarily correspond to the task graphs of the start-up of the K-1000-60/3000 turbine from various thermal states, is solved. Graphs of start-up with cold state are shown in Fig. 1.

Technical audit of the K-1000-60/3000 steam turbine of the NPP power unit

During start-up tests of the system of the turbine automatic adjustment at idle speed, an event occurred. That event was related to the destruction of the 5th stage of the left flow of the HPC rotor of the turbine, as a result of the destruction of the supporting ridge of the working disc on the side of the steam inlet due to the occurrence of fatigue damage in the area of the holes for the blade fastening pins. Post-repair tests of the automatic regulation system of the power unit were performed.

During the inspection, the following damages of the 5th stage of the left flow of the HPC rotor were found: the destruction of the disc of the blades in the blades fastening pins; mechanical damage (burrs, color change) of the end of the working disc on the side of the steam inlet; breakage of all stators from the lower and upper half of the diaphragm rim and body; rotation of the diaphragm clamp by ~50 mm counterclockwise; breakage of four bolts that connect the upper and lower clamps of the diaphragm; traces of friction "metal on metal" (with characteristic colors of variability) on the upper body of the diaphragm and the end of the working disc; mechanical damage (burrs, dents, metal encrustation) of the labyrinth seals of the HPC rotor diaphragm; on the entire surface of the lower and upper diaphragm rim and body in the places where the stators are attached, mechanical damage (burrs, dents, metal wear); the destroyed profile of the upper and lower cage of the diaphragms in the blades area.

According to the results of the inspection of the corrosion condition after opening the HPC, it was established that the surface of the HPC rotor blades and the inner surface of the HPC body are covered with a thin, uniform layer of finely dispersed brown deposits that are easily removed. The conducted analysis indicates that damage to the rotor blade apparatus of the HPC rotor at the NPP is quite rare. A similar incident occurred in 1995 and was caused by the separation of the diaphragm stators between the 4th and 5th stages on the generator side. After breaking off, the stator got into the gap between the diaphragm and the blades of the 5th degree, which led to the breakage of all the blades of the lower half of the diaphragm and two blades of the upper half of the diaphragm (in our case, both the lower and upper blades of the diaphragm were broken). After the destruction of the diaphragm blades, the body of the diaphragm fell on its rim and hit the blades of the 5th degree. As a result of the impact, the blades of the 5th stage of the HPC rotor and the disc rim were destroyed. According to the conclusion of the manufacturer, the cause of this event is the rupture of the diaphragm stators due to the action of loads during operation, that is, the manufacturer recognized a possible structural defect of the K-1000-60/3000 turbine.

Based on operational experience and taking into account the conclusions made by the commission during the investigation of the event of 1995, it is proposed: to measure the deflection under the control load of the diaphragm of the 5th degree and the diaphragm of the 4th degree of both steam flows (left and right); to conduct similar checks on all K-1000-60/3000 turbines operated in Ukraine; to organize the manufacture and replacement of diaphragms that have a deflection higher than the permissible one, with improved ones.

According to the act of defecting of the HPC of the K-1000-60/3000 power unit and in view of the destruction of the 5th stage of the left flow of the HPC rotor, the main damages of the high-pressure rotor of the NPP turbine, detected during the diagnostic control of the technical condition, are: damage to the fastening of the blades of the 5th stage of the left flow with separation from the disc and destruction of the ridges of the disc of the rotor shaft of the 5th stage of the left flow in the fastening points of the blades; mechanical damage (scratches, burrs) on the end surfaces of the shaft discs of the 5th stage of the left flow on the side of the steam inlet and the 4th stage of the left flow on the side of the regulator; mechanical damage (wear) on the rotor shaft of the labyrinth seal of the diaphragm of the 5th stage of the left flow; other damage to the blades of the 4th stage and the bandages of the blades of the 3rd, 4th stages.

In relation to these damages, the following repairs were carried out: the ridges of the 5th stages of the left and right flow were drilled until they were completely removed; grinding of the damaged end surfaces of the discs of the 4th and 5th stages of the left flow was performed; diaphragmatic labyrinth sealing of the 5th degree was carried out until the damaged places were eliminated; new blades of the 3rd and 4th stages were installed with the adoption of accompanying technical measures. According to the results of non-destructive testing of the state of the metal of the high-pressure rotor, no other defects were found.

A geometric model of the HPC rotor was created during the calculated assessment of changes in the thermal and stress-strain state of the HPC rotor, taking into account the data of the technical audit. For the structurally complex high-pressure rotor, the geometric model is made in a three-dimensional setting taking into account the main structural elements based on the passport drawing of the K-1000-60/3000 turbine (Fig. 2).

Due to the symmetry of the HPC flows, it is allowed to consider one flow for conducting numerical studies [5, 6]. During the development of the calculated analogue of the HPC rotor, the construction of all forged surfaces of the left flow of the rotor from the central plane of symmetry between the two steam flows to the last segment of the end seals was carried out. All structural elements, including welding fillets and radial transitions of stage discs, unloading holes, tail mounts of blades, geometry of end and diaphragm seals are reproduced according to industrial drawings of the manufacturing plant (Fig. 3, a).

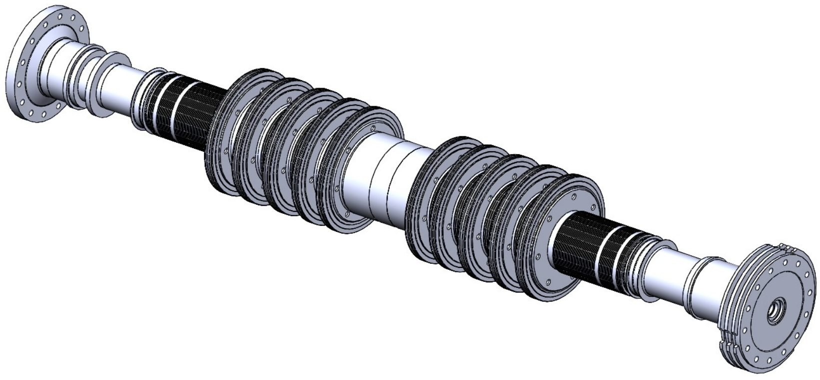
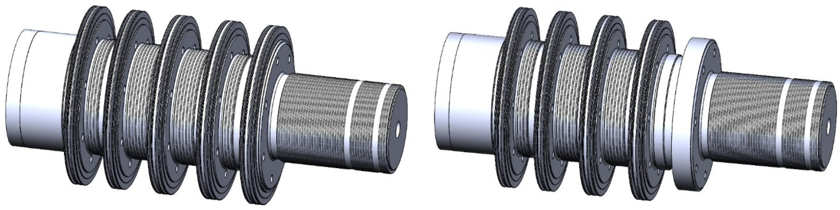


Fig. 2. Complete geometric model of the HPC rotor of the K-1000-60/3000 turbine



a b

Fig. 3. Calculated geometric analogue of the HPC rotor of the K-1000-60/3000 turbine:
a – in the design, b – after repairs

In addition, another geometric analogue of the HPC rotor, which takes into account the results of repairs and restorations in accordance with the act of defecting, was developed. The main changes made to the project design are the grooving of the ridges of the tail mounts of the 5th stage blades until they are completely removed, the grooving of the labyrinth seal of the diaphragm of the 5th stage on the shaft, cleaning and grinding of the ends of the discs of the 5th and 4th stages, etc. (Fig. 3, b).

Study of the thermal and stress-strain state of HPC rotors of NPP steam turbines

The calculated assessment of the thermal and stress-strain state of the HPC rotor contains equations of unsteady thermal conductivity with boundary conditions of heat exchange on the rotor surfaces according to the developed software complex [1].

The boundary conditions of heat exchange were determined on the basis of a detailed calculation of the HPC steam compartments for various design features of the HPC rotor.

The equation of non-stationary thermal conductivity has the form [7, 8]

$$\mathrm{div}[\lambda(T)\cdot \mathrm{grad}(T)] = c(T)\cdot\gamma(T)\cdot\frac{\partial T}{\partial\tau},$$

where λ, c, γ are functions of temperature and coordinates under the initial condition $T_0=T(x, y, z, 0)=f_0(x, y, z)$ and boundary conditions of the I–IV type.

Boundary conditions of the I–IV kind have the form

$$T_{\mathrm{w}\,1} = T_{\mathrm{w}\,2};$$

$$T_{\mathrm{w}} = f(x, y, z, \tau) ;$$

$$q = -\lambda\left(\frac{\partial T}{\partial n}\right) = f_2(x, y, z, \tau) ;$$

$$-\lambda\left(\frac{\partial T}{\partial n}\right) = \alpha(t_{\mathrm{m}} - t_{\mathrm{w}}) ;$$

$$\begin{cases} T_{\mathrm{w}\,1} = T_{\mathrm{w}\,2} \\ -\lambda_1\left(\dfrac{\partial T}{\partial n}\right)_1 = -\lambda_2\left(\dfrac{\partial T}{\partial n}\right)_2 \end{cases} .$$

When determining the boundary conditions of the heat exchange of the HPC rotor, it is necessary to have information about its characteristic dimensions and to perform a detailed calculation of the flow part at the nominal operating mode. During the detailed calculation of the compartment, the main thermodynamic parameters of the steam (pressure, temperature, specific volume), enthalpy differences, loss values, and velocity values on the average cross-section for the nozzle and blades of each studied stage are determined. In case of non-stationary operating modes, the calculated estimate of the above steam parameters is used for costs corresponding to the start-up schedules of the NPP power unit (Figs. 1–3).

After determining the main parameters of the steam in the nominal and variable operating modes, the boundary conditions were determined according to the normative document [7, 8].

The coefficient of heat transfer from the steam to the inter-blade surfaces of the HPC rotor is determined by the formula

$$Nu = 0.206 \cdot \mathrm{Re}^{0.66} \cdot s_r^{-0.58} ;$$

$$s_r = \frac{\sin\beta_1}{\sin\beta_2}\sqrt{\frac{2b_0}{\bar{t}\cdot l\cdot\sin(\beta_1+\beta_2)\cdot\cos^2\left(\dfrac{\beta_1-\beta_2}{2}\right)}} .$$

To establish the Reynolds and Prandtl similarity criteria, the length of the surface in the direction of the blade was used. At the same time, the speed is calculated as the arithmetic mean value of the relative velocity at the inlet and outlet of the blade, and the temperature is the arithmetic mean temperature of the medium at the inlet and outlet of the blade.

For rotor stage discs rotating in a large volume, the similarity equation has the form

$$Nu = 0.0197 \cdot (n + 2.6)^{0.2} \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.6} ,$$

where n is exponent in the equation of temperature pressure change along the disc radius

$$t_{\mathrm{w}} - t_{\mathrm{m}} = c \cdot r^n .$$

The following similarity equation is used for rotor stage discs rotating in a casing between adjacent diaphragms:

$$Nu = 0.0256 \cdot (1 - z_{\varphi})^{0.75} \cdot \mathrm{Re}^{0.75} \cdot \mathrm{Pr}^{0.6} \cdot \left(\frac{s}{r}\right)^{0.25} .$$

The determining size is the radius of the calculated section, the determining speed is the circular speed at a given radius, the determining temperature is the temperature of the medium that washes the disc.

For sections of the rotor with direct-flow seals, the equations of convective heat exchange were used in the form

$$\begin{cases} Nu = \dfrac{0.256 \cdot \mathrm{Re}^{0.6} \cdot \mathrm{Pr}^{0.43}}{\left(\dfrac{s}{\delta}\right)^{0.085} \cdot \left(\dfrac{h}{\delta}\right)^{0.075}}, \text{ at } \mathrm{Re} = 2.4 \cdot 10^2 \ldots 8.7 \cdot 10^3 \\ Nu = \dfrac{0.0454 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.43}}{\left(\dfrac{s}{\delta}\right)^{0.1} \cdot \left(\dfrac{h}{\delta}\right)^{0.1}}, \text{ at } \mathrm{Re} = 8.7 \cdot 10^3 \ldots 1.7 \cdot 10^5 \end{cases},$$

where s is the step between the ridges of the seals; h is the distance between the surface of the rotor and the cylinder body; δ is clearance between the rotor surface and the ridges of the seals.

For stage seals, the formula is applied

$$\begin{cases} Nu = 2.04 \cdot \mathrm{Re}^{0.5} \cdot \left(\dfrac{h}{\delta}\right)^{-0.56} \cdot \mathrm{Pr}^{0.43}, \text{ at } \mathrm{Re} \le 1 \cdot 10^4 \\ Nu = 0.476 \cdot \mathrm{Re}^{0.7} \cdot \left(\dfrac{h}{\delta}\right)^{-0.56} \cdot \mathrm{Pr}^{0.43}, \text{ at } 6 \cdot 10^3 < \mathrm{Re} < 1.2 \cdot 10^5 \end{cases}.$$

For diaphragm and intermediate seals with straight-flow or stage labyrinths, an equation of the form is used

$$Nu = \frac{0.052}{k} \cdot \mathrm{Re}^{0.9} \cdot \left(\frac{\delta}{h}\right)^{0.7} \cdot \mathrm{Pr}^{0.43}, \text{ at } 3.5 \cdot 10^3 < \mathrm{Re} < 2.5 \cdot 10^4,$$

where z is the number of seal ridges; p_1, p_2 is full pressure before and after the labyrinth; k is flow rate for a given type of seal, determined by equation

$$k = \frac{G}{f\sqrt{\dfrac{g(p_1^2 - p_2^2)}{z \cdot R \cdot T}}}.$$

For these types of seals, the determining size is twice the size of the gap $2 \cdot \delta$.

The determining speed is the average speed of steam in the seal:

$$W_{\mathrm{m}} = \frac{G_y \cdot v_{\mathrm{m}}}{F_y},$$

where G_y is consumption of sealing steam

$$G_y = \mu_y \cdot F_y \cdot \sqrt{\frac{p_0}{v_0}} \cdot \sqrt{\frac{1 - \left(\dfrac{p_2}{p_1}\right)^2}{z}};$$

F_y is working area of seals

$$F_y = \pi \cdot d_y \cdot \delta.$$

The determining temperature is the arithmetic mean temperature of the steam at the inlet and outlet of the seals.

Heat exchange on the surfaces of the rotor shaft in contact with air is described by the criterion equation

$$Nu = 0.11 \cdot (0.5 \cdot \mathrm{Re}^2 + Gr)^{0.33}$$

for the range $10^5 < 0.5 \cdot Re + Gr < 10^9$.

The determining size is the outer diameter of the rotor, the determining speed is the circular speed of the rotor at the outer radius, the determining temperature is the average temperature of the boundary layer.

For the part of the rotor surface located in the bearings, the heat transfer coefficient is found from the similarity equation of the following form

$$Nu = 6 \cdot (\mathrm{Re_o} \cdot \mathrm{Pr_o})^{0.33} \cdot \frac{d_{\mathrm{jb}}}{l_{\mathrm{jb}}},$$

where d_{jb} is diameter of the journal bearing of the rotor shaft; l_{jb} is the length of the oil-washed surface of the journal bearing of the rotor shaft.

When establishing the similarity criteria of Reynolds and Prandtl, the diameter of the shaft journal bearing is the determining dimension d_{jb}, the determining speed is the circular speed on the given diameter u, and the determining temperature is the average arithmetic temperature of the oil at the entrance and exit from the bearing t_{o}.

Thus, on the heat exchange surfaces of the HPC rotor of the K-1000-60/3000 turbine, boundary conditions of the III type were set using hyperbolic interpolation, and on the surface of the axial channel, boundary conditions of the II type were set. Schemes of steam leaks in the flow part and in seals were taken into account, as well as real work schedules under typical operating modes, namely stationary and starts from cold, uncooled and hot states.

The stress-strain state of the HPC rotor was evaluated in the elastic-plastic formulation using the finite-element method of discretization of the computational domain. The main types of stress were taken into account, namely temperature stress, non-uniformity of temperature fields, pressure stress and centrifugal force [7, 8].

The equilibrium equation in tensometric form is

$$\{\sigma_i\}_j + \rho X_i = 0\,;\ i,j=1, 2, 3;\ \ p_i = f(x, y, z, 0),$$

where $\{\sigma_i\}_j$ are normal and tangential stresses in HPC rotor elements; X_i is the mass force acting in the elements of the rotor (centrifugal force, gravity, resistance reactions, etc); p_i is the external distributed load; ρ_i is the density of steel turbine.

The equation of the compatibility of deformations and the law of elasticity in matrix form has the form

$$\{\varepsilon_{ij}\} = [a]\{\sigma_{ij}\} + \{\beta \cdot \Delta T\},$$

where $\{\varepsilon_{ij}\}$ is the deformation vector; $[a]$ is the matrix of elasticity coefficients; $\{\sigma_{ij}\}$ is the stress vector; $\{\beta \cdot \Delta T\}$ is the vector of temperature deformations; β is the volumetric expansion coefficient; ΔT is the change in temperature of HPC rotor elements during operation.

During calculation studies of the thermal and stress-strain state of the HPC rotor, all thermal-physical and physical-mechanical characteristics of 30HN3M1FA (KP-60) steel were determined depending on the temperature in accordance with regulatory documents [8].

Discussion of results

The thermal and stress-strain state for the stationary operating mode is performed in a quasi-stationary setup. The temperature level is 270 °C. The maximum stress intensity is observed in the axial hole and in the unloading holes of the discs of all five stages and is equal to 158 MPa, which is explained by the large values of centrifugal forces acting on the discs of the pressure stages and blades. The highest level of stress occurs in the zone of the fifth degree, which is the most massive and covered with the heaviest blades.

The starting operating modes are considered in the non-stationary setting. Information on the unevenness of temperature fields over time, which is depicted in the form of the dynamics of temperature gradient changes for the most characteristic areas, is of particular interest in variable operating modes [9, 10]. For start-up from a cold state, the temperature gradient reaches 1200–1300 K/m in the initial stages of start-up, which indicates the existing non-uniformity of temperatures.

A numerical study of the thermal and stress-strain state of the HPC rotor of the K-1000-60/3000 turbine was performed for the most typical operating modes, namely: nominal at an electric power of 1000 MW, start-ups from cold and hot metal states. At the same time, three options of the design of the HPC were considered: the project design, without the blades of the 5th stage of the HPC with the regular nozzle apparatus, without the entire 5th stage.

The finite element method was used to solve the boundary value problem of non-stationary thermal conductivity and calculate the thermal stress state at typical operating modes. The calculated model of the rotor in a three-dimensional setting was discretized by more than 10 million finite elements with a thickening of the grid in the radial direction, especially in the zones that are stress concentrators, which include the relief holes and the rounding of the stage discs, the ridges of the diaphragm and end seals, etc. In addition, significant attention was paid to places where repairs and restorations were carried out in accordance with the act of defecting. The grid of finite elements thickens to the outer surface of the rotor according to the law of geometric progression, when each finite element closer to the outer surface of the shaft is 1.4 times smaller than the previous one. The size of the smallest element is 1 mm.

Calculation studies of the thermal and stress-strain state of the HPC rotor at the nominal operating mode for various structural versions of the HPC were carried out in a quasi-stationary setting. Thermophysical and mechanical properties of 30HN3M1FA steel are temperature-dependent and specified using exponential approximation.

For the original option, the highest temperature of the base metal is typical for the disc of the first stage and is 264.5 °C (Fig. 4, a). The temperature of the metal of 2–5th stages decreases uniformly from 235.5 to 169 °C. In the area of end seals, the temperature of the metal also gradually decreases from 146.5 to 114.3 °C. Thus, it can be stated that there is a uniform temperature field of the HPC rotor, and that the value of the temperature gradients does not exceed 800 °C/m. The largest temperature gradients are observed at the ends of the discs of all five stages along the steam flow after the blades and in the area of the chimney chamber of the end seals.

The obtained stress-strain state shows that for the design structure (Fig. 4, b) when working at the nominal parameters of the steam, the most stressed areas are the discharge openings of the 5th stage (σ_i=202.8 MPa), the axial opening of the rotor in the area of 5-th degree (σ_i=195.2 MPa), as well as the 5th degree welding fillets on the side of the end seals (σ_i=200.3 MPa) and the unloading holes of the 4th and 3rd degrees with a stress intensity of about 170–185 MPa. The high values of the stress intensity in the area of the 5th stage can be explained by the significant mass concentration of both the stage itself and its blades, which provoke significant centrifugal forces when working at the nominal rotation frequency.

For the option of the HPC without the blades of the 5th stage (Fig. 5, a), the temperature field of most of the rotor is similar to the original option. Higher temperatures are observed for the end wall of the 5th stage disc on the exhaust side and for the metal in the area of the final seals (from 161.2 to 123.5 °C). This is due to the higher retardation temperature of the steam after the five stages of the cylinder.

For a HPC without blades of the 5th stage, there is a shift of the maximum stress intensity to the area of the unloading holes of the 4th and 3rd stages, as well as the axial hole of the shaft under the same stages (Fig. 5, b). The maximum stress value is $\sigma_{i\ \max}$=184.8 MPa. At the same time, the stress intensity in the area of the discharge holes of the 5th degree decreased almost by half to the level of 124 MPa.

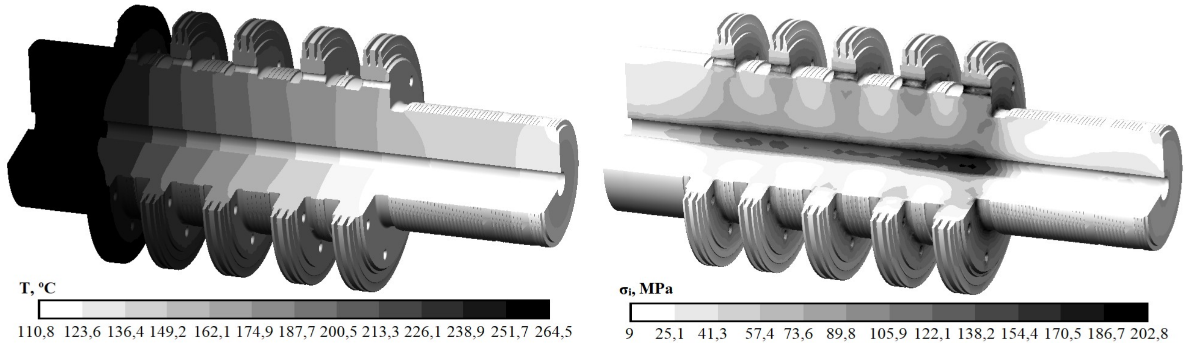
The HPC option without the 5th degree differs most significantly from the design one (Fig. 6, a). Due to the increase in thermal differences that trigger the 3rd and 4th stages, the temperatures of the metal of the discs are lower by 2–6 °C. At the same time, the temperature of the metal of the 5th stage disc drops significantly to 157 °C (by 12 °C). The temperature of the metal in the area of the end seals is similar to the original option (from 151.1 to 115.3 °C). There is a noticeable increase in the temperature gradients at the ends of the discs of the 3rd and 4th stage to 857 and 943 °C/m, respectively, which is still a relatively low value, but indicates the complication of the working conditions of the metal of these stages.

The geometric and mass characteristics of the rotor in the design of the HPC without the entire 5th stage and without the blades of the 5th stage are the same. However, due to significant differences in the thermal state, there are similar differences for the stress-strain state. Thus, the largest stress concentrators continue to be the unloading holes of the 4th and 3rd stages and the axial hole below them, but the maximum value of the stress intensity increases to $\sigma_{i\ \max}$=193.7 MPa (Fig. 6, b), which indicates more difficult working conditions of the 4th grade metal in this design of the HPC.

At the same time, it should be noted that the maximum level of stress intensity, which occurs in the main metal of the HPC rotor, does not exceed the yield strength of 30HN3M1FA steel ($\sigma_{0.2}$=493 MPa) at the calculated temperature ($t_{\max}$=270 °C) for all three design options of HPC.

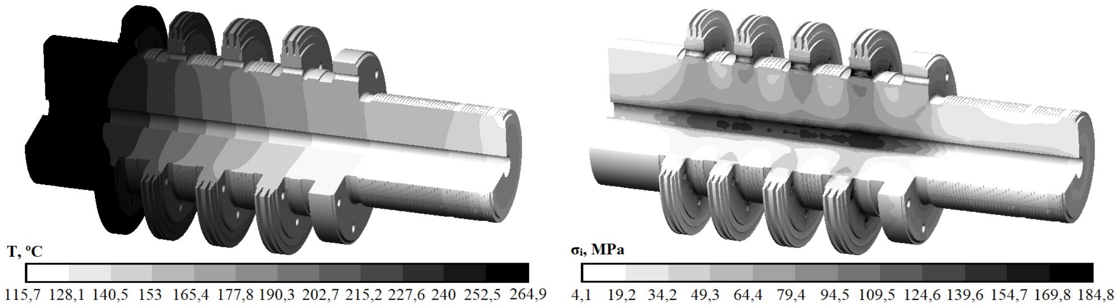
When studying the thermal state of the HPC rotor when operating at the nominal parameters of the steam, it was established that the thermal field is quite similar in the three design options of the HPC flow part. The key differences are: for HPC without blades of the 5th stage, there is an increase in the temperature of the metal in the area of the final seals from the exhaust to chamber *A* by 9–15 ºC; for HPC without the 5th stage, the temperature of the end part of the disc of the 5th stage decreases by 12 ºC.

The specified changes in the thermal state, as well as changes in the geometry and mass characteristics of the shaft, also lead to changes in the stress-strain state of the HPC rotor. For the option of the HPC without blades of the 5th stage, the maximum stress intensity decreased by 8.9%, for the option without the 5th stage – by 4.5%.



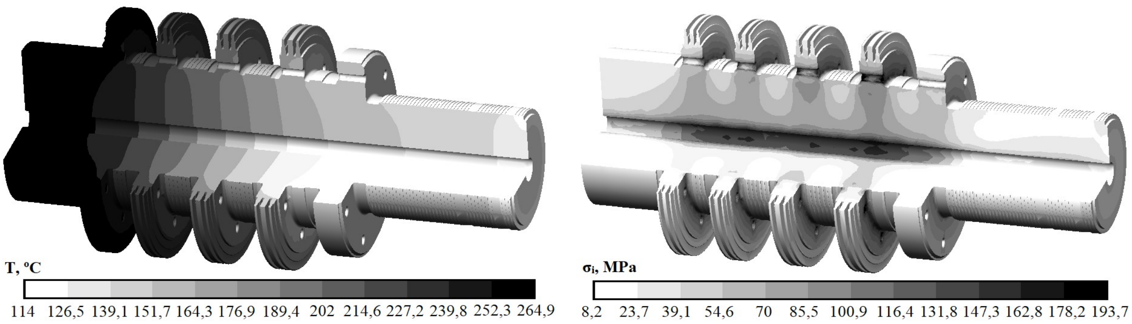
a b

Fig. 4. Thermal (a) and stress-strain state (b) at the nominal operating mode (1000 MW) of the HPC rotor of the K-1000-60/3000 turbine in the design option



a b

Fig. 5. Thermal (a) and stress-strain state (b) at the nominal operating mode (1000 MW) of the HPC rotor of the K-1000-60/3000 turbine without blades of the 5th pressure stage with standard nozzle apparatus



a b

Fig. 6. Thermal (a) and stress-strain state (b) at the nominal operating mode (1000 MW) of the HPC rotor of the K-1000-60/3000 turbine without all the 5th pressure stage

It should be noted that information on the non-uniformity of temperature fields in the form of non-stationary temperature gradients is of considerable interest for the start-up modes of power equipment. In the HPC rotor, 12 characteristic areas of research are chosen. They are shown in Fig. 7, namely:

1 – axial hole of the rotor in the region of the 4th pressure stage;
2 – tail fastening of the blades of the 1st stage;
3 – 2nd-stage welding fillet from the exhaust side;
4 – 3rd-stage exhaust welding fillet;
5 – unloading hole of the 3rd stage disc;
6 – diaphragm sealing of the 4th degree;
7 – unloading hole of the 4th stage disc;
8 – diaphragm seal of the 5th degree;
9 – unloading hole of the 5th stage disc;
10 – 5th-stage exhaust welding fillet;
11 – ridges of final seals on the side of chamber *A*;
12 – rotor shaft in the area of chamber *B* of end seals.

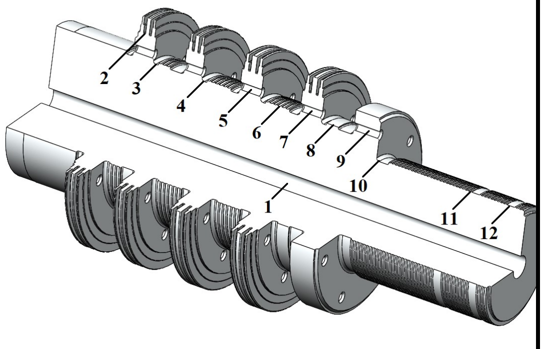
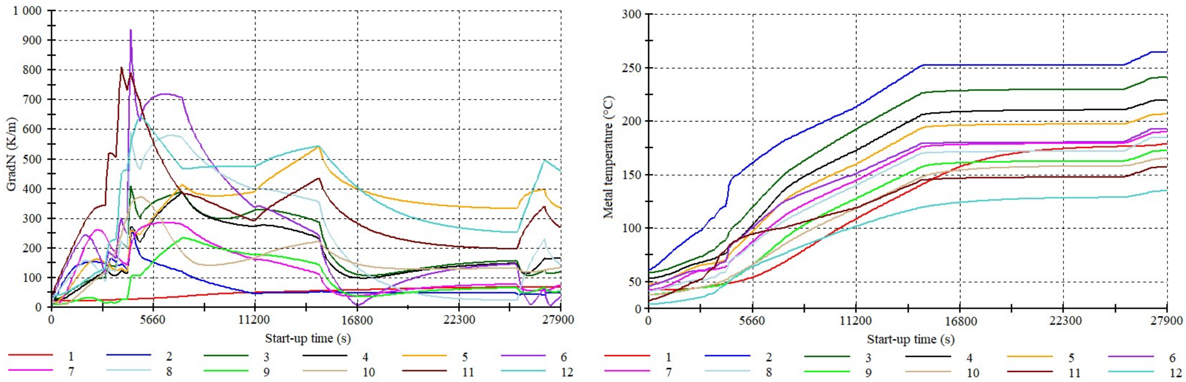


Fig. 7. Characteristic areas of HPC rotor research with repair restorations

Fig. 8 shows the dynamics of changes in temperature gradients and metal temperature during start-up from a cold state in the characteristic areas of research of the HPC rotor of the K-1000-60/3000 turbine.

During start-up from the metal cold state, high values of temperature gradients are observed in most of the studied areas (Fig. 8, a) starting from the moment of time of 3040 s, which corresponds to the moment of the rotor shock, and the maximum value of the temperature gradient during the entire start-up is grad *T*=952 K/m – in the area of the diaphragm seal of the 4th degree at the moment of time of 4400 s (synchronization with the power grid). Such a high value can be explained by the unformed contact flow of the flow of the sealing steam throttled in the previous segments of the seals and by the loss of sharp steam after the nozzle apparatus of the 3rd stage. In the subsequent stages of start-up, together with the normalization of the nature of the flow in the turbine flow part and end seals, as well as the stabilization of the thermal field of the HPC rotor, the values of the temperature gradients are significantly reduced to 400 K/m.



a b

Fig. 8. The dynamics of changes in temperature gradients in the characteristic areas of the rotor study with start-up from a cold state of the metal:
a – temperature gradients; b – temperatures

While studying the dynamics of changes in the temperature of the metal in the characteristic areas of the study, it can be noted that the heating of the base metal occurs quite smoothly. The rotor undergoes the fastest heating during 4200–4480 s (from the moment of synchronization with the load up to 50 MW), but at the same time, the metal heating rate of 5.2 °C/min still does not exceed the permissible values (7 °C/min). Together with the end of the start-up period, the stabilization of the temperature field of the rotor is almost completed, which is associated with a sufficiently long endurance of the rotor at the electric power of the generator of 750 MW.

The analysis of the above data allows to establish the moments of time at which the unevenness of the temperature fields has the greatest impact on the stress-strain state of the HPC rotor. At the same time, it should be taken into account that the cyclic stresses will not always be maximum at these moments of time, since there are other mechanical loads that the metal of the HPC rotor is subjected to, such as internal stresses from thermal expansion, tensile centrifugal forces of shaft rotation, pressure forces of the steam medium, bearing supports reactions and others.

The main stress concentrators in the nominal operating mode are the unloading holes and the rounding of the stage discs, as well as the axial hole of the shaft.

Conclusions

1. When studying the thermal state of the HPC rotor when working at the nominal parameters of the steam, it was established that the thermal field is quite similar for the three design options of the HPC flow part. The key differences are: for HPC without blades of the 5th stage, there is an increase in the temperature of the metal in the area of the final seals from the exhaust to chamber *A* by 9–15 ºC; for HPC without the 5th stage, the temperature of the end part of the disc of the 5th stage decreases by 12 ºC. The indicated changes in the thermal state, as well as changes in the geometry and mass characteristics of the shaft, also lead to changes in the stress-strain state of the HPC rotor.

2. The maximum intensity of stresses for the option without the entire 5th stage in comparison with the option of the HPC without blades of the 5th stage increased by 12%.

3. To increase the reliability of turbine elements, reduce heat loads and improve operating conditions, it is recommended to modernize the control system of the main parameters of the turbine with the registration of parameters affecting the reliability and resource of the turbine; implement systems for monitoring the vibration activity of the turbine unit with diagnostics of the condition of the elements of the shaft pipeline, including the presence of cracks in the rotors; adhere to the task schedules developed by the manufacturing plant and try to minimize significant deviations; implement systems of control and technical diagnostics of the thermal and stress-strain state of HPC rotors, as well as HPC housings, stop and control valves, based on real-time modeling of the thermal and stress-strain state of the equipment.

References

1. Chernousenko, O. Yu., Ryndiuk, D. V., & Peshko, V. A. (2020). *Otsinka zalyshkovoho resursu ta podovzhennia ekspluatatsii parovykh turbin velykoi potuzhnosti* [Estimation of residual life and extension of operation of high-power steam turbines]. Part 3: monograph for scientists and doctors of philosophy in the specialty 144 "Heat Power Engineering". Kyiv: National Technical University of Ukraine "Igor Sikorsky Kyiv Polytechnic Institute", 297 p. (in Ukrainian).
2. Mazur, Z., Herna'ndez-Rossette, A., & Garcí'a-Illesoas, R. (2006). L-0 blades failure investigation of a 110 MW geothermal turbine. ASME 2006 Power Conference proceedings, USA, Atlanta, May 2–4, 2006, no. POWER2006-88024, pp. 281–289. https://doi.org/10.1115/POWER2006-88024.
3. Suzuki, T., Matsuura, T., Sakuma, A., Kodama, H., Takagi, K., & Curtis, A. (2005). Recent upgrading and life extension technologies for existing steam turbines. ASME 2005 Power Conference, USA, Chicago, April 5–7, 2005, no. PWR2005-50342, pp. 577–582. https://doi.org/10.1115/PWR2005-50342.
4. Sanders, W. P. (1988). Turbine steam path engineering for operations and maintenance staff. Canada, Toronto: Turbo-Technic Services Inc.
5. Peshko, V., Chernousenko, O., Nikulenkova, T., & Nikulenkov, A. (2016). Comprehensive rotor service life study for high & intermediate pressure cylinders of high power steam turbines. *Propulsion and Power Research*, vol. 5, iss. 4, pp. 302–309. https://doi.org/10.1016/j.jppr.2016.11.008.
6. Chernousenko, O., Butovsky, L., Rindyuk, D., Granovska, O., Moroz, O. (2017). Analysis of residual operational resource of high-temperature elements in power and industrial equipment. *Eastern-European Journal of Enterprise Technologies – Energy-saving technologies and equipment*, vol. 1, no. 8 (85), pp. 20–26. http://dx.doi.org/10.15587/1729-4061.2017.92459.
7. (2021). *Kontrol metalu i prodovzhennia terminu ekspluatatsii osnovnykh elementiv kotliv, turbin i truboprovodiv teplovykh elektrostantsii* [Metal control and extension of service life of the main elements of boilers, turbines and pipelines of thermal power plants]. Regulatory document of the Ministry of Fuel and Energy of Ukraine. Typical instruction SOU-N MPE 40.17.401:2021. Kyiv: HRIFRE, Ministry of Fuel and Energy of Ukraine, 214 p. (in Ukrainian).
8. Shulzhenko, M. H., Hontarovskyi, P. P., Matiukhin, Yu. I., Melezhyk, I. I., & Pozhydaiev, O. V. (2011). *Vyznachennia rozrakhunkovoho resursu ta otsinka zhyvuchosti rotoriv i korpusnykh detalei turbin.* [Determination of estimated resource and evaluation of rotor life and body parts of turbines: Methodological guidelines. Regulatory

document SOU-N MEV 0.1–21677681–52:2011: approved by the Ministry of Energy and Coal Mining of Ukraine: effective as of 07.07.11. Kyiv: Ministry of Energy and Coal Mining of Ukraine (in Ukrainian).
9. Chernousenko, O. Yu., Ryndiuk, D. V., & Peshko, V. A. (2019). *Napruzheno-deformovanyi stan rotora turbiny K-1000-60/3000 pry typovykh rezhymakh ekspluatatsii* [Stress-strain state of the turbine rotor K-1000-60 / 3000 under typical operating conditions]. *Visnyk NTU «KhPI». Seriia: Enerhetychni ta teplotekhnichni pro-tsesy y ustatkuvannia – Bulletin of NTU "KhPI". Series: Power and Heat Engineering Processes and Equipment*, no. 3 (1328), pp. 4–10 (in Ukrainian). https://doi.org/10.20998/2078-774X.2019.03.01.
10. Chernousenko, O. Yu. & Peshko, V. A. (2019). Assessment of resource parameters of the extended operation high-pressure rotor of the K-1000-60/3000 turbine. *Journal of Mechanical Engineering – Problemy Mashynobuduvannia*, vol. 22, no. 4, pp. 41–47. https://doi.org/10.15407/pmach2019.04.041.14.

Received 08 August 2023

Зміни теплового та напружено-деформованого стану ротора ЦВТ потужної турбіни АЕС після пошкодження лопаток

[1] О. Ю. Черноусенко, [1] В. А. Пешко, [2] О. П. Усатий

[1] Національний технічний університет України «Київський політехнічний інститут імені Ігоря Сікорського»
03056, Україна, м. Київ-56, пр. Перемоги, 37

[2] Національний технічний університет «Харківський політехнічний інститут»
61002, Україна, м. Харків, вул. Кирпичова, 2

На практиці при експлуатації парових турбін мають місце аварійні пошкодження робочих лопаток роторів і направляючих апаратів потужних парових турбін. Головними причинами аварійних зупинок парових турбіни були вібраційна втома матеріалу лопаток, ерозійне пошкодження тіла лопаток і резонансні проблеми при роботі енергообладнання. Виходячи з цього дослідження, пов'язані з оцінкою змін теплового й напружено-деформованого стану елементів енергетичного обладнання, які на АЕС значно впливають на продовження експлуатації турбіни після її пошкодження, є досить актуальними. Розглянуто й проаналізовано зміни теплового й напружено-деформованого стану, які можуть виникнути після пошкодження ротора циліндра високого тиску (ЦВТ) турбіни К-1000-60/3000 енергоблоку ЛМЗ в умовах станції і забезпечать можливість оцінки індивідуального ресурсу й продовження роботи енергоблоку. При розрахунковій оцінці змін теплового та напружено-деформованого стану ротора ЦВТ, беручи до уваги дані технічного аудиту щодо пошкоджень, створена геометрична модель ротора. Проведені дослідження для трьох варіантів конструкцій: вихідний варіант (п'ять ступенів ротора ЦВТ), варіант без робочих лопаток останнього ступеня і варіант без п'ятого ступеня (з чотирма першими ступенями). Для проєктної конструкції при роботі на номінальних параметрах пари найбільш напруженими областями є розвантажувальні отвори 5-го ступеня (σ_i=202,8 МПа), осьовий отвір ротора в області 5-го ступеня (σ_i=195,2 МПа), а також галтель 5-го ступеня з боку кінцевих ущільнень (σ_i=200,3 МПа) і розвантажувальні отвори 4-го і 3-го ступенів з інтенсивністю напружень близько 170–185 МПа. Високі значення інтенсивності напружень в області 5-го ступеня можна пояснити суттєвим зосередженням маси як самого ступеня, так і його робочих лопаток, що провокують значні відцентрові зусилля при роботі на номінальній частоті обертання. Для ротора ЦВТ без робочих лопаток 5-го ступеня спостерігається зміщення максимумів інтенсивності напружень в область розвантажувальних отворів 4-го і 3-го ступенів, а також осьового отвору валу під цими ж ступенями. Максимальне значення напружень складає $\sigma_{i\ max}$=184,8 МПа. У той же час інтенсивність напружень в області розвантажувальних отворів 5-го ступеня зменшилася майже вдвічі, до рівня в 124 МПа.

***Ключові слова**: атомна електростанція, парова турбіна К-1000-60/3000, ротор циліндра високого тиску, потужність, тиск, температура, втрата, парковий ресурс, нестаціонарна теплопровідність, тепловий стан, напружено-деформований стан, малоциклова втома, довготривала міцність, залишковий ресурс, допустиме число пусків.*

Література

1. Черноусенко О. Ю., Риндюк Д. В., Пешко В. А. Оцінка залишкового ресурсу та подовження експлуатації парових турбін великої потужності (частина 3): монографія для науковців та докторів філософії за спеціальністю 144 «Теплоенергетика». Київ: НТУУ «КПІ імені Ігоря Сікорського», 2020. 297 с.
2. Mazur Z., Herna'ndez-Rossette A., Garcí'a-Illesoas R. L-0 blades failure investigation of a 110 MW geothermal turbine. ASME 2006 Power Conference proceedings, USA, Atlanta, May 2–4, 2006. No. POWER2006-88024. P. 281–289. https://doi.org/10.1115/POWER2006-88024.
3. Suzuki T., Matsuura T., Sakuma A., Kodama H., Takagi K., Curtis A. Recent upgrading and life extension technologies for existing steam turbines. ASME 2005 Power Conference, USA, Chicago, April 5–7, 2005. No. PWR2005-50342. P. 577–582. https://doi.org/10.1115/PWR2005-50342.
4. Sanders W. P. Turbine steam path engineering for operations and maintenance staff. Canada, Toronto: Turbo-Technic Services Inc., 1988.
5. Peshko V., Chernousenko O., Nikulenkova T., Nikulenkov A. Comprehensive rotor service life study for high & intermediate pressure cylinders of high power steam turbines. *Propulsion and Power Research*. 2016. Vol. 5. Iss. 4. P. 302–309. https://doi.org/10.1016/j.jppr.2016.11.008.
6. Chernousenko O., Butovsky L., Rindyuk D., Granovska O., Moroz O. Analysis of residual operational resource of high-temperature elements in power and industrial equipment. *Eastern-European Journal of Enterprise Technologies – Energy-saving technologies and equipment.* 2017. Vol. 1. No. 8 (85). P. 20–26. http://dx.doi.org/10.15587/1729-4061.2017.92459.
7. СОУ-Н МПЕ 40.17.401:2021. Контроль металу і продовження терміну експлуатації основних елементів котлів, турбін і трубопроводів теплових електростанцій. Типова інструкція. Офіц. вид. Київ: ГРІФРЕ: Міністерство палива та енергетики України, 2021. 214 с.
8. Визначення розрахункового ресурсу та оцінка живучості роторів і корпусних деталей турбін. Методичні вказівки: СОУ- Н МЕВ 40.1–21677681– 52:2011 / М. Г. Шульженко, П. П. Гонтаровський, Ю. І. Матюхін, І. І. Мележик, О. В. Пожидаєв. Київ: ОЕП «ГРІФРЕ»: М-во енергетики та вугільної пром-сті України, 2011. 42 с.
9. Черноусенко О. Ю., Риндюк Д. В., Пешко В. А. Напружено-деформований стан ротора турбіни К-1000-60/3000 при типових режимах експлуатації. *Вісник НТУ «ХПІ». Сер.: Енергетичні та теплотехнічні процеси й устаткування.* 2019. № 3 (1328). С. 4–10. https://doi.org/10.20998/2078-774X.2019.03.01.
10. Chernousenko O. Yu., Peshko V. A. Assessment of resource parameters of the extended operation high-pressure rotor of the K-1000-60/3000 turbine. *Journal of Mechanical Engineering – Problemy Mashynobuduvannia*. 2019. Vol. 22. No. 4. P. 41–47. https://doi.org/10.15407/pmach2019.04.041.14.