Evaluation of the operating resource of the most loaded rotor element of the additional steam turbine with steam-hydrogen overheat of the working fluid at a nuclear power station

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Abstract. The operation of a nuclear power plant with a hydrogen energy complex and a constantly operating low capacity additional steam turbine makes it possible to improve the reliability of the power supply to the needs of a nuclear power plant in the face of major systemic accidents. In this case, the additional steam turbine is always in operation. This determines the alternation of the operating conditions of the additional steam turbine, and, at the same time, the alternation of the loads attributable to the rotor, which affects its working life. The aim of the article is to investigate the effect of cyclic loads on the number of cycles before the destruction of the most important elements of the rotor of an additional steam turbine due to the alternation of operating conditions when entering the peak load and during unloading at night. The article demonstrates that the values of the stress range intensity index for the most important elements of the rotor of an additional steam turbine lie in the area of the threshold values of the fatigue failure diagram. For this region, an increase in the frequency of loading is associated with the phenomenon of closure of the fatigue crack and, as a consequence, a possible slowing of its growth. An approximate number of cycles before failure for the most loaded element of the rotor is obtained.

1. Introduction

An essential increase in number of nuclear power stations in the power systems of the European part of country according is suggested as a part of the nuclear power engineering development program. Nuclear power engineering is one of the country’s strategic goals according to Russian Energy Program until 2035. The first paragraph after a heading is not indented.

Traditionally pump-storage power plants are used for adjusting nuclear power plant load. Pump-storage power plants require special natural conditions and their construction near nuclear power plants is not possible. Therefore, hydro pumped power plants are supplied with electricity from power system. In this case the nighttime electricity tariff exceeds the cost of power generated by a nuclear power plant. It increases the cost of electricity generated by a pump-storage power plant. Therefore it is necessary to work out new competitive ways of electricity generation. Energy complex using hydrogen fuel can be a solution. An energy complex using hydrogen fuel can be constructed near a nuclear power plant and produce power at the cost value. Hydrogen and oxygen are produced by means of water electrolysis during the period of nighttime minimal electricity consumption. Then hydrogen
and oxygen are accumulated in the storage system. Hydrogen and oxygen are supplied to the combustion chamber which disposition in steam cycle in period of peak electric load. During peak hours of electrical load, hydrogen and oxygen are fed to the combustion chamber in the cycle of a steam turbine plant with the help of booster compressor installations with increasing the rated power of the main turbine unit due to steam-hydrogen overheating of fresh steam [2-15] or with displacement of the heating steam for reheat using the steam generated by hydrogen oxidizing with the help of displaced steam in the additional low capacity steam turbine.

The combination of a nuclear power plant with a constantly operating additional low capacity steam turbine allows not only to increase the maneuverability of the nuclear power plant, to obtain additional peak or semi-peak capacity but also to increase the reliability of the power supply to the NPP's own needs in case of major system accidents. In case of an emergency shutdown of the reactor and a blackout at a nuclear power plant, all steam produced by residual heat will be sent to an additional steam turbine to generate electricity to support its own needs.

Figure 1 presents the scheme demonstrating the NPP operation with the use of a hydrogen-fueled energy complex and an additional steam turbine. The turbine consists of a forged high pressure rotor and a low pressure rotor of low pressure with capped discs. The turbine has 14 discs in total.

Figure 1. The scheme of the NPP operation with the use of a hydrogen-fueled energy complex and an additional steam turbine: 1 – the system of water electrolysis; 2 – the system of hydrogen and oxygen compression; 3 – the system of hydrogen and oxygen storage; 4 – cooling heat exchanger; 5 - intermediate tanks containing hydrogen and oxygen; 6, 7 – the combustion chamber where hydrogen burns with oxygen for overheat steam before the additional steam turbine and before the intermediate steam superheaters respectively; 8 - the high and low pressure cylinders of additional steam turbine respectively; 9 – the recirculation of additional working fluid; 10 - the tank-accumulator; 11 – the separator; 12 – the steam overheater

The constantly operating turbine.

In the nominal operating mode, by using the combustion chamber 7 by means of a two-stage oxidation of hydrogen with oxygen, followed by mixing of the high-temperature steam produced with the main one [7], the displaced heating vapor intended for the re-heating enters an additional steam
turbine. Thus, combustion chamber 6 is not involved in operation. In this case, it is possible that the displacement of fresh steam occurs partially or completely, depending on the capacity according to which additional steam turbine is designed.

In the peak mode, the fresh steam is superheated to the temperature of 440 °C in front of the high-pressure cylinder of an additional steam turbine. Combustion chambers 6 and 7 are involved in the operation. At the end of the peak mode, the steam-hydrogen overheating of fresh steam ceases and the operation of the additional steam turbine returns to the nominal parameters.

In the nighttime period, the operation of an additional steam turbine in the mode of a hot rotating reserve is considered. The following modes can be considered.

1) No-load operation mode with steam throttling. The generator is disconnected from the power system. The rotor speed is maintained at ~ 1020 rpm [20] due to the minimum steam supply to the turbine head in the amount of 10% of the nominal flow [21,22]. The combustion chamber 7 is engaged, while the combustion chamber 6 does not operate. At the same time, the consumption of hydrogen and oxygen in the combustion chamber 7 is insignificant.

2) Motor mode - the generator is connected to the power system, without supplying steam to the head of the turbine while maintaining the rotor speed at the rated speed (3000 rpm). Combustion chambers 6 and 7 do not operate.

Thus, the operation of an additional steam turbine in conditions of daily use of steam-hydrogen overheating of the working fluid is associated with mode switching. This causes cyclic loading of the rotor, which can influence its working life in terms of the highest cyclic stresses in the root section of the blades, in the disc hubs or on the bore of the rotor, which will determine the highest stress range intensity index.

2. The main aspects of fatigue failure of metallic materials as a result of cyclic loading

The entire process of fatigue failure is conventionally divided into four stages: the initiation and initial development of cracks, stable growth, accelerated crack propagation and the final destruction of a sample or article [23-26].

The process of development of a fatigue crack, as a rule, is the only way of a part or a structure failure under cyclic loading in operation [23].

The frequency of loading affects the dynamics of the development of cracks. The frequency of cycles is the ratio of the number of stress cycles (strains) to the time interval of their action [24]. Moreover, an increase in the loading frequency can intensify the closure of cracks caused by appropriate oxide formation in the metal structure at low frequencies, either due to plastic deformation or due to the roughness of the fracture surfaces near the threshold voltages of the stress intensity factor, as well as the viscosity of the working medium or volumetric changes as a result of phase transformations of the failure area material. Such dependency is characteristic for some structural steels, titanium and aluminum alloys (Figure. 2) [23].

![Figure 2](image.png)

**Figure 2.** The effect of the loading frequency on the rate of growth of the fatigue crack in the aluminum alloy 2024-T3 (a) and steel of the type 13M2S (b)
3. The basic methods for assessing the growth rate of fatigue cracks and the number of cycles before failure and the initial data for calculation

The dependence of the growth rate of the fatigue crack on the loading frequency for fixed values of the stress range intensity index (ΔK) is described by the equations, mm / cycle [23]:

\[ v = v_0 f^{-\tan \alpha} \]
\[ v = v_0 f^{\tan \alpha} \]

where \( v_0 \) is the threshold (initial) speed of fatigue crack growth, mm/cycle; \( f \) is the frequency of cycles tensions, hertz; \( \tan \alpha \) is tangent of angle inclination to axis abscissa showing the change in the speed of fatigue crack growth with the increase in load frequency.

The index of stress range intensity is the algebraic difference of maximum and minimum cycle stress MPa (MPa / mm) [26]:

\[ \Delta K = K_{max} - K_{min} \]

where \( K_{max} \) and \( K_{min} \) are at the corresponding maximum and minimum cycle stress (\( \sigma_{max} \) and \( \sigma_{min} \)).

The equation determining stress frequency:

\[ f = \frac{n_c}{\tau} \]

where \( n_c \) determines number of cycles stress; \( \tau \) – time interval of the action of stress cycles, sec.

The length of the fatigue crack with the appropriate number of cycles is determined by the expressions, mm:

\[ l_N = l_0 + \Delta l \leq 0.1 \ldots 1 \]
\[ l_N = l_0 + \Delta l \leq 0.001 \ldots 0.01 \]

where \( l_0 \) is the threshold (minimum) value of crack length, mm; \( \Delta l \) is the increment of crack length, mm; 0.1-1 is the critical fatigue crack length for low-strength steels (tensile strength up to 1500 MPa), mm; 0.001-0.01 is the critical value of the fatigue crack length for high-strength steels (tensile strength is more than 1500 MPa), mm.

In the above evaluations, the threshold value of crack length is assumed to be 0.002 mm [24].

The increase in crack length is estimated by the expression, mm:

\[ \Delta l = v_0 N \]

where \( v_0 \) is the calculated value of the fatigue crack growth rate in accordance with expression (1) or (2), mm / cycle; \( N \) is the number of cycles.

Substituting (1) or (2) into (8) and then the result obtained in (6) into (7) we obtain a complex criterion for estimating the limiting number of cycles before failure in the form:

\[ v_0 f^{-\tan \alpha} N + l_0 \leq 0.1 \ldots 1 \]
\[ v_0 f^{\tan \alpha} N + l_0 \leq 0.1 \ldots 1 \]
\[ v_0 f^{-\tan \alpha} N + l_0 \leq 0.001 \ldots 0.01 \]
\[ v_0 f^{\tan \alpha} N + l_0 \leq 0.001 \ldots 0.01 \]

Thus, the determination of the limiting number of cycles before failure is reduced to the fulfillment of conditions (9) - (12), depending on the type of steel.
4. The estimation of the rate of fatigue crack growth and the limit number of cycles before the fatigue of the most loaded rotor element of the additional steam turbine

The rotor of the additional steam turbine is under constant tensile (positive) stresses caused by the centrifugal force.

The additional steam turbine operates in no-load mode from 00h to 7h, which corresponds to the minimum arising stresses in the rotor elements. By 8h its power increases first to nominal, and then it can participate in covering the morning peak of electric load due to the use of steam-hydrogen superheating of the steam in front of the high pressure cylinder up to approximately 13h. At the end of work in the morning peak, the load of the additional steam-turbine unit returns to the nominal parameters. At night, the turbine gets back to the no-load mode. Depending on the condition of the annual number of working days that roughly equals 330 combining the hydrogen-fueled complex to the NPP unit, taking into account the planned annual repairs, the number of load cycles will be 330. The frequency of loading is estimated to be $1.16 \cdot 10^{-5}$ Hz.

Moreover, we take into consideration the situation when an additional steam turbine switches to the no-load mode twice a day. This accounts for the doubling in number of load cycles, equal to 660. The frequency of loading in this case was $2.31 \cdot 10^{-5}$ Hz.

The tensile stresses in the root section of the working blades were determined taking into account the centrifugal force for the banding tape and the fastening wire.

The cross-sectional area of the blades for each stage of the rotor of an additional steam turbine was determined by the trapezium method.

The evaluation of the stresses occurring in the disks is based on the fact that the disk is divided into sections and, sequentially, for each section, radial and tangential stresses are determined in the direction from the rim to the hub (or vice versa) in two calculations: the first calculation is performed with a given number of revolutions of the disk (rotor) in minute; the second one is performed for a fixed disk (rotor). Then the true (total) stresses are determined. When going from one site to another, it is assumed that the voltage across the thickness of the disk is uniformly distributed [29]. In this article, the calculation was made in the direction from the rim to the hub of the disc.

Tangential stresses on the boring of the rotor were determined taking into account the calculated value of the load acting on the discs on the shaft [30].

Figures 3-5 show the results of calculation of tensile stresses in the root section of the blades, on the inner radius of the disk hub and on the rotor boring, respectively, of an additional steam turbine at a given number of rotor revolutions per minute. It should be noted that in the nominal and peak conditions, the tensile stresses in the root section of the blades are unchanged due to the unchanged rotation speed of the turbine rotor, which causes the centrifugal force to remain unchanged. The same applies to radial and tangential stresses on the inner radius of the disc hub, which also remain unchanged. In this case, the negative value of radial stresses on the inner radius of the disk hub indicates that the interference is not lost and the discs continue to remain under compression, but decrease due to the presence of a centrifugal force in comparison with a stationary resting rotor. In this case, the true (total) stresses on the inner radius of the disk hub are in the range from -5 to -15 MPa, taking into account the conversion factor according to the recommendations [29].

Difference in tensions occurs while switching to no-load mode because this mode is connected with low number of rotations. Thus, difference appearing of tensions in section root of blades, in discs and in bore of rotor. Tensions increase on internal radius of discs hub as radials because centrifugal force is lowering. These tensions become more negatives and approach to the value corresponding to the condition of rotor in immobility (line 3 on figure 4). It is caused by the increase in tightness which decreases with the increase in rotations and vice versa. Radial stress tension must not be zero or positive in the internal radius of discs hub. Otherwise tightness will disappear. It is can provoke the rotation of disc on shaft, vibration and emergency [29].
Figure 3. Tensile stresses in the root section of the blades: 1 – nominal and peak modes; 2 – no-load mode

Figure 4. Radial and tangential stresses on the inner radius of the disk hubs: 1, 2 - radial and tangential stresses, respectively, in nominal and peak conditions; 3, 4 - radial and tangential stresses respectively in no-load mode

Figure 5. Tangential stresses on the rotor bore:
1 - nominal and peak modes; 2 - no-load mode

The permissible stresses on the example of stainless steel 15H12BMF (EI802), that is used to make both steam turbine blades and discs with a shaft for operating temperature up to 560 ° C, were ± 416.7
MPa, taking into account the yield strength of this alloy steel of 750 MPa with the coefficient Safety factor 1.8 [30]. Thus, the obtained values of the stresses do not exceed the permissible values.

In the motor mode, due to the absence of steam passage through the turbine, the radial and tangential stresses on the inner radius of the disc hub slightly increase compared to the nominal mode. This is due to the fact that the discs experience somewhat less radial stress at the rim radius due to the absence of influence of the vapor pressure transmitted by the spatula, i.e. the resultant force of steam pressure on the blades is absent. This in turn creates a slight unloading of the discs, which eventually leads to a corresponding slight increase in the interference on the inner radius of the disc hub. The tangential stresses on the boring of the rotor remain approximately the same as in the nominal mode. The tensile stresses in the root section of the blades do not change in comparison with the nominal mode, due to the invariance of the rotor speed.

On the basis of the obtained results on the evaluation of stresses that occur in the main elements of the rotor of an additional steam turbine, it is obvious that the greatest difference in the arising tensile stresses is observed in the root section of the blades of the last two stages during the transition from the nominal mode to no-load mode and back. Thus, in the nominal mode, the tensile stresses in the root section of the blades of the last two stages are approximately 407 MPa, and in idle mode approximately 47 MPa. This causes the greatest value of the intensity factor of the stress range, equal to the difference between the stress intensity factors determined at the corresponding maximum and minimum stresses according to the expression (3). At the same time, the model of a flat body of finite dimensions with an initial crack length of 0.002 mm, to which tensile stresses are applied, was used as the basis for the tentative definition of the dimensionless coefficient $\alpha$, taking into account the geometric factor and the nature of the voltage distribution with respect to the blades [28]. Thus, the intensity factor of the stress range was $\Delta K = 0.9$ MPa.

For the frequency of loading of the rotor of an additional steam turbine $\nu = 1.16 \cdot 10^{-5}$ Hz, the calculated value of the fatigue crack growth rate was $\nu = 7.51 \cdot 10^{-7}$ mm / cycle, starting from the initial growth rate of the order of $\nu_0 = 10^{-8}$ mm / cycle based on the fatigue fracture diagram, the number of cycles before failure was $N \approx 168$ thousand cycles.

For an increased frequency of rotor loading of an additional steam turbine $\nu = 2.31 \cdot 10^{-5}$ Hz, the calculated value of the fatigue crack growth rate was $\nu = 5.78 \cdot 10^{-7}$ mm / cycle based on the initial growth rate of the order of $\nu_0 = 10^{-8}$ mm / cycle, the number of cycles before failure was $N \approx 168$ thousand cycles.

The increased number of cycles before failure is due to the fact that on the basis of the fatigue fracture diagram the magnitude of the stress intensity intensity factor equal to 0.9 MPa is in the first region called threshold, and only for this region the increase in the frequency of loading is associated with the closure of the fatigue cracks [23], which ultimately affects the slowing of the rate of growth of the fatigue crack, which is taken into account in expression (1).

Thus, from the condition of the initial crack length equal to 0.002 mm and the initial rate of growth of the fatigue crack equal to about 10-8 mm / cycle, the blades of the last two stages of the additional steam turbine, loaded with cyclic stresses, can withstand more than 100,000 load cycles.

5. Conclusions
1. The use of steam-hydrogen overheating at an additional steam turbine is associated with an alternation of operating modes when entering a peak load and during a no-load period at night. This, in turn, causes the occurrence of cyclic stresses in the most important elements of the rotor - in the root section of the blades, in the discs and in the boring of the turbine rotor.

2. The greatest values of the intensity factor of the stress range in the most important elements of the rotor are manifested when the additional turbine is unloaded to idling. At the same time, the largest value of the intensity factor of the stress range when the turbine enters the nominal and peak modes is manifested in the root section of the blades of the last stages of the turbine rotor.

3. The lowest values of the stress range intensity index in the most important elements of the rotor are when the additional turbine is unloaded and transferred to the motor mode. At the same time, due to
to the invariability of the rotor speed, the cyclic stresses do not arise when the turbine enters the nominal and peak modes in the root section of the blades, but in discs and in the boring of the rotor it does not significantly differ from similar voltages at nominal or peak operating conditions.

4. In the peak load mode with a slightly increased steam flow through the turbine due to steam-hydrogen overheating of the working fluid in comparison with the nominal flow, there are no significant changes in the radial and tangential stresses in the discs and in the boring of the rotor.

5. The values of the stress range intensity index for the most important elements of the rotor of the additional steam turbine in all the considered modes are in the area of the threshold values of the fatigue failure diagram. For this area, an increase in the frequency of loading is associated with the phenomenon of closure of the fatigue crack and, as a consequence, a possible slowing of its growth.

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