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CALCULATION STUDY OF THERMAL STRESSES IN THE MEDIUM-PRESSURE ROTOR OF THE K-200-130 TURBINE DURING START-UP FROM A COLD STATE

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The paper is devoted to the study of temperature and stress distribution in the medium-pressure rotor of the K-200-130 turbine, which are of considerable interest when predicting the durability of this equipment and extending its operation beyond the service life. A geometric model of the most loaded part of the rotor – from the middle of the shaft neck in the thrust bearing area to the 5th stage disc – was developed. The study of the thermal and stress-strain state of the rotor during start-up from a cold state was performed in a two-dimensional formulation using the finite element method. The non-stationary problem of heat conduction during start-up was solved. The obtained results indicate a fairly uniform thermal state during variable operating conditions. The largest temperature gradient (1200–2200 K/m) is observed at the time points from the rotor push to the synchronization of the turbine generator with the power system. After the turbine generator is loaded with up to 30 MW of electric power, a decrease in the temperature field irregularity and its gradual stabilization are observed. It was found that when operating at the nominal steam parameters, the maximum metal temperature is 508 °C in the region of the control stage and decreases when the distance from it increases. The stress-strain state of the rotor was evaluated taking into account the unevenness of temperature fields during start-up, stresses from thermal expansion, and centrifugal forces. The highest stresses are characteristic of the moment when the turbine comes to idle in the area of thermal compensation grooves of the rotor and the control gate and amount to 440–472 MPa. It is noted that these areas are the most likely zones of ring crack nucleation during turbine start-up operations. Subsequently, the stress level gradually decreases as the turbine unit reaches its rated power. It has been established that the most stressed area of the rotor during stationary operation is the area of the axial bore under the control stage and its diaphragm seal (121–134 MPa).

***Keywords**: steam turbine, rotor, start-up, thermal state, stress state, temperature gradient, stress intensity.*

Introduction

Characteristic features of the energy system of Ukraine are a certain overload of basic power generating capacities and a significant shortage of maneuverable power units. As a result, power units with a capacity of 200–300 MW designed to operate in basic and semi-basic modes are actively involved in compensating for peaks and drops in electricity consumption [1]. Taking this into account, a large number of starts and stops of these power units is observed.

In [2] it is noted that the variable mode is characterized by an uneven initial temperature field, and the rotors are characterized by the most difficult operating conditions. High temperature, intense static and dynamic loads in a corrosive environment are specific factors of steam turbine operation, which inevitably cause fatigue damage to turbine structural elements – shafts, discs, blades. Transient fluctuations and periodic heating-cooling of turbine structural elements contribute to the development of microdefects and cracks. In addition, due to the complex geometry of the turbine design elements with fillets and grooves, thermal stresses are important factors in the development of fatigue damage [3].

In addition, it is noted in [4] that cyclic stress and deformation due to temperature fluctuations in the nominal mode cause additional fatigue damage. Thus, temperature transients during the start-up phase, steady-state operation and shutdown can lead to premature failure from the point of view of allowable working time.

In [5] it is emphasized that the thermal gradient in the rotor is a key factor in the formation of the stress-strain state. Fluctuations in the steam parameters cause fluctuations in the temperature of the rotor metal, which leads to a change in the temperature gradient and the nature of the deformation of the main equipment.

According to [6], when the turbine rotor is gradually cooled from the operating temperature to the restart temperature, the outer surface cools faster than the inner surface. And this causes a high temperature gradient along the radial direction, which leads to the development of high surface tensile stresses. Stresses of the opposite nature occur when the rotor is heated from restart temperature to operating temperature.

Importantly, when operating under conditions of complex thermomechanical loading (start-up, steady-state operation, shutdown, etc.), critical structural elements, such as notches and grooves, may be susceptible to damage associated with the interaction of creep and fatigue. In addition, the increasing number of start-stops of high-capacity power plant equipment also raises concerns about the safe operation of steam turbine rotors [7].

Calculation model of the medium-pressure rotor of the K-200-130 turbine

The object of the study is the medium-pressure rotor (MPR) of the K-200-130 steam turbine, which is a single-shaft three-cylinder unit consisting of a high-pressure cylinder, a medium-pressure cylinder and a two-flow low-pressure cylinder with intermediate steam superheat and two exhausts designed for an AC generator drive. 37 power units with a capacity of 200 MW are operated in Ukraine.

The MPR (Fig. 1) of a combined type: the front part of the rotor is solidly forged from P2MA steel (25Kh1M1FA), the last four discs are mounted from 34KhNZM steel. The blades of the first seven stages are attached to the discs with T-shaped roots, the remaining four are Y-shaped. The front ring seals are made without bushings, multistep rings are made on the shaft, and the sealing segments are inserted into the vane carriers. The rear seals of the medium-pressure cylinder are made on bushings mounted on the shaft in a hot state.

Since the most loaded and high-temperature region of the MPR of the K-200-130 turbine is the regulating stage zone, its calculation model was shortened. The part of the rotor from the middle of the shaft neck in the area of the thrust bearing to the 5th stage disc is considered (marked by line *A* in Fig. 1).

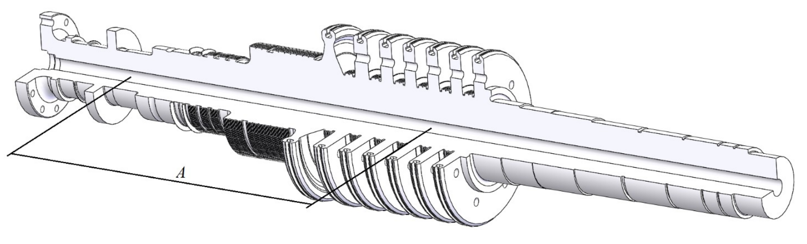


Fig. 1. Geometrical model of the MPR of the K-200-130 turbine (line A shows a part of the rotor for studying the thermal and stress state)

In order to determine the distribution of thermal stresses in a MPR, the problem of non-stationary thermal conductivity is solved, which in general can be described by the equation

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

where λ – coefficient of thermal conductivity of steel; *c* – specific heat capacity; γ –specific gravity, which is a function of temperature *T* and coordinates under initial conditions $T_0=T(r, z, \theta, 0)$; *r*, *z*, θ – cylindrical coordinates; τ – estimated time.

To solve this problem, heat transfer boundary conditions of II and III kind are set on all surfaces of the rotor. Boundary conditions of the II kind are set by the heat flux on the surface of the rotor

$$q=-\lambda\left(\frac{\partial T}{\partial n}\right)=f(r,z,\theta,\tau).$$

Boundary conditions of the III kind describe the patterns of convective heat exchange between the steam medium and the outer surface of the rotor metal

$$-\lambda\left(\frac{\partial T}{\partial n}\right)=\alpha(t_{\mathrm{med}}-t_{\mathrm{surf}}),$$

where α is the heat transfer coefficient; t_{med} is the temperature of the steam medium; t_{surf} is the temperature of the outer surface of the rotor.

Determining the boundary conditions of heat exchange is a complex task that requires a detailed calculation of the turbine flow part both in stationary and variable modes of operation. It is also necessary to establish the heat exchange conditions using special experimental equations of similarity, which are significantly different for each surface of the rotor.

This temperature distribution at any moment of time, obtained as a result of research, will be used as boundary conditions in the calculation of the rotor stress-strain state. In addition, centrifugal forces affecting the base metal, the mass of the working blades and similar factors are taken into account. The pressure of the steam medium on the outer surface of the rotor was not taken into account due to its relatively moderate value for this object, which is from 0.36 to 2.2 MPa.

Since the rotor is axisymmetric, the study of its thermal and stress-strain state was performed in a two-dimensional setting.

The system of differential equations of equilibrium for this case has the form

$$\begin{cases} \dfrac{\partial \sigma_{rr}}{\partial r} + \dfrac{\partial \sigma_{rz}}{\partial z} + \dfrac{\sigma_{rr} - \sigma_{\theta\theta}}{r} + \rho X_r = 0 \\ \dfrac{\partial \sigma_{rz}}{\partial r} + \dfrac{\partial \sigma_{zz}}{\partial z} + \dfrac{\sigma_{rz}}{r} + \rho X_z = 0 \end{cases},$$

where σ – stress components in the base metal; ρ – density of steel; X_r, X_z are components of the mass force density vector.

Hooke's law relations, written through deformations, have the form

$$\begin{cases} \sigma_{rr} = \dfrac{E}{1+\nu}\left[\dfrac{\nu}{1-2\nu}\left(\varepsilon_{rr} + \varepsilon_{zz} + \varepsilon_{\theta\theta}\right) + \varepsilon_{rr}\right] \\ \sigma_{zz} = \dfrac{E}{1+\nu}\left[\dfrac{\nu}{1-2\nu}\left(\varepsilon_{rr} + \varepsilon_{zz} + \varepsilon_{\theta\theta}\right) + \varepsilon_{zz}\right] \\ \sigma_{\theta\theta} = \dfrac{E}{1+\nu}\left[\dfrac{\nu}{1-2\nu}\left(\varepsilon_{rr} + \varepsilon_{zz} + \varepsilon_{\theta\theta}\right) + \varepsilon_{\theta\theta}\right] \\ \sigma_{rz} = \dfrac{E}{2(1+\nu)}\varepsilon_{rz} \end{cases},$$

where E is the elasticity modulus of steel; ν is the Poisson's ratio; ε are components of base metal deformation.

The calculation model of the rotor was discretized into 55 thousand finite elements of the triangular type (Fig. 2). The density of the calculation grid is significantly increased for the entire outer surface of the rotor. The size of the largest element is 15 mm, and of the smallest one – 1 mm.

In Fig. 2 characteristic areas of the study of the temperature changes dynamics and the intensity of stresses in the MPR are noted as: 1 – the shaft neck in the area of the first seal chamber on the side of the steam inlet; 2 – the thermal compensation groove between the second and third segments of the front end seal; 3 – the radial transition from the end seal segments to the flow part of the turbine; 4 – tail end attachment of the regulating stage; 5 – the fillet of the regulating stage from the side of the next stage; 6 – the axial hole of the rotor under the regulating stage.

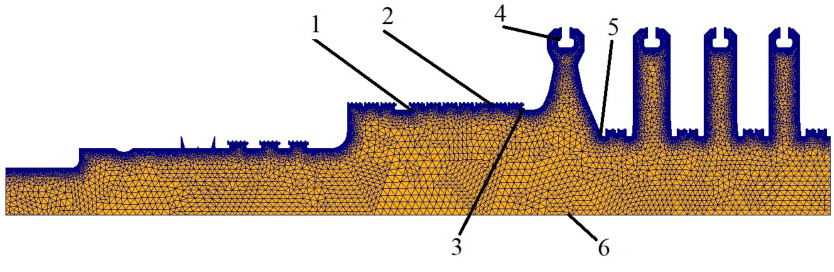


Fig. 2. A model of the MPR of the K-200-130 turbine with a finite element grid (the numbers indicate the areas of research of the temperature and stress changes dynamics)

Study of the thermal and stress-strain state of the MPR of the K-200-130 turbine when starting from a cold state of the metal

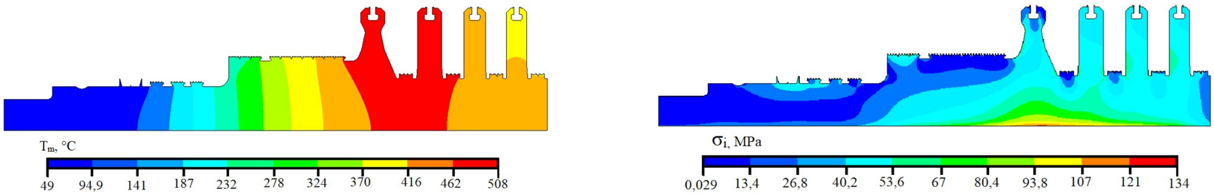
The thermal and stress-strain state of the rotor at start-up was studied in a non-stationary setting. The power unit start-up technology varies significantly depending on the initial temperature of the metal. For the K-200-130 turbine, the temperature of the metal of the steam pipes at the entrance to the high- and medium-pressure cylinder is decisive: a temperature below 150 °C corresponds to start-up from a cold state of the metal, 150–400 °C – from an uncooled state, and at a temperature above 400 °C – from a hot state [8]. For example, the distribution of temperatures and stress intensity in the rotor when operating at a nominal power of 200 MW is shown in Fig. 3.

The obtained temperature distribution shows that the maximum temperature of the metal when working in the nominal mode is 508 °C in the area of the MPR regulating stage and decreases to 370 °C in the direction of the steam and to 187 °C in the direction of the end seals as it moves away from it. The axial bore temperature under the control stage is 416 °C.

The distribution of stress intensity in the rotor when operating at a power of 200 MW is shown in Fig. 3, b. The area of the axial opening under the control stage and its diaphragm seals is the most loaded (σ_i=121–134 MPa). Also, a high level of stress (60–70 MPa) is characteristic of the step discs bodies. Significantly lower stress intensities are noticeable for the rest of the regions. At the same time, in some areas, a significant increase in the stress level during start-up operations is noticeable.

Similar distributions of temperatures and stress intensity were obtained for each moment in time of the turbine start-up mode. The total duration of the turbine start-up from a cold state according to the technology used at the Burshtyn TPP is 5.22 h (18800 s) [7]. The dynamics of temperature changes in characteristic regions of the MPR (Fig. 2) during start-up is shown in Fig. 4.

The resulting temperature distribution is fairly uniform. A smooth increase in temperature is observed during the pre-heating period of the steam pipes and the flow part until the moment of time of 4200 s. From the turbine push (4200 s) to synchronization (7400 s), there is a more rapid rise in temperature for point 4 (tail end attachment of the first stage blades) from 120 °C to 380 °C, for the rest of the points the temperature difference is somewhat smaller. After this stage, the temperature at all points rises smoothly and at the moment of time of 18800 s for point 4 it is 498 °C, for the rest of the points it is lower. Subsequently, the temperature field stabilizes to the state shown in Fig. 3, a.



a b

Fig. 3. Distribution of temperatures (a) and stress intensity (b) in the MPR when operating at the nominal mode

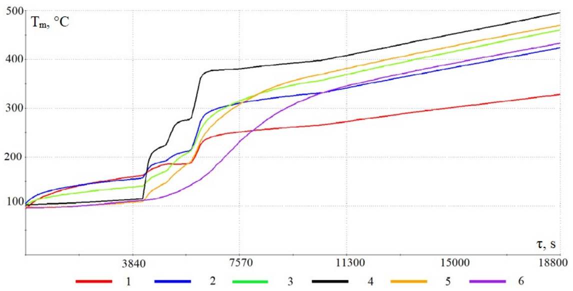


Fig. 4. Dynamics of temperature changes in characteristic regions of the MPR of the K-200-130 turbine during start-up from a cold state

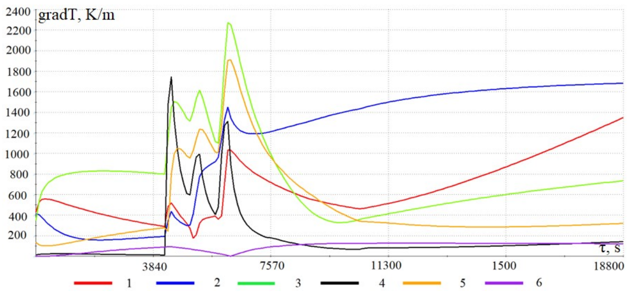


Fig. 5. Dynamics of changes in temperature gradients in characteristic regions of the MPR of the K-200-130 turbine during start-up from a cold state

The dynamics of changes in temperature gradients is of significant interest in researching the start-up modes of steam turbines (Fig. 5). The greatest non-uniformity of the temperature field is visible at the time points of 4000–7500 s for all areas of the study, except for the axial opening of the rotor (point 6 in Fig. 2). This circumstance is explained by the heating mechanism of this area – heat conduction, while other areas undergo direct convective heat exchange with steam. The highest level of temperature gradients is characteristic of the areas of the final rotor seals (points 1–3 in Fig. 2), which is associated with significant steam throttling in the seal segments. The obtained distribution of temperature gradients makes it possible to establish the periods of the greatest influence of stresses from the unevenness of the temperature field on the general complex stress-strain state of the rotor (Fig. 6).

Most of the studied areas are affected by compressive forces most of the time during the start-up (lines 1–5 in Fig. 6). During the preliminary heating of the turbine flow part, the stress level is quite moderate. However, during the rotors shock, a significant increase in stresses is observed in all the studied areas. The highest stresses are characteristic for the moment of time of 6300 s, which corresponds to the turbine coming to idle. Such significant stresses are associated with the combined action of significant centrifugal forces and significant non-uniformity of the temperature field at this moment in time (Fig. 5).

The highest stress modulus is characteristic of areas 2 (thermocompensation groove between the second and third segments of the front final seal) and 5 (the control stage flange on the side of the next stage) and are 458 and 472 MPa, respectively. In general, high stresses are observed in all the thermal grooves of the seals and the flanges of the turbine stages. These areas are the most likely zones for the initiation of annular cracks during turbine start-up operations.

Zone 6 (the axial opening of the rotor under the control stage) is always affected by tensile forces, which primarily depend on the turbine rotation frequency. The maximum stress is 220 MPa at the time of 6300 s, which gradually decreases to 129 MPa when the turbine reaches the nominal operating mode (Fig. 6).

In general, changes in stresses for given points in time have a number of common features: insignificant stresses at the stage of preliminary heating of the rotor, their rapid growth during the turbine push to idle and a gradual decrease with the completion of stabilization of the thermal field after start-up.

The obtained results regarding the temperature and stress distribution in the MPR are of considerable interest when predicting the durability of the K-200-130 turbine and extending its operation beyond the equipment resource.

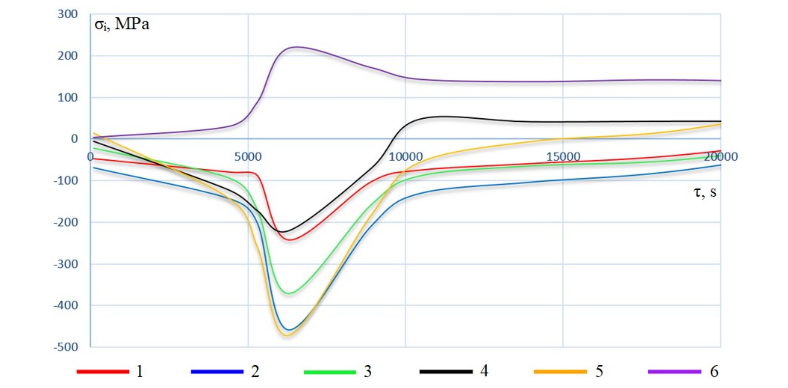


Fig. 6. Dynamics of changes in stress intensity in characteristic regions of the MPR of the K-200-130 turbine during start-up from a cold state

Conclusions

A geometric model of the most heavily loaded part of the rotor - from the middle of the shaft neck in the thrust bearing area to the 5th stage disc – has been developed. The study of the thermal and stress-strain state of the rotor during start-up from a cold state was performed in a two-dimensional setting using the finite element method. The rotor model was discretized into 55,000 finite elements of the triangular type.

It was established that the temperature distribution in the MPR is sufficiently uniform. A smooth increase in temperature is observed during the preheating period of the steam pipes and the flow part. From the turbine push to synchronization, there is a rapid increase in temperature, after which the temperature gradually increases until the moment of stabilization of the temperature field at the nominal parameters. The maximum temperature value is 508 °C and is typical for the regulating stage disc. The greatest non-uniformity of the temperature field is visible at the time points of 4000–7500 s for all areas of the study, except for the rotor axial opening. The maximum temperature gradient is characteristic for the regions of the final rotor seals, which is associated with significant steam throttling in the seal segments.

The calculation of the stress-strain state of the MPR showed that during the initial stages of start-up and preliminary heating of the turbine flow part, the stress level is quite moderate. However, during the thrust of the rotor, a significant increase in stresses is observed in all the studied areas. The highest stress modulus is characteristic of the thermocompensating grooves of the front final seal and the fillet of the regulating stage and is 440–472 MPa. These areas are the most likely zones for the initiation of annular cracks during turbine start-up operations. The zone of the rotor axial opening is always affected by tensile forces, which primarily depend on the turbine rotation frequency. The maximum stress in this area is 220 MPa when the turbine is comes to idle, which then gradually decreases to 129 MPa when moving to stationary operating modes.

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Розрахункове дослідження термічних напружень у роторі середнього тиску турбіни К-200-130 при пуску з холодного стану

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Робота присвячена дослідженню розподілу температур і напружень у роторі середнього тиску турбіни К-200-130, які становлять значний інтерес при прогнозуванні довговічності роботи даного обладнання і продовженні експлуатації на понадпарковий строк служби. Розроблено геометричну модель найбільш навантаженої частини ротора – від середини шийки валу в зоні опорно-упорного підшипника до диску 5-го ступеня. Дослідження теплового й напружено-деформованого стану ротора під час пуску з холодного стану виконано у двовимірній постановці з використанням методу скінченних елементів. Розв'язувалася нестаціонарна задача теплопровідності під час пуску. Отримані результати свідчать про достатньо рівномірний тепловий стан протягом змінних *режимів роботи. Найбільший градієнт температур (1200–2200 К/м) спостерігається в моменти часу від поштовху ротора до синхронізації турбогенератора з енергосистемою. Після навантаження турбогенератора до 30 МВт електричної потужності має місце зменшення нерівномірності температурного поля та його поступова стабілізація. Встановлено, що при роботі на номінальних параметрах пари максимальна температура металу становить 508 °С в області регулюючого ступеня і по мірі віддалення від нього зменшується. Напружено-деформований стан ротора оцінювався з урахуванням нерівномірності температурних полів під час пуску, напружень від температурних розширень і відцентрових сил. Найвищі напруження характерні для моменту виходу турбіни на холостий в зоні термокомпенсаційних канавок ротора й галтелі регулюючого і складають 440–472 МПа. Відмічено, що дані області є найбільш вірогідними зонами зародження кільцевих тріщин під час пускових операцій турбіни. У подальшому рівень напружень плавно зменшується протягом виходу турбоагрегату на номінальну потужність. Встановлено, що найбільш навантаженою зоною ротора під час стаціонарної експлуатації є область осьового отвору під регулюючим ступенем та його діафрагмовим ущільненням (121–134 МПа).*

***Ключові слова**: парова турбіна, ротор, пуск, тепловий стан, напружений стан, градієнт температур, інтенсивність напружень.*

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