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Impact analysis of a transient temperature field on the service life of the high pressure rotor of K-1000-60/3000 turbine

T. V. Nikulenkova*, A. G. Nikulenkov

National Technical University of Ukraine "Igor Sikorsky" Kyiv Polytechnic Institute, Kyiv, Ukraine

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Ensuring continued operation of NPP turbines envisages carrying out a set of works to assess the technical conditions of the turbine to detect and analyze damages, defects, identify causes and mechanisms for their occurrence and possible development. Further, the residual service life is assessed and recommendations in aging management are developed to ensure robust and safe operation of the turbine beyond the design period.

It is well-known that increasing the installed capacity of a power unit requires modernization of the flow section of the turbine high pressure cylinder, which, accordingly, will affect the service life of the rotor of the high-pressure cylinder. Therefore, one of the purposes of this paper is to investigate the impact on the service life of the high-pressure cylinder rotor of a typical high-speed turbine K-1000-60/3000. The residual life assessment of power equipment would require determining viability and damage of its base metal. Typical degradation mechanisms of steam turbine equipment include long-term strength reduction and low cycle fatigue accumulation. Intensity of their impact is determined by a numerical examination of equipment thermal (TS) and stress strain states (SSS) for standard operation modes. To perform a numerical examination of the stress strain state would require solving a thermal conductivity boundary problem in quasi-stationary (for normal operation modes) and nonstationary modes (for transients). It is convenient to solve such problems of mathematical physics through discretization of the calculation object using the finite element method.

Keywords: steam turbine, service life, stress strain states, high-pressure cylinder, rotor, initial and boundary conditions, CFD-codes, turbine K-1000-60/3000

INTRODUCTION

With extended operation of power plant equipment, the definition of its service life has changed. After the fleet life is reached, an in-depth diagnostic is carried out for specific nodes of the electric power installation, including:

- analysis of its operating conditions;
- measurement of the actual component geometry;
- examination of steel structure, its properties and accumulated damage;
- non-destructive testing and calculated estimate of stress state and residual service life of components.

Based on the results of performed studies an individual residual service life is established for a specific component of the power equipment.

The decision-making algorithm regarding capability and conditions for equipment operation throughout the individual service life can be described as follows:

- assessment of actual operating conditions for the entire period of component use;
- conducting repeated strength analysis based on the refined operation data;
- defectoscopy (visual inspection, ultrasonic testing, X-ray examination, magnetic particle testing, lab examinations of steel samples);
- assessment of the expired service life and forecasting further operation after defining equipment conditions.

This approach allows combining calculation methods, results of steel examination and samples tests, which complement each other.

LITERATURE REVIEW

An important task of the power sector is to ensure maintaining electricity production at the attained level until new generating capacities are commissioned. That is why lifetime extension of operating power units is one of the most efficient ways of ensuring return on investments in the power sector.

The U.S.A. developed a Plant Life Extension Programme for TPPs and NPPs way back in the 1980s. Implementation of the programme at thermal power units is 10 times cheaper than building a new unit.

Since the 1980s and until the present day a range of scientific and research studies have been conducted with relevant operating experience accumulated which allowed a twofold increase of the service life of steam turbines. The approaches have been elaborated for service life extension of

* To whom all correspondence should be sent:
E-mail: tetyana.nikulenkova@gmail.com

equipment being in different stages of its life expiration.

In previous times, when performing calculated estimates of the service life of steam turbines authors of the methodology guidelines for assessment of individual life of steam turbines [1, 2] did not take into account the engineering changes in power equipment components formed during the entire previous period of steam turbine operation, specifically repair and renewal changes of rotor structure.

Later, the National Technical University of Ukraine "Igor Sikorsky Kyiv Polytechnic Institute" conducted a number of scientific and research studies. These studies were devoted to the development of a methodology for calculated assessment of a low cycle fatigue of the steel of high-temperature casings and rotors of К-200-130 steam turbines using ANSYS and COsikoSMOS. These software suits allow building a diagnostic system for the stress strain state (SSS) of steam turbines [3, 4]. A comprehensive approach was proposed to extending operation of steam turbines while maintaining the required safety level.

A similar approach is used by specialists from different countries of the world – conducting a calculation analysis while taking maximum account of research results and then defining the most stressed components of plant power equipment. Thereafter, these components are checked using instrumentation monitoring methods [5, 6].

Nowadays, particularly relevant are scientific and technical, as well as economic studies aimed at justifying the capability and economic feasibility of NPP operation beyond its design life as an alternative to decommissioning. Economic expediency of certain actions is evaluated based on operation track record and considering equipment repairs and replacements important for safety. This should properly reflect the present requirements and recommendations.

When forecasting service life of NPP units the decisive factors are ageing processes undergoing in equipment materials and limiting its life. The main mechanisms defining degradation of steel properties of NPPs are considered to be: radiation embrittlement of RPV steels, fatigue damages affected by thermal fatigue and mechanical loadings, primary water stress corrosion cracking, thermal ageing, corrosion.

The estimates obtained by different authors vary from 15 to 40 % of the cost of a new power plant. It is a rather significant element of costs, even if considering it at minimum.

RESULTS AND DISCUSSION

Approaching the end of the established life of an NPP equipment poses a number of challenges to nuclear industry:

- to increase the installed capacity while maintaining the required safety level of operating power units using the built-in engineered margins of operating units along with the ever increasing pace of science and technology development, and taking due account of international practice;
- to carry out a range of works and upgrades to ensure operation of power units beyond their design life while maintaining the required safety level.

The range of works to be carried out when assessing the service life of the critical elements of the turbine is comprised of several phases.

In the fist phase, the 3-D analogues of the turbine machine elements were built based on the results of the technical audit and conclusions of the visual inspection, when different types of damages are localized in the geometrical model of a turbine element in the form of steel samples of differenet shape. Such approach allows to bring the calculation model of the steam turbine element close to real conditions after continiuous operation.

The next phase is to calculate initial and boundary conditions (using CFD-codes or criterial equations as specified in [7, 8]) and determine a nonstationary temperature field in the solid critical element for further calculation of thermal load.

It starts with solving the non-stationary thermal conductivity equation and assigning boundary conditions for heat transfer on the surfaces of the rotor based on 2-D and 3-D geometrical models. The non-stationary thermal conductivity equations are given below:

$$\mathrm{div}[\lambda(T)\ \mathrm{grad}(T)] = c(T)\ \gamma(T)\ dT/d\tau \qquad (1)$$

where λ – coefficient of thermal conductivity, c – specific heat capacity, γ – specific weight, which are functions of temperature and coordinates under initial conditions $T_0 = T(x, y, z, 0) = f_0(x, y, z)$.

The third phase involves using the ANSYS package to determine the stress strain state of the high pressure cylinder rotor considering its complex spatial geometry, damages over ther period of operation, and repair and renewal changes of the design geometry [9-12].

The outcome is calculations of equivalent elastic strain, von Mises equivalent strain, principal stresses, taking into account the effect of the centrifigual forces, temperature and steam pressure loads during start-up of the К-1000/60-3000 turbine from cold (CS), warm (WS) and hot states (HS).

1. Equivalent elastic strain shall be calculated by the formula:

$$\varepsilon_e=(1/1+\nu')(1/2[(\varepsilon_1-\varepsilon_2)^2+(\varepsilon_2-\varepsilon_3)^2+(\varepsilon_3-\varepsilon_1)^2])^{1/2} \quad (2)$$

where ν' – the effective Poisson's ratio defined as: i) the Poisson's ratio at the relevant temperature of the considered body for elastic and thermal deformations; ii) 0.5 for plastic deformations.

2. Von Mises equivalent strain shall be calculated by the formula:

$$\sigma_e=[((\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2)/2]^{1/2} \quad (3)$$

3. Principal stresses

It is well-known from the theory of elasticity that an infinitely small volume of material in the arbitrary point of the solid body and inside it can be rotated so that there remain only normal stresses, and all shear stresses are equal to zero. The three remaining normal stresses are called principal stresses (see Fig. 1).

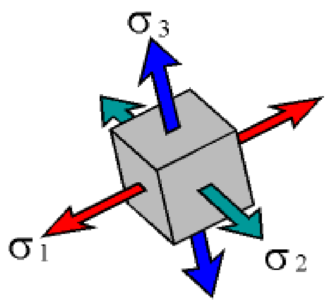

Fig. 1. Principal stresses

Principal stresses are always arranged as follows: $\sigma_1 > \sigma_2 > \sigma_3$, wher σ_1 – maximum principal stress, σ_2 – middle principal stress, σ_3 – minimum principal stress.

To calculate transient temperature fields for CS, WS and HS the boundary conditions were determined for temperature distribution in the rotor (t=f(x,y) at the time τ=0). The boundary conditions are presented in Figs. 2-4.

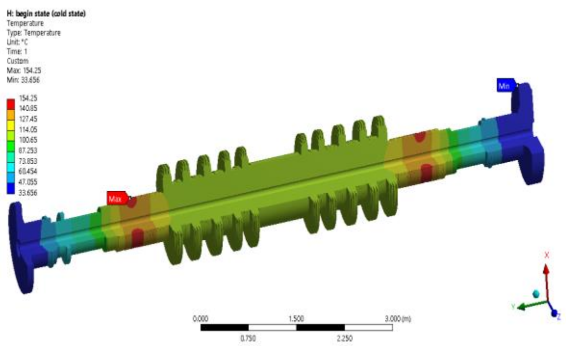

Fig. 2. Initial temperature distribution in the rotor during cold start-up

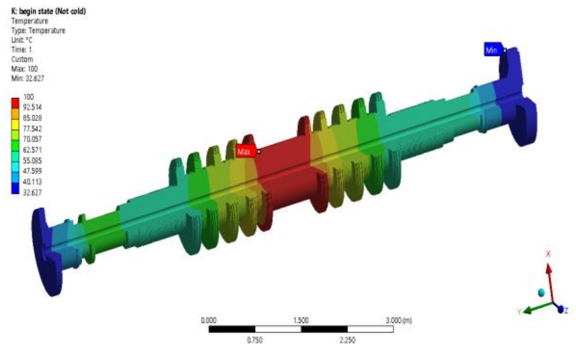

Fig. 3. Initial temperature distribution in the rotor during warm start-up.

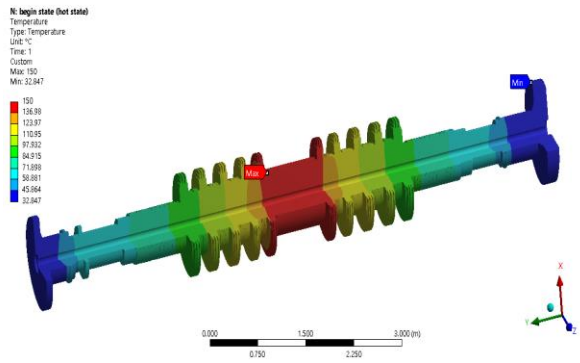

Fig. 4. Initial temperature distribution in the rotor during hot start-up

The calculations determine principal stresses and intensity of stresses over the entire life corresponding to start-up and stationary operating conditions in all division points of hight temperature elements of the stream turbine.

The results of stress and deformation calculations under cold start-up of the K-1000/60-3000 turbine are provided below.

Figs. 5-8 show the change of maximum stress (equivalent (von-Mises) stress, principal sress) during the cold start-up.

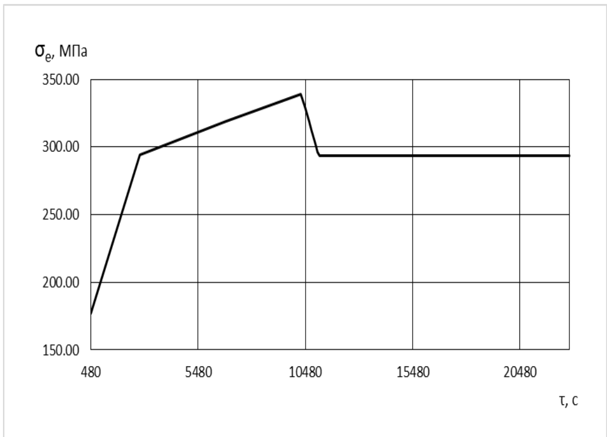

Fig. 5. Time change of maximum equivalent stress (von Mises) during cold start-up

The maximum equivalent stress is 338.9 MPa reached at 10260 s. The stress peak shifts from end seals (during warming-up of the turbine due to steam directed onto seals) to the first stage disk and further to the coupling when reaching nominal conditions.

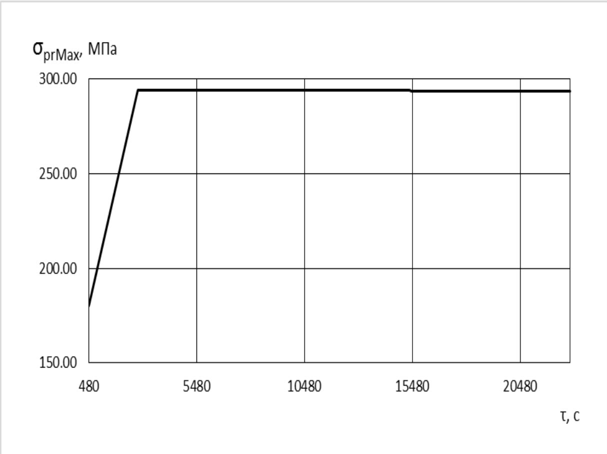

Fig. 6. Time change of the maximum value of maximum principal stress during cold start-up.

The maximum value of the maximum principal stress is 294.16 MPa reached at 2760 s.

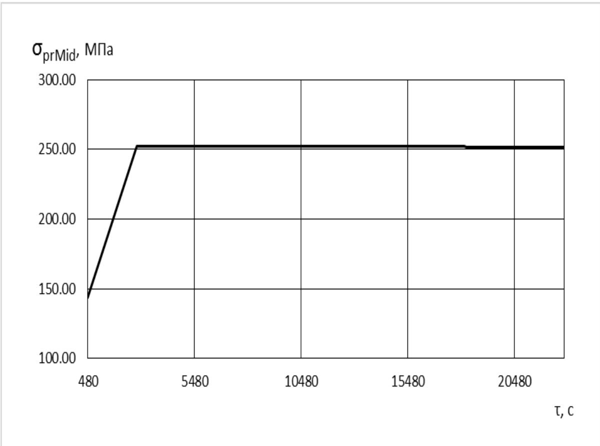

Fig. 7. Time change of the maximum value of middle principal stress during cold start-up.

The maximum value of middle principle stress is 252.05 MPa reached at 2760 s.

The maximum value of minimum principal stress is 109.09 MPa reached at 6780 s.

Fig. 9 illustrates the time change of maximum equivalent elastic strain during cold start-up.

As it can be seen from Fig. 9, the maximum equivalent elastic strain is 1.76E-03 m/m reached at 10260 s.

Figs. 10, 11 show the stress strain state of the K-1000/60-3000 steam tubine at the starting point (480 s) and at the end of the calculation (22740 s). According to the presented calculations the maximum stress is 338.9 MPa reached at 10260 s. The stress peak shifts from end seals to the first stage disk and further to the coupling when reaching nominal conditions.

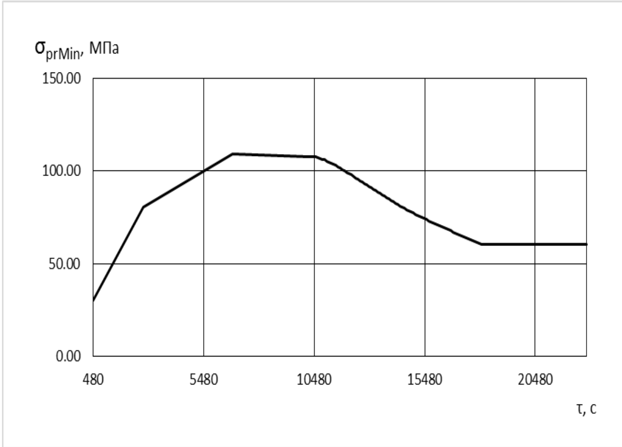

Fig. 8. Time change of the maximum value of minimum principal stress during cold start-up.

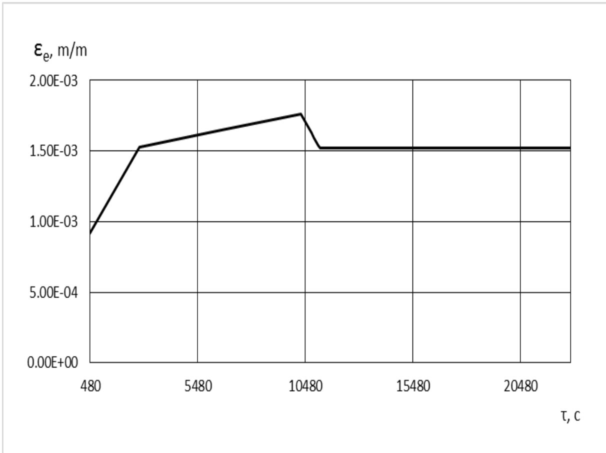

Fig. 9. Time change of maximum equivalent elastic strain during cold start-up.

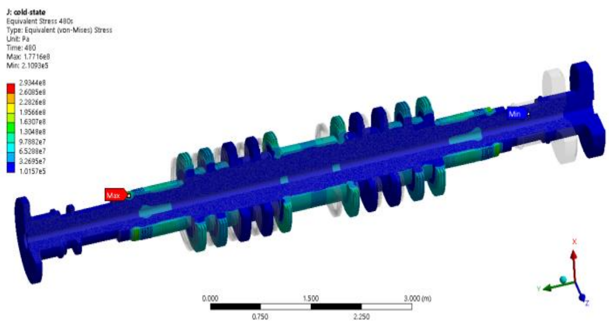

Fig. 10. Stress strain state of the K-1000/60-3000 steam turbine at the starting point (480 s).

The calculation results of the stress strain state of the high-temperature elements of the steam turbine obtained using ANSYS in different spatial settings (2-D and 3-D) were compare between each other and with the results obtained by other authors [13].

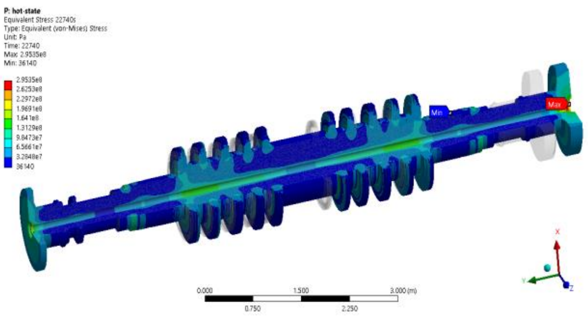

Fig. 11. Stress strain state of the K-1000/60-3000 steam turbine at the end of the calculation (22740 s)

The final phase implies a development of a methodological approach to the calculaton of the low cycle fatigue with account of changes in the stress strain state of the K-1000/60-3000 turbine elements for optimization of strength margins by number of cycles and deformations [14-17].

The described methodology was used to develop a software for numerical analysis of the residual service life of steam turbines used at thermal and nuclear power plants.

The individual residual service life is defined as the time margin of potential operation of equipment after inspecting its technical conditions (or repair) during which all its main technical, operation and safety indicators shall meet the requirements of scientific and technical documentation.

The individual residual service life should be evaluated in three main phases:

- evaluation of turbine's technical conditions by controlling parameters with conclusions;
- evaluation of turbine's technical conditions by strength parameters with conclusions;
- evaluation of the residual service life of a turbine by life parameters with conclustions.

The possibility of forecasting the residual service life is ensured under the conditions as follows:

- parameters defining technical conditions of the equipment are know;
- criteria of the boundary conditions of the equipment are known;
- there is possibility of periodical and continious inspection of technical condition parameters.

CONCLUSIONS

1. The set of works to estimate the service life of the critical elements of the K-1000/60-3000 turbine was described in phases.

2. The calculations were made for equivalent elastic strain, von Mises equivalent strain, principal stresses, taking into account the effect of the centrifugal forces, temperature and steam pressure loads during start-up of the K-1000/60-3000 turbine from cold, warm and hot states.

3. The maximum stress was calculated as 338.9 MPa reached at 10260 s during cold start-up of the K-1000/60-3000 steam turbine.

4. The maximum equivalent elastic strain was calculated as 1.76E-03 m/m reached at 10260 s during cold start-up of the K-1000/60-3000 steam turbine.

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