

Estimation of resource capabilities of the NPP turbine unit under the primary frequency control of the current in the power system

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Abstract. The purpose of the article is to estimate the life cycle on the example of NPP with the K-1000-60/1500 turbine and with the VVER-1000 reactor under cyclic load changes using the primary frequency control. The criterion for estimating the life cycle is the rate of the fatigue crack growth of the most loaded rotor element. The main requirements for the NPP power units under conditions of involvement in the primary frequency control are provided. The cyclic loading of the NPP turbine unit blades under conditions of maintaining the power reserve due to a reduction in the load by 2% by throttling the steam, and in line with the requirements for participation in the primary frequency control by 8% of the nominal capacity is substantiated. At the same time, the cyclic mechanical and thermal stresses are determined for the HPC and the LPC rotor blades. In this regard, it is shown that the first stage blades of the HPC rotor during unloading by 8% of the nominal capacity of the most cyclically loaded. The loading intensity and the rate of the fatigue crack growth of the first-stage blades of the HPC rotor are determined on the basis of the cyclic crack-resistance methodology and the Paris equation. It is shown that the rate of the fatigue cracks growth is negligible which determines their long-term life cycle.

1. Introduction

One of the requirements of power systems is the need to control the frequency of the current and keep it within the acceptable limits. For this purpose, hydropower plants and specially selected thermal power plants are involved. According to the order of JSC SO UPS dated August 19, 2013 No. 314, the main requirements for the primary frequency control by the NPP units were formulated.

The frequency should be within 50 ± 0.2 Hz and at least 95% of the day time and not exceed the maximum permissible 50 ± 0.4 Hz, which is due to the standardization requirements for operational dispatch regulation in the power industry. At the same time, the means of the secondary control together with the primary frequency control should ensure that the current frequency be kept within 50 ± 0.05 Hz (normal level), and within 50 ± 0.2 Hz (allowable level) with the recovery of the normal frequency level within no more than 15 minutes.

Traditionally, nuclear power plants are used in the base part of the load curve. This is facilitated by two important circumstances:

1. Large capital investments, low cost of the generated electricity compared to other thermal power plants due to low proportion of the fuel component.



2. Technical challenges with unloading and subsequent power ascension in certain periods of the fuel cycle at nuclear power plants, which are caused by xenon poisoning of the reactor core (“iodine well”). In addition, the base-load operation condition stabilizes the NPP reliability indexes at a sufficiently high level and ensures a long life cycle of expensive equipment.

It should be noted that involvement of NPPs in the primary frequency control of the current in the power system is associated with the possible frequent load changes.

The maneuverability of the NPP generating equipment should meet the following requirements for participation in the primary frequency control:

1) a guaranteed participation of the generating equipment in the primary frequency control should be ensured by implementing the required primary power within the control range:

- for the load of up to 2% of the nominal electric capacity of the power unit;
- for the unloading up to 8% of the nominal electric capacity of the power unit;

2) in case of an abrupt frequency deviation, which necessitates realization of the primary capacity within the indicated ranges, it should be ensured:

- implementation of at least half of the required primary capacity within no more than 10 sec.;
- implementation of all the required primary capacity within no more than 2 minutes.

Thus, the dynamics of the changes in the primary capacity of the NPP unit at the maximum required primary capacity per load $\Delta P_p = 2\% P_{nom}$ should be no worse than $1\% P_{nom}$ in 10 sec. and $2\% P_{nom}$ in 30 sec. in accordance with the requirements; at the maximum required primary capacity for the unloading $\Delta P_p = -8\% P_{nom}$, the dynamics of the change in the primary capacity of the power unit should be no worse than $-4\% P_{nom}$ in 10 sec. and $-8\% P_{nom}$ in 120 sec.

In this regard, under participation of NPPs in the primary frequency control these requirements determine the daily operation of a turbine unit with the cyclic load changes, as well as cause inefficient part-load operation of the power unit to ensure the maintenance of the required capacity margin by 2% of the nominal capacity [1]. In this case, there is a need for additional unloading up to 8% of the nominal capacity during the periods of increasing current frequency.

2. Methods for assessing the life cycle of the turbine unit under conditions of frequency regulation

A cyclical change in the pressure of the working fluid occurs in the flow path of the turbine due to the changes in the steam consumption and its throttling in the steam admission under participation of the NPP units in the primary frequency control. The impact of steam on the blade causes a resultant force that can be decomposed into circumferential and axial components (figure 1) [2].

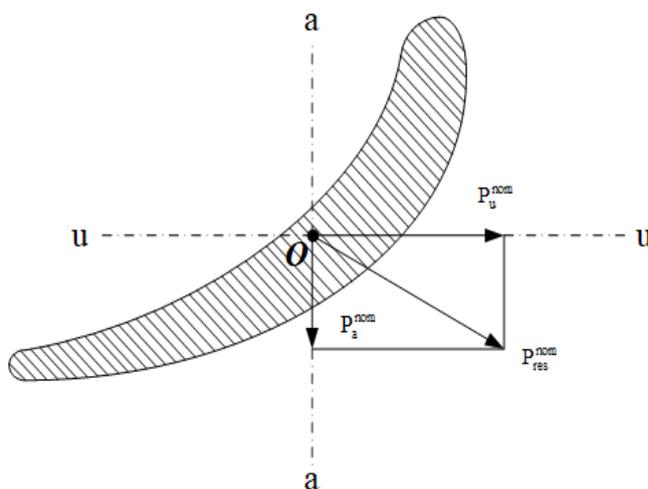


Figure 1. The forces that bend the blade in the nominal mode: P_u^{nom} is the circumferential force in the nominal mode, N; P_a^{nom} is the axial force in the nominal mode, N; P_{res}^{nom} is the resultant force in the nominal mode, N; and O is the gravity center of the profile.

The working blades experience a cyclical change in the resultant force under conditions of the cyclical changes in the pressure of the working fluid. It should be noted that the cyclic loading causes the growth of fatigue cracks [3-12]. In this regard, the rotating blades in the NPP turbine are subjected to the cyclic effects of the resulting effort. A change in the steam pressure leads to a difference in the resultant force affecting the blade, which is defined by the expression, N:

$$\Delta P_{res} = P_{res}^{nom} - P_{main\ res}^{derate}, \quad (1)$$

where P_{res}^{nom} , $P_{main\ res}^{derate}$ is the resultant force from the vapor pressure in the nominal mode, and main resultant force in the part-load operation mode, respectively, N.

The resultant force exerting on the blade in the nominal mode is determined by the expression, N [2]:

$$P_{res}^{nom} = \sqrt{P_u^{nom^2} + P_a^{nom^2}}, \quad (2)$$

where P_u^{nom} is the circumferential force in the nominal mode, N; and P_a^{nom} is the axial force in the nominal mode, N.

The circumferential force exerting on the blade in the nominal mode, N [2]:

$$P_u^{nom} = \frac{G_{nom}}{\varepsilon z_2} (c_{1u} - c_{2u}), \quad (3)$$

where G_{nom} is the nominal steam consumption through turbine stage, kg/s; ε is the partial admission ratio; z_2 is the number of blades; and c_{1u} , c_{2u} is the circumferential component of the steam release rate from the nozzle vanes channel and from the rotating blades channel, respectively, m/s.

The axial force exerting on the blade in nominal mode, N [2]:

$$P_a^{nom} = \frac{G_{nom}}{\varepsilon z_2} (c_{1a} - c_{2a}) + (P_1^{nom} - P_2^{nom}) t_{2opt} l \quad (4)$$

where c_{1a} , c_{2a} is the axial component of the steam release rate from the nozzle vanes channel and from the rotating blades channel, respectively, m/s; P_1^{nom} , P_2^{nom} is the vapor pressure in front of the rotating blade and behind the rotating blade in nominal mode, Pa; t_{2opt} is the optimal pitch of rotating blade, m; and l is the blade height, m.

The expressions (3) and (4) will take the following form under conditions of a reduction in the load by throttling the steam. The circumferential force exerting on the blade in the part-load operation mode, N [2]:

$$P_u^{derate} = \frac{G_{derate}}{\varepsilon z_2} (c_{1u} - c_{2u}), \quad (5)$$

where G_{derate} is the steam consumption through the turbine stage in the part-load operation mode, kg/s.

The axial force exerting on the blade in the part-load operation mode, N [2]:

$$P_a^{derate} = \frac{G_{derate}}{\varepsilon z_2} (c_{1a} - c_{2a}) + (P_1^{derate} - P_2^{derate}) t_{2opt} l \quad (6)$$

where P_1^{derate} , P_2^{derate} is the vapor pressure in front of the rotating blade and behind the rotating blade in the part-load operation mode, Pa.

In accordance with the Flugel formula, a proportional cyclical change of pressure in the working fluid within the flow path of the turbine leads to a cyclic temperature change during the cyclic reduc-

tion in the load by throttling the steam with the subsequent loading. This causes the cyclic thermal stresses. Using the Ansys software package, the thermal stresses are represented by three components, including the normal thermal stresses in the direction of three axes beginning from a common point. According to figure 1, the two components of the normal thermal stresses ($\Delta\sigma_{therm}^u$ and $\Delta\sigma_{therm}^a$ in figure 2) are codirectional with the circumferential and axial forces, respectively. The third component of the normal thermal stresses ($\Delta\sigma_{therm}^{along}$ in figure 2) is directed along the blade in the direction perpendicular to the plane of the circumferential and axial forces. In this regard, when evaluating the life cycle, the cyclical nature of the thermal stresses can be taken into account by adding $\Delta\sigma_{therm}^u$ and $\Delta\sigma_{therm}^a$.

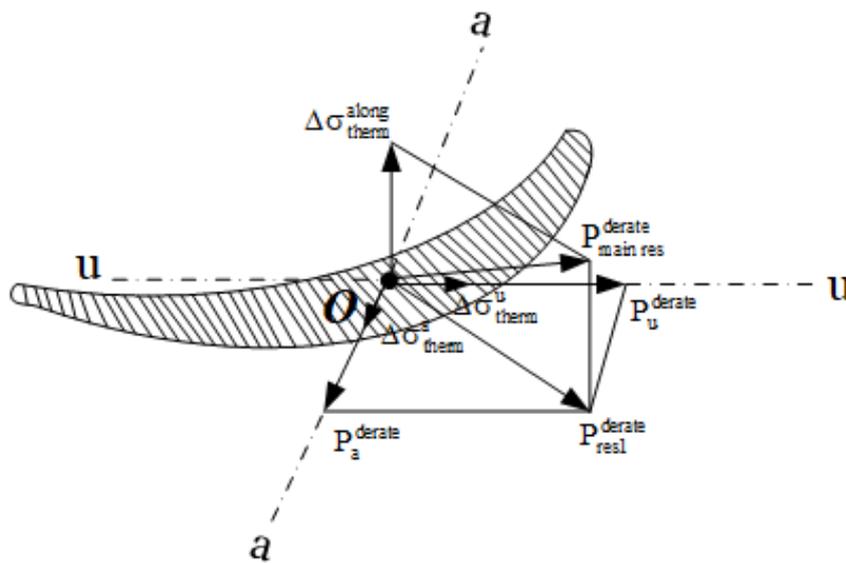


Figure 2. The forces that bend the blade in the part-load operation mode: P_u^{derate} is the circumferential force in the part-load operation mode, N; P_a^{derate} is the axial force in the part-load operation mode, N; P_{res1}^{derate} is the first resultant force in the part-load operation mode, N; $\Delta\sigma_{therm}^{along}$ is the thermal stresses in the longitudinal direction relative to the blade; and $P_{main res}^{derate}$ is the main resultant force in the part-load operation mode, N.

The circumferential and axial forces in the cyclic unloading mode taking into account thermal stresses are given as:

$$P_u^{derate.therm} = \frac{P_u^{derate}}{f} + \Delta\sigma_{therm}^u \quad (7)$$

$$P_a^{derate.therm} = \frac{P_a^{derate}}{f} + \Delta\sigma_{therm}^a, \quad (8)$$

where f is the cross-sectional area of blade profile, m^2 .

The cross-sectional area of the blade profiles by the turbine stages was determined by the trapezium method.

Thus, according to figure 2, the first resultant force which affects the blade in the cyclic mode a reduction in the load by throttling the steam, taking into account thermal stresses is calculated as follows, N:

$$P_{res1}^{derate} = \sqrt{P_u^{derate.therm^2} + P_a^{derate.therm^2}}. \quad (9)$$

In this case, after determining the first resultant force according to (9), taking into account the third component of normal thermal stresses ($\Delta\sigma_{therm}^{along}$ in figure 2) effecting along the blade, the main resultant force is determined, N:

$$P_{main\ res}^{derate} = \sqrt{P_{res1}^{derate^2} + \Delta\sigma_{therm}^{along^2}}, \quad (10)$$

where $\Delta\sigma_{therm}^{along}$ is the thermal stresses in the longitudinal direction relative to the blade, MPa.

With a reduction in the load by throttling the steam, the cyclic temperature changes in the working fluid across the blades of the turbine K-1000-60/1500 for the rotor of high-pressure cylinder, when average, was 2°C, and for a rotor of a low pressure cylinder was 1.6°C. In this case, the components of normal thermal stresses on average amounted to no more than 0.5 MPa for the rotor blades of the HPC and LPC. Thereby, the influence of the thermal stresses on the rate of the fatigue crack growth is negligible. Thus, the greatest pressure change in the flow range occurs when the capacity decreases with the unloading up to 8% of the nominal capacity than when working with the unloading of 2% for the K-1000-60/1500 turbine. According to calculations, the greatest difference in the resultant force, in line with (1) on the blades of the first stage of the HPC rotor was 0.92 MPa when a reduction in the load by 8% by throttling the steam, and 0.4 MPa when working with unloading by 2%. In this regard, based on modeling the steam flow in the turbine stages using the Ansys software package [13] (figure 3), we obtained the field of cyclic thermal stresses arising in the root section of the blade profile for the most loaded blade of the first stage of the HPC when a reduction in the load by 8% of the nominal capacity by throttling the steam. It was determined that the greatest tensile thermal stresses occur in the area of the entrance edge.

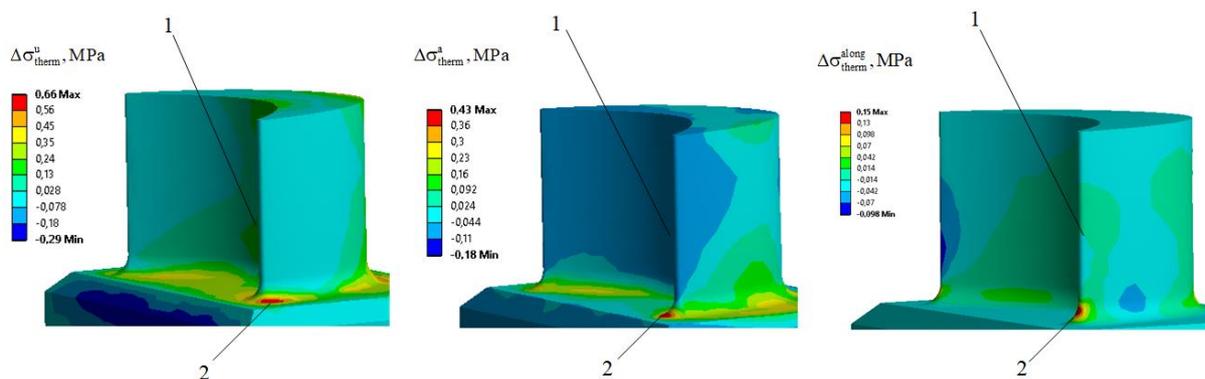


Figure 3. The field of the cyclic thermal stresses in the root section of the blade profile of the first stage of the high-pressure cylinder rotor: 1 – the entrance edge; 2 – the area of the greatest cyclic tensile thermal stresses.

According to (1), the difference in the resultant force is gradually reduced for the remaining stages of the turbine flow path along the steam expansion and reaches 0.1 MPa or less for the blades of the low-pressure cylinder stages. It should be noted that the change in steam temperature in the turbine flow range is practically absent when unloading at 2%; therefore, the cyclic change in the thermal stresses does not occur.

The difference in the resultant force (1) affecting the blades does not lead to appearance of the corresponding changes in the radial and tangential stresses in the turbine disc hub based on the strength analysis performed to estimate the radial and tangential stresses in the turbine rotor disks in nominal mode, and in the part-load operation mode based on the method [2], which indicates maintaining the necessary tension.

The basis for determining the maximum number of cycles to failure, and estimating the resource costs for the steam turbine blades is a flat body model with an applied tensile force along perpendicular to the fatigue crack [14, 15].

The value of the stress intensity coefficient is determined by the expression, $\text{MPa}\sqrt{\text{m}}$ [7,14]:

$$K = \sigma\alpha\sqrt{\pi l} \quad (11)$$

where σ is the cyclic stress, MPa; α is the nondimensional coefficient that takes into account the geometric factor and the nature of the stress distribution; and l is the threshold (minimum) crack length, m.

In this case, the magnitude of the stress intensity coefficient of the cycle, $\text{MPa}\sqrt{\text{m}}$ [7,14]:

$$\Delta K = K_{max} - K_{min} , \quad (12)$$

where K_{max} and K_{min} are determined at the corresponding maximum and minimum rate of the cycle stress (σ_{max} and σ_{min}).

According to (2), K_{max} is determined based on the resultant force from the vapor pressure on the blade in the nominal mode (P_{res}^{nom}) in relation to the sectional area of the blade profile. According to (10), K_{min} is determined based on the resultant force from the vapor pressure on the blade, taking into account the thermal stresses in the cyclic unloading mode ($P_{main\ res}^{derate}$) in relation to the sectional area of the blade profile.

It is obvious that intensity of stresses generally occurs where there is a stress difference in the structural elements of the equipment, and this difference is basically caused by a change in either the operating steam pressure or temperature, or both.

According to the fatigue failure diagram [7], an increase in the stress intensity ΔK leads to the fact that the rate of the fatigue crack growth tends from the threshold section to the section with an average amplitude the rate of the fatigue crack growth receives a greater increment per cycle, and the effect of the loading frequency on the crack closure decreases. At the small ΔK , when the growth rate of the fatigue crack is in the region of the threshold section and is very small at the same time, the influence of the loading frequency can have a noticeable effect on its closure. Increasing the frequency can close the fatigue crack and slow down its growth, and declining the frequency will accelerate its growth.

The loading frequency is understood as the number of loading cycles in relation to the time interval over which they occur and is measured in Hz. Under conditions of daily participation in the primary frequency control, the time interval is 24 hours by the example of the received number of adjustments (N) from 1 to 6 per day. Under these conditions, the frequency of cyclic loading is very small and is about 10^{-5} Hz.

Taking into account thermal stresses, the ΔK value was much less, than $1 \text{ MPa}\sqrt{\text{m}}$ for the most loaded blades of the first stage of the HPC rotor. The rate of the fatigue cracks growth was determined in accordance with the Paris equation, mm / cycle:

$$v = \frac{dl}{dN} = C(\Delta K)^n , \quad (13)$$

where C and n are characteristics of cyclic crack resistance of steel.

Fatigue crack length, mm:

$$l_N = l_0 + \nu N_{cycles}^{max} \leq 0,1 \dots 1, \quad (14)$$

where l_0 is the threshold (minimum) value of the crack length, mm (accepted to be equal to 0.002 mm); N_{cycles}^{max} is the maximum number of cycles to failure; and 0,1-1 is the critical value of the fatigue crack length for the steels with a tensile strength of up to 1500 MPa.

Based on (14), the limit number of the loading cycles (N_{cycles}^{max}) is determined by the selection method from the condition for the fatigue crack to reach the lower limit of the critical length (0.1 mm) with a known rate of the fatigue crack growth.

It is accepted that the most loaded blades of the first stage of the HPC rotor are made of the stainless steel 13Cr11H2W2MoV. According to the experimental data, this steel is used for the manufacture of turbine blades. The coefficients C and n in (13) are taken $4.59 \cdot 10^{-11}$ and 4.56, respectively. The rate of the fatigue crack growth was $2.2 \cdot 10^{-23}$ mm / cycle. The maximum number of cycles to failure (N_{cycles}^{max}) is significantly large (over 10^6 cycles) due to the negligibly low rate of the fatigue crack growth (13). Thus, it shows a negligible damage to the most loaded blades of the first stage of the HPC rotor, which determines their long life cycle.

3. Conclusions

Using the NPP units in the primary frequency control is associated with a decrease in the capacity of the steam turbine when operating in the standby mode due to the throttling of steam consumption and subsequent capacity catch up. This causes the cyclical changes in the flow rate, pressure, and steam temperature in the turbine stages, which is associated with the cyclic mechanical and thermal stresses. The cyclic load changes cause the fatigue cracks growth. On the example of the turbine K-1000-60/1500, and using the method of strength calculation of the turbine blades it is shown that the cyclic change in the load is most susceptible to the working blades of the first stage of the HPC rotor. In this regard, the loading intensity of the first stage blades of the HPC rotor is determined using the methodology for estimating the cyclic crack resistance. It is also shown that the rate of the fatigue crack growth within the blade is negligible, which determines their long life cycle.

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