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Risk-Based Operation and Maintenance Planning of Steam Turbine with the Long In-Service Time

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Abstract: In order to ensure the safety of power generation in Poland and to maintain energy production from coal-fired units with the long in-service time, it is required to develop a strategy for the further operation of the conventional power plants in conditions of increased flexibility. The presented research focuses on the critical component of the steam turbine, which is the high-pressure rotor. The methodology of the forecasting of crack propagation and growth of life-consumption processes was described, and the probability of a failure in subsequent years was estimated. The development of the identified phenomena depends mainly on the stress increases during start-ups; therefore, these increases were determined to ensure the safety of the turbine's operation during the assumed period of operation (13 years). The permissible stress for rotor central bore (threatened with crack propagation) was 220 MPa for start-ups which were not carried out "on demand", and for heat grooves (threatened with life-consumption processes) it was 420 MPa or 210 MPa, depending on the initial wear level of the material. An algorithm for online stress monitoring was presented, taking into account the variability of the heat transfer coefficients. The compiled method can be transformed into a real-time stress level control system. As a result, it is possible to obtain the desired increase in stress during start-up. For a longer service life (20 years), a method of selecting the optimal time interval to carry out preventive actions based on a risk analysis was additionally delineated. The optimal year to perform repair was between the 14th and 15th year of operation. The developed research allows presenting a strategy for further operation and maintenance (O&M) of the turbine, which can be adapted to a real unit.

Keywords: risk; O&M strategy; steam turbine; stress monitoring

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1. Introduction

Maintaining energy production from long-operated 200 MW coal-fired units in Poland is a key task to ensure adequate safety of the power system operation by guaranteeing energy supplies to consumers, as well as balancing the current demand, especially in the presence of unstable renewable sources [1–4]. Due to the significant life consumption of individual elements [5], it is necessary to develop a strategy for further operation under flexible load regime, with the lowest possible financial outlays for preventive maintenance of the unit.

Numerous studies on the selection of an appropriate method of operation and maintenance of power units for various energy generation technologies were carried out. An example may be the paper by Zhao et al. [6] concerning the carburizing system of a heat and power plant. The main failures of mechanical devices and methods of their reduction during further operation of the installation are presented, which significantly improves production safety while increasing efficiency. The influence of flexible operation of coal-fired units on the resilience of critical elements is described by Stoppato et al. in [4]. The procedure to predict the remaining life time was developed. This enables rational planning of further operation. The procedure is presented in detail for a steam superheater.

Gabbar et al. [7] focus on a system which is designed to capture the experience of employees in the operation and maintenance of nuclear power plants. The system processes data from text and voice documents and creates an integrated scheme of actions for different situations, which ensures the implementation of appropriate procedures even by less experienced employees. Another study dedicated to nuclear power plants was described by Ikegami et al. [8] and concerns the planning of further operation of older units, for which the effects of degradation processes are observed. Various aging management methods are reviewed, taking into account the need to reduce operating costs. In Hatti's work [9], the problems related to the operation and maintenance of solar power plants are highlighted, especially for those located in areas where it is impossible to perform remote control requiring a permanent Internet connection. Costa et al. analyze the new trends in the operation and maintenance of wind turbines [10]. An extensive review of the literature has been carried out in terms of the analysis of number and types of failures, reliability, maintenance strategies, monitoring systems and service optimization. Increased interest in the subject of turbine failure was found.

In the case of research related to the management methods for power plants, risk analysis is frequently used. An example may be the article [11] by Liu et al., which focuses on the wear degree assessment for the steam superheater in the supercritical coal-fired plant. The determined risk of failure is used to define the safety of further operation. In Hardy's et al. paper [12], the corrosion-related risk is described. A diagnostic system for the evaporator walls of a coal-fired boiler was developed to predict the risk of a failure by measuring the exhaust gas composition, which was directly correlated with the speed of the corrosion process. Another example of risk analysis is described in [13] by Feili et al. This time, the geothermal source is taken into consideration. The Failure Modes and Effects Analysis (FMEA) was carried out and was used to determine the risk of an identified event. As the result of research, it is possible to develop recommendations for corrective and preventive maintenance for various types of failures.

The presented article is a continuation of research related to the critical components of 200 MW turbines, such as the high-pressure rotor. The previously published papers [5,14] focused on presenting the methodology for forecasting the development of degradation processes such as crack propagation or low-cycle fatigue. The optimization of time interval of preventive repairs based on the risk analysis was also described. This proven method can be used to prepare a complete strategy of turbine further operation and maintenance (O&M) using the stress-control system which allows for moderating stress increases during start-up and sustaining them at the desired level for both identified risk areas. Section 2 presents the object of research and two main failure scenarios: crack propagation in the rotor's central bore and the life-consumption processes in the heat grooves. The remaining operation time and its possible course for the analyzed power unit were also defined. Section 3 describes the stress-control algorithm, which is especially useful during the start-ups, dedicated for both defined critical areas. Section 4 presents the proposed risk-based strategy for turbine operation and maintenance, which allow for avoiding serious failure. Section 5 is a brief summary of the research.

2. The Research Object and Its Operation

2.1. Critical Areas of a Steam Turbine Rotor

The presented research concerns the operation and maintenance of the long in-service 200 MW steam turbine, mainly the high-pressure rotor. In this rotor, two risk areas were identified [5], which are marked in Figure 1. In area I, located in the central bore of the impulse stage, there is a risk of crack propagation, which may cause one of the most serious turbine failures. The crack may appear as a residue of technological processes and propagate due to the increased level of circumferential stresses. In area II, the material is damaged due to the intensification of low-cycle fatigue and creep in the thermal grooves, due to the shape and dimensions of the grooves, which favor the accumulation of effective stresses during transient states.

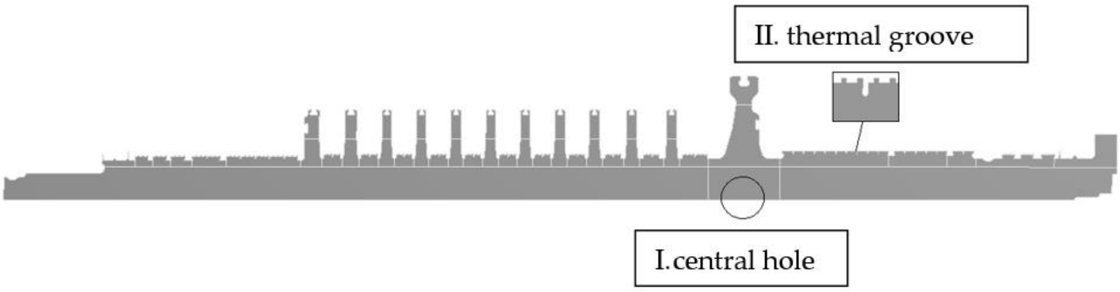

Figure 1. Critical areas of a steam turbine rotor.

2.2. Crack Propagation

In order to estimate the increase in the crack dimension in the rotor central bore, the stress intensity coefficient is used [15–17]:

$$K_I = M\sigma\sqrt{a} \tag{1}$$

where:

M—nondimensional function of geometrical parameters of the crack and the element;
σ—tensile stress level in the crack area;
a—crack dimension.

The crack propagation is described by Formulas (2) and (3) for the operation under variable load conditions and in the quasi-steady-state [16]:

$$\frac{da}{dN} = C(\Delta K)^m \tag{2}$$

$$\frac{da}{dt} = AK^n \tag{3}$$

where:

$C,\ m,\ A,\ n$—material constants;
$\Delta K = K_{Imax} - K_{Imin}$—amplitude of the stress intensity factor;
N—number of fatigue cycles;
t—operation time in steady state.

The rotor failure is caused by exceeding crack critical dimension a_{cr}, which can be determined using the formula [15,18]:

$$a_{cr} = \left(\frac{K_{IC}}{M\sigma}\right)^2 \tag{4}$$

where:

K_{IC}—material toughness.

The probability of such a situation is described by the relation [5,14,15]:

$$P_f = P(a_{cr} - a \leq 0) \tag{5}$$

When both the actual crack dimension a and the critical dimension a_{cr} are described by normal distributions, the probability can be calculated as [15]:

$$P_f = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{z=z_1} exp\left(-\frac{1}{2}z^2\right) dz \tag{6}$$

$$z_1 = -\frac{\mu_{a_{cr}} - \mu_a}{\sqrt{\delta_{a_{cr}}^2 + \delta_a^2}} \tag{7}$$

where:

$\mu_{a_{cr}}$—critical crack dimension mean value;
μ_a—actual crack dimension mean value;
$\delta_{a_{cr}}$—critical crack dimension standard deviation;
δ_a—actual crack dimension standard deviation.

2.3. Low Cycle Fatigue and Creep Processes

The fatigue-creep consumption level for material in thermal grooves can be estimated using the hypothesis of linear damage accumulation [15,19]:

$$Z = N \sum_{i=1}^{m} \frac{1}{a(\Delta\varepsilon_i)^b} + t\frac{1}{c\sigma^d} \tag{8}$$

where:

a, b, c, d—material constants;
$\Delta\varepsilon$—strain changes;
m—number of significant strain changes during one cycle;
N—number of fatigue cycles;
t—number of hours of operation in steady-state;
σ—stress during steady-state.

The failure occurs when the wear limit value $Z_g = 1$ for life-consumption processes is exceeded. The probability of such an event is then defined as [5,15]:

$$P_f = P(Z_g - Z \leq 0) \tag{9}$$

When both the variables Z and Z_g are described by normal distributions, failure probability can be calculated from the Equation (6), where:

$$z_1 = -\frac{\mu_{Z_g} - \mu_Z}{\sqrt{\delta_{Z_g}^2 + \delta_Z^2}} \tag{10}$$

where:

μ_{Z_g}—wear limit mean value;
μ_Z—actual wear mean value;
δ_{Z_g}—wear limit standard deviation;
δ_Z—actual wear standard deviation.

2.4. Steam Turbine Operation Conditions

Both the crack propagation and life-consumption processes strongly depend on the stress level during variable load and the quasi-steady state. It is assumed that under creep conditions (after the relaxation process), the stress in area I is approximately 55 MPa [5]. In area II in turn, it is so low that the total creep life is assumed at the level of 10^6 h. The rotor start-up is considered as the variable load state of the greatest importance for the damage processes. When an analyzed plant is intended to be operated as the power-regulatory unit, some start-ups are carried out "on demand", which means that the heating process may be much faster than in the manufacturer's recommendations. In such a situation, the level of circumferential stresses (σ_Z) affecting the crack propagation in area I may increase to 300 MPa [5,14]. The effective stresses in the thermal grooves (σ_{red}) then reach the highest level in the entire rotor and may increase to 400 MPa [5], and even more due to the significant contribution of thermal stresses in this area.

The oldest coal units will be slowly withdrawn from the Polish energy system; however, some of them are still planned to be operated in the long term [20]. Therefore, it was assumed that the considered turbine will be operated for at least the next 13 years (until 2035) with the possibility of extending its operation time to 20 years. Each year, the number of start-ups will be equal to 200, and 25% of them will be carried out in the "on-demand" mode, resulting in maximum stress increases up to 300 MPa in the central bore and 450 MPa in the most loaded heat groove. The operation time in steady-state will be 30 h.

3. Online Stress Control in the Turbine Rotor

3.1. Online Stress Monitoring Algorithm

The stresses occurring in the critical areas of the turbine rotor are related to the steam pressure, rotational speed of the rotor's own mass and the heating process. The dependencies allowing the determination of the level of stresses coming from the surface load (related to pressure) and the mass load (related to the rotational speed) are described as [15,21]:

$$\sigma^p = b \cdot p \tag{11}$$

$$\sigma^n = c \cdot n^2 \tag{12}$$

where:

b, c—coefficients determined for each stress component and for each analyzed area;
p—steam pressure;
n—rotational speed.

The coefficients b and c can be determined using the finite element method (FEM). The steam pressure and rotational speed are quantities measured with a certain time step.

In the case of thermal load, for which the greatest increase occurs during start-ups, the influence functions can be used. They are illustrating the material response to the unit temperature increase in the steam washing the rotor. Such functions are determined by means of FEM both for the metal temperature in a given area and for the material's stress response (so-called Green functions) [15,21]. The application of the superposition method allows the summation of unit responses of the material at a given time instant and is carried out using the Duhamel integral. In order to take into account the variability of the heat transfer coefficient, instead of the steam temperature increase, the modified temperature is used. It is calculated as [22]:

$$T_{mod} = \frac{\alpha(\tau) \cdot (T(\tau) - T_m(\tau))}{\alpha_{mod}} + T_m(\tau) \tag{13}$$

where:

$\alpha(\tau)$—heat transfer coefficient at a given time instant;
α_{mod}—heat transfer coefficient for which the Green function was determined;
$T_m(\tau)$—metal temperature at a given time instant.

The Duhamel integral is also used to calculate the temperature of the metal according to the relation:

$$T_m = \int_0^t \frac{dT(\tau)}{d\tau} f_{T_n}(t - \tau) \tag{14}$$

where:

$f_{Tn}(\tau)$—influence function for metal temperature.

Additionally, when the range of variability of the heat transfer coefficient is wide (as in the case of the analyzed rotor), it is possible to divide the start-up into individual stages. For each of them, dedicated influence functions are determined and a constant or in-time-varying correction coefficient k is applied, in place of the α_{mod} coefficient. Then, greater compliance with the results calculated from FEM is obtained [23].

The thermal stress for individual stress components is determined by the formula:

$$\sigma^T = \int_0^t \frac{dT_{mod}(\tau)}{d\tau} f_n(t-\tau)d\tau \tag{15}$$

where:

$f_n(\tau)$—Green's function determined for a specific heating stage and a specific stress component.

Taking into account the surface, mass and thermal stress components, the following equation for total stress is obtained:

$$\sigma = \sigma^T + \sigma^p + \sigma^n \tag{16}$$

It can be used in the case of area I of the turbine rotor for the circumferential component leading to crack propagation in central bore. For area II, where there is a risk of high level of effective stress, the Huber−Mises hypothesis is used [23]:

$$\sigma_{red} = \sqrt{\sigma_X^2 + \sigma_Y^2 + \sigma_Z^2 - \sigma_X\sigma_Y - \sigma_Y\sigma_Z - \sigma_X\sigma_Z + 3\tau_{XY}^2} \tag{17}$$

where:

the indexes X, Y, Z refer to the individual normal and shear stress components.

The correct operation of the presented online stress monitoring algorithm was verified by comparing the results with those obtained using the FEM for a number of different start-ups. An example comparison for a cold start-up, carried out in accordance with the increase in parameters shown in Figure 2, is presented in Figure 3 for area I and II of the analyzed rotor. A satisfactory agreement was obtained and the maximum stress values for the monitored period are compatible for both areas.

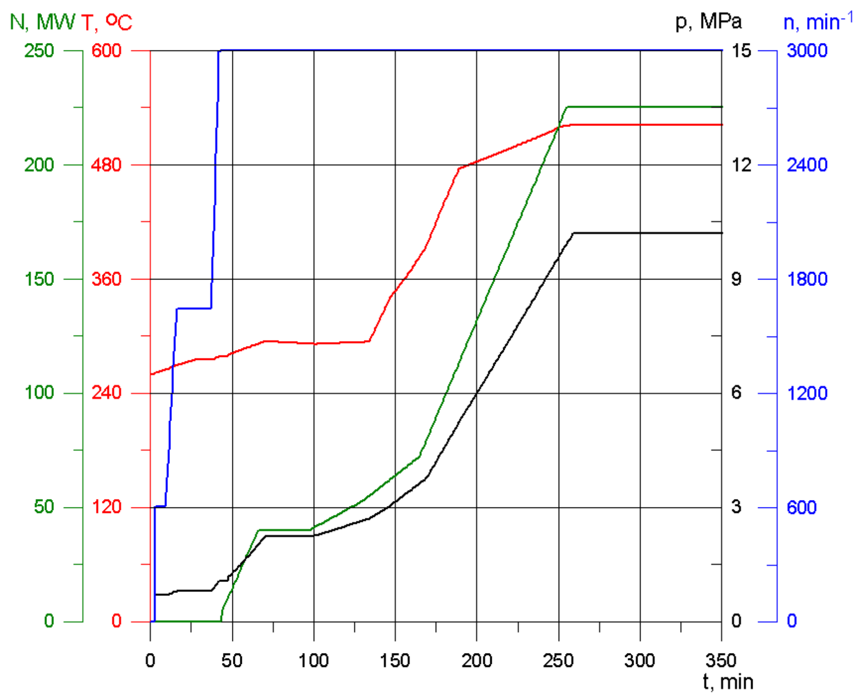

Figure 2. Change in parameters during original rotor cold start-up: N—turbine power, T—steam temperature, p—steam pressure, n—rotational speed.

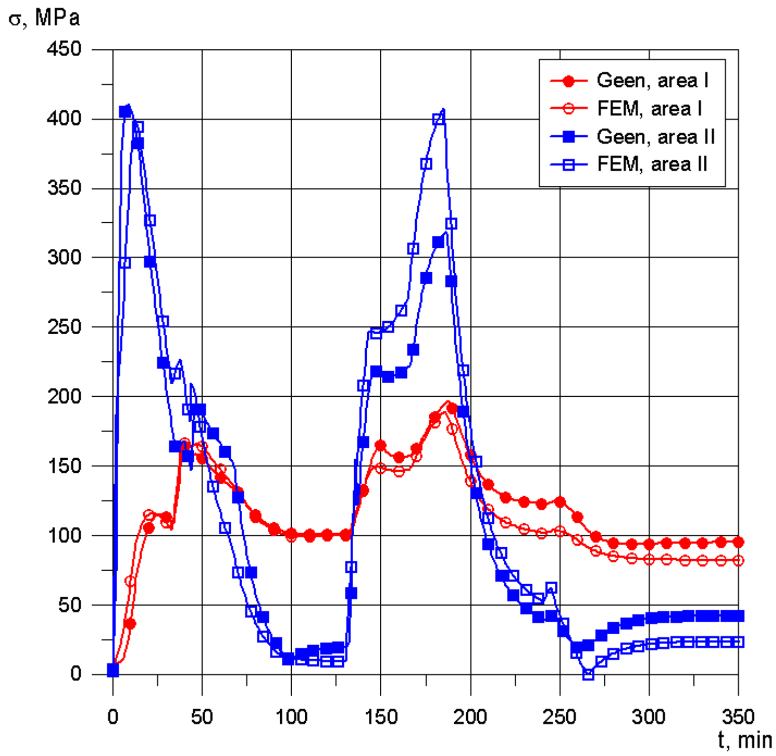

Figure 3. The course of stresses in rotor critical areas calculated using online stress monitoring algorithm (based on Green's functions) and FEM.

3.2. Stress Level Control in the Critical Rotor Areas

The online monitoring system described in Section 3.1 can be used to supervise the stresses during the rotor operation, but it has limited possibilities of their adjustment due to the use of measurements of rotor rotational speed and steam parameters in the calculation process. In order to be able to control the stress level in defined critical areas, a modification of the algorithm is used, which consists in using only the measurements of the rotational speed n and steam pressure p. The suggested jump of the steam temperature $dT(\tau_i)$ (after discretization ΔT_i) is selected for the time step τ_i in such a way as to approach the imposed permissible stress σ_{per} by solving Equation [15]:

$$\sigma_{per} = \int_0^{t_i-\Delta t_i} \frac{dT(\tau)}{d\tau} f(t-\tau)d\tau + \int_{t_i-\Delta t_i}^{t_i} \frac{dT(\tau_i)}{d\tau} f(t-\tau)d\tau + \sigma^p + \sigma^n \tag{18}$$

In the case of area I of the rotor, Equation (18) is used directly because of the single stress component. In the case of area II, it is necessary to use effective stresses according to Formula (17). Equation (18) then takes an implicit form and can be solved by the iterative method. Additionally, in the stress monitoring algorithm, the modified steam temperature T_{mod} is used; therefore, the increase in this temperature will be determined from Equation (18), and it must be converted into the temperature of the steam washing the rotor, which will ensure that the imposed level of permissible stresses is maintained. For this purpose, an equation is used:

$$T_i = \frac{\alpha_i \left[\left(\int_0^{t_i-\Delta t_i} \frac{dT(\tau)}{d\tau} \cdot f_{T_n} \right) - T_{i-1} \left(\int_{t_i-\Delta t_i}^{t_i} f_{T_n} \right) \right] + k \left[T_{mod_i} - \left(\int_0^{t_i-\Delta t_i} \frac{dT(\tau)}{d\tau} \cdot f_{T_n} \right) + T_{i-1} \left(\int_{t_i-\Delta t_i}^{t_i} f_{T_n} \right) \right]}{\alpha_i - \int_{t_i-\Delta t_i}^{t_i} f_{T_n} (\alpha_i - k)} \tag{19}$$

where:

index i refers to the current time step;
index $(i-1)$ refers to the previous time step.

In order to limit the operation of the presented stress control algorithm, basic restriction is introduced in the scope of the allowable, determined temperature changes ΔT_i. The maximum permissible increase in the further calculations is at the level of 4 °C, but it can be individually selected for the real power unit. The moment of increasing the rotor rotational speed is treated as the beginning of the start-up. Then there is a rapid increase in stresses resulting from the initial temperature difference between the steam and the metal. In the next steps, the stress control system is activated so that the assumed value of σ_{per} is not exceeded. It is possible to control a single area, especially when there is an identified advanced damage degree or crack of significant dimension. However, in the analyzed rotor, the control is carried out for both areas simultaneously. The suggested temperature increase is selected individually as ΔT_I and ΔT_{II}, then both values are compared and the smaller one is selected. As a result, the permissible stresses are not exceeded in any area. The described control process is schematically presented in Figure 4.

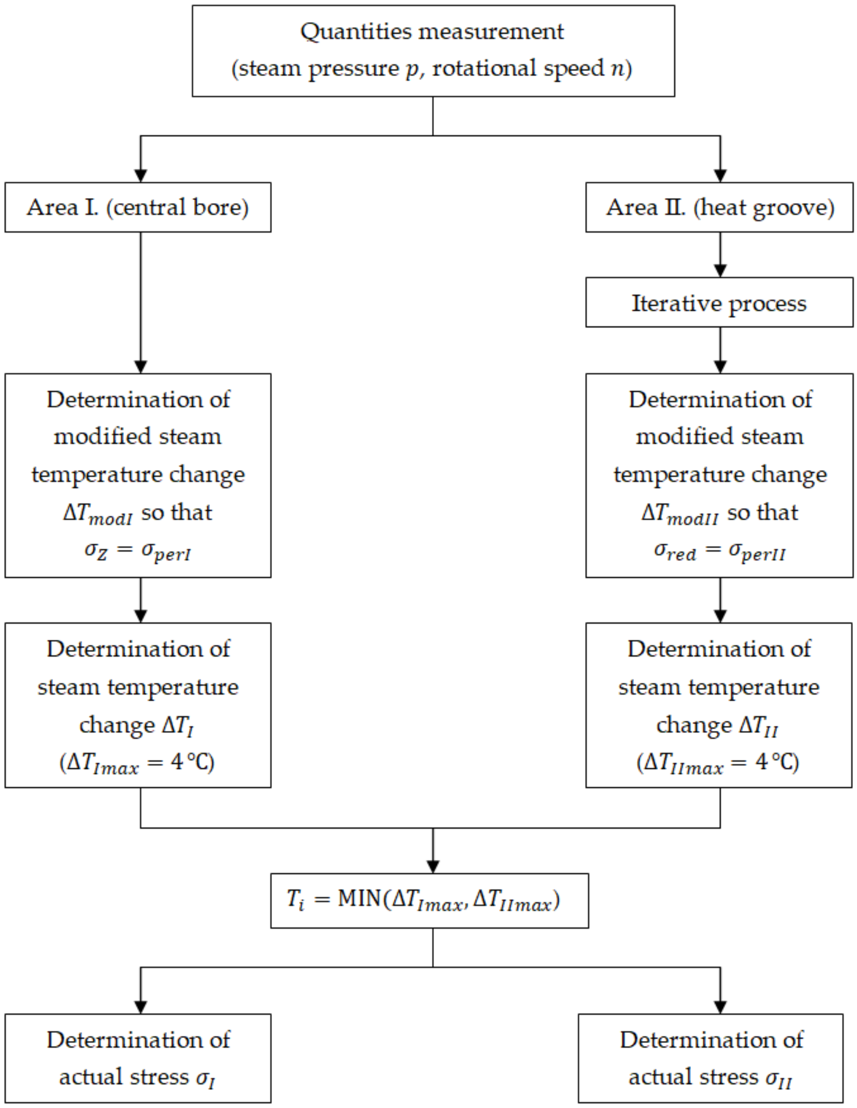

Figure 4. Stress control algorithm for the critical areas of the steam turbine rotor.

4. Operation and Maintenance Strategy for Steam Turbine

4.1. Remaining Operation Time t = 13 Years

The first developed strategy for further operation and maintenance of the analyzed steam turbine assumes the remaining in-service time of 13 years. Additionally, no preventive repairs, restoring the original condition of the material, are planned for the entire period. According to the description of the operating conditions in Section 2.4, it was assumed that the number of start-ups per year will be 200, and 25% of them will be carried out in the "on demand" mode, in order to meet the requirements of the regulatory work regime. During such a quick start-up, the stresses in the central bore for the circumferential component each time increase to 300 MPa, and the effective stresses in the thermal grooves to 450 MPa. For the analyzed period, the levels of allowable stresses during the remaining start-ups were selected in such a way that the risk of failure during 13 years is at a low level by preventing the failure probability from increasing above 0.001. For this purpose, a series of calculations related to crack propagation and life-consumption processes development were made. To take into account the random nature of these phenomena, the Monte Carlo method was used with the equations described in Section 2. Eight hundred simulations were performed for each calculation step. For the sampling method, a random number generator was used. Table 1 shows the mean values and standard deviations of individual variables characterized by normal distributions, which were chosen because of their popularity in the nature and based on central limit theorem. To change uniform distribution of variables into a normal one, the Box−Muller transform was used. The initial dimension of the crack was assumed as $a_0 = 3$ mm. For fatigue-creep life consumption, two variants of the initial wear of the material were considered: $Z_0 = 0$ and $Z_0 = 0.5$. Similar analyses were presented in [5,14] for other crack initial dimensions and for three operation scenarios in order to present the potential development of destruction processes for turbine rotors. In this paper, the estimation of the crack dimension and the damage degree over time is used to determine safe values of the stresses during start-ups.

Table 1. Mean values and standard deviations of quantities affecting crack propagation and life-consumption process.

Input Data	Mean Value	Standard Deviation
Crack Propagation		
C	2×10^{-12}	1×10^{-13}
m	3.4537	0.173
A	3×10^{-14}	1.5×10^{-15}
n	5.6572	0.283
$\Delta\sigma_{max}$	300 MPa	15 MPa
σ_{creep}	55 MPa	2.75 MPa
a_0	3 mm	0.5 mm
K_{IC}	60 MPa$\sqrt{\text{m}}$	3 MPa$\sqrt{\text{m}}$
Life consumption processes		
a	464	46.4
b	−1.589	−0.079
E	180 GPa	9 GPa
$\Delta\sigma_{max}$	450 MPa	22.5 MPa
Z_t^{-1}	1,000,000 h	50,000 h
Z_0	0/0.5	−/0.05
Z_g	1	0.03

Figure 5 shows the crack propagation and the probability of failure due to exceeding the critical dimension for the assumed service life. The calculations were made for various values of circumferential stresses, generated as a result of start-ups, which are not carried out in the "on demand" mode. The calculations show that the allowable stress value in the central bore should be $\sigma_{perI} = 220$ MPa. Then the probability of 0.001 will not be exceeded in the entire analyzed period.

Figure 6 represents the increase in wear degree resulting from the low-cycle fatigue and creep in the rotor's thermal grooves, as well as the probability of reaching the limit value $Z_g = 1$, which corresponds to the loss of material lifetime in this area. Initial wear was assumed as $Z_0 = 0$. The calculations were made again for a number of different stress values during start-ups. The probability of 0.001 will not be exceeded during 13 years of operation, when the permissible value σ_{perII} is not higher than 420 MPa.

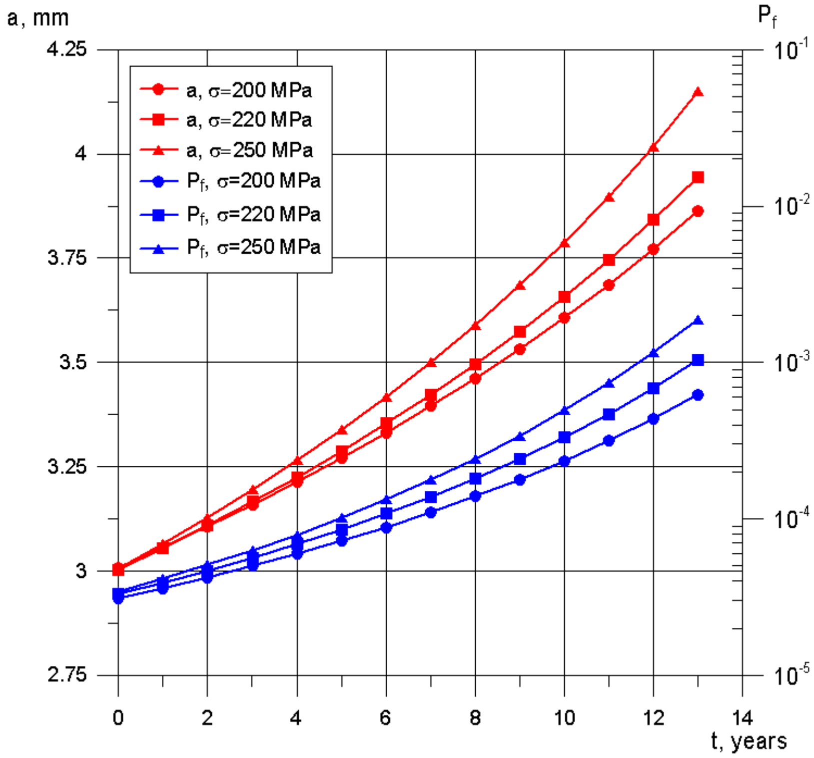

Figure 5. Crack propagation and turbine rotor failure probability over 13 years of operation.

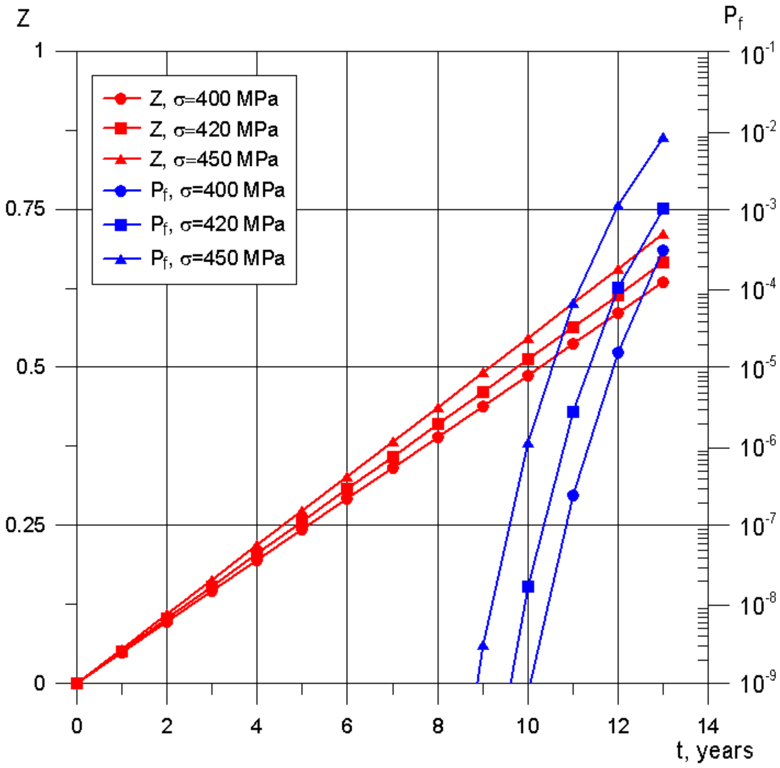

Figure 6. Material fatigue-creep wear and probability of turbine rotor failure over 13 years of operation, $Z_0 = 0$.

For the values of permissible stresses in both critical areas of the rotor and the material cold initial state, the algorithm presented in Figure 4 was applied. In this way, optimal temperature increases of the steam washing the rotor during its cold start-up which is not carried out in the "on demand" mode were selected, ensuring the maintenance of the appropriate level of stresses. The calculated course of stresses and the course of the steam temperature are shown in Figure 7. The initial stress increase is caused by incipient temperature difference between the metal and the steam. Then the stress control system is used to achieve the allowable stress in both critical areas. The exact σ_{per} value is not reached for entire heating process due to the introduced limitations on the permissible temperature jump, hence peaks and valleys appear. The great advantage of using the proposed algorithm is the significant reduction in start-up time from approx. 250 min (Figure 2) to approx. 130 min. The final drop in stresses is related to the slow stabilization of the temperature due to the cessation of the heating process. The maximum value of circumferential stress for area I is 227 MPa, and the effective stress for area II is 424 MPa.

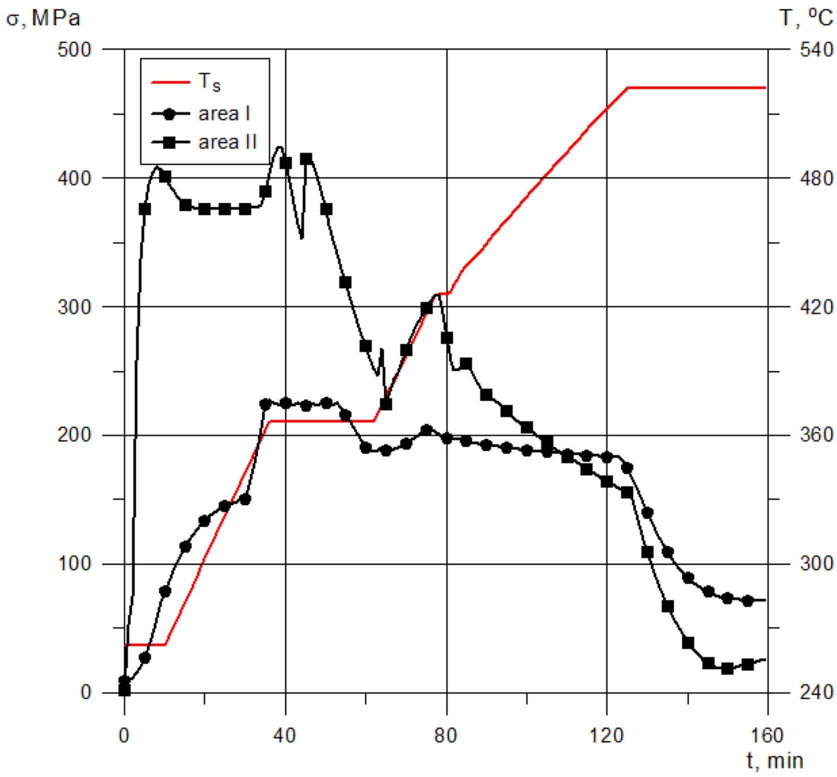

Figure 7. The course of stresses in critical areas and steam temperature after applying the stress control algorithm for $\sigma_{perI} = 220$ MPa and $\sigma_{perII} = 420$ MPa.

Analogous calculations were made for a higher level of initial wear in the thermal grooves $Z_0 = 0.5$. In such a case, it is not possible to select the stress that would ensure that the probability of 0.001 is not exceeded in the whole service life, assuming that 25% of starts are carried out in the "on demand" mode. The turbine then should not be dedicated to a regulatory work regime, and it is possible to select the permissible stress values which should be maintained for each start-up. Figure 8 shows the material fatigue-creep life consumption and the related failure probabilities for this variant. The allowable stress value is $\sigma_{perII} = 210$ MPa.

Again, the stress control algorithm was used for the analyzed variant, in which the start-ups are not carried out in the "on-demand" mode, and the permissible stress in area II is 210 MPa (Figure 9). The stress is so low that in the case of a cold start-up, it will be necessary to preheat the rotor so that the temperature difference between the steam and the material does not exceed 60 °C. The greater temperature difference will result in exceeding the allowable stresses from the very beginning. Additionally, it determines the speed of heating, i.e., the permissible stress in the central bore will not be significant during the operation of the algorithm and its maximum value is approximately 190 MPa. The disparities in the course of stresses are visible again due to the limitations related to temperature increase for each time step. The designated heating method is not as monotonous as in the case of Figure 7, because of the required lower stress values. The heating procedure is therefore slightly slower and the start-up (without turbine preheating time) has been shortened to approx. 140 min.

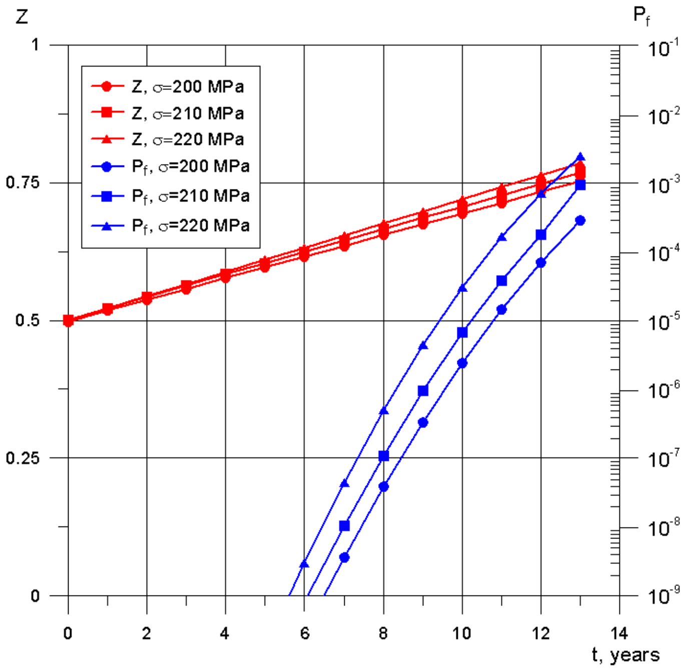

Figure 8. Material fatigue-creep wear and probability of turbine rotor failure over 13 years of operation, $Z_0 = 0.5$.

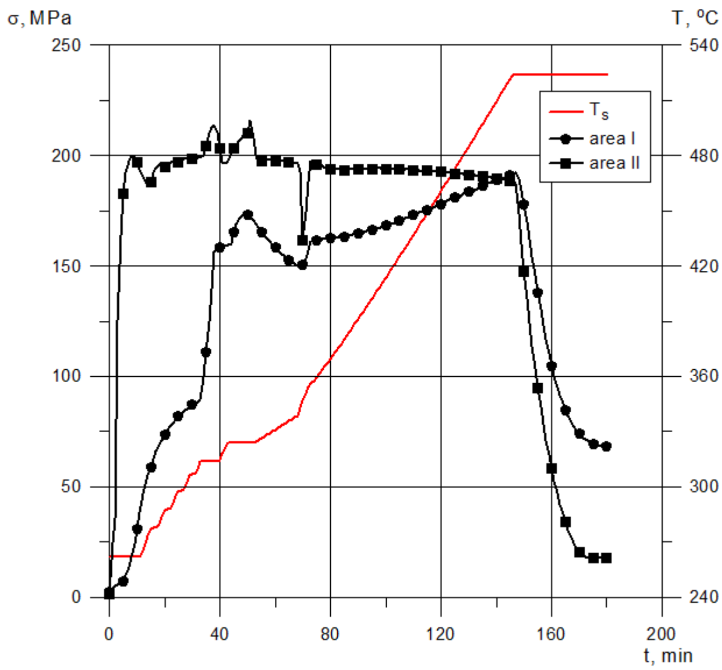

Figure 9. The course of stresses in critical areas and steam temperature after applying the stress control algorithm for $\sigma_{perII} = 210$ MPa.

4.2. Remaining Operation Time t = 20 Years

The second developed strategy assumes the possibility of extending the operation time after 13 years by another 7 years. In such a case, in order to ensure an appropriate level of safety, it will be necessary to carry out preventive activities, such as diagnostic tests and corrective repair in critical areas. Without proper preventive maintenance, the probability of failure within 20 years of operation may even reach values close to 1, which is also confirmed by research conducted by Krishnasamy et al. [24] for steam turbines.

The new strategy is based on the earlier calculations presented in Chapter 4.1. This means that in the first step, the stress values were selected in such a way that the probability of an undesirable event did not exceed the value of 0.001 within 13 years. To implement the assumptions, the stress control algorithm is used (Figure 4). Table 2 presents a summary of the previously determined values.

Table 2. Stress values during steam turbine rotor start-up.

Critical Area	"On Demand" Mode Start-Ups	Remaining Start-Ups
	$a_0 = 3$ mm, $Z_0 = 0$	
Area I (central bore)	$\sigma = 300$ MPa	$\sigma = 220$ MPa
Area II (heat groove)	$\sigma = 450$ MPa	$\sigma = 420$ MPa
	$a_0 = 3$ mm, $Z_0 = 0.5$	
Area I (central bore)	–	$\sigma = 190$ MPa
Area II (heat groove)	–	$\sigma = 210$ MPa

In order to determine the optimal time of carrying out preventive actions, the NPV index [5,14] was used, described by the formula:

$$NPV = \sum_{j=1}^{l}\sum_{t=0}^{N}\frac{P_{f0j}C_{tj}}{(1+r)^t} - \sum_{j=1}^{l}\sum_{t=n}^{N}\frac{C_{rtj}}{(1+r)^t} - \sum_{j=1}^{l}\sum_{t=0}^{n}\frac{P_{f0j}C_{tj}}{(1+r)^t} - \sum_{j=1}^{l}\sum_{t=n}^{N}\frac{P_{f1j}C_{tj}}{(1+r)^t} \quad (20)$$

where:

P_{f0}—failure probability in the period prior to preventive activities;
P_{f1}—failure probability in the period after preventive activities;
C_t—failure-related costs;
C_{rt}—preventive activity-related costs;
r—discount rate;
N—total planned service time (years);
n—year in which preventive activities are planned;
l—number of elements/risk areas/failure scenarios.

The NPV index in this form was used for turbine rotors in [5,14], where the possibilities of its application to optimize the period of diagnostic inspection with the material condition correction were presented. Due to the changing economic situation, additional changes should be introduced by applying a real discount rate that takes into account the possibility of increased inflation. Then the equation is used:

$$r_r = \frac{r-i}{i+1} \quad (21)$$

where:

r—interest rate on the financing source (credit or own funds);
i—inflation rate.

Before NPV calculation, the following assumptions were made:

- While operating the turbine, a stress control algorithm is used to maintain the values shown in Table 2 until preventive activities, after which all other start-ups can be carried out in the "on demand" mode.
- Corrective repair consists in the complete reduction in the crack. Then, in the calculations, it was assumed that $a_0 = 2$ mm, i.e., the crack is so small that its detection may not be possible due to the limited accuracy of the measurement methods, but its existence and further propagation cannot be ruled out. Moreover, it was assumed that after the correction was performed, the material consumption in the thermal grooves decreased to the level of 0 (by rolling the grooves, removing the degraded material layer).
- Initially, the costs of failure C_t due to the crack propagation are 100 conventional units and due to the life-consumption processes, they are 75 conventional units. The costs of preventive maintenance C_{rt} for both areas of the rotor amount to 5, 15 or 30 conventional units and are paid once. The costs in the following years change due to the assumed inflation rate.
- The inflation rate was assumed at the level of $i = 5\%$ (moderate inflation), the interest rate on the financing source was assumed as $r = 15.5\%$. The real discount rate is then $r_r = 10\%$, which corresponds to the values dedicated for energy investments in Poland [25].

Figure 10 shows the results of NPV calculations for the initial crack dimension $a_0 = 3$ mm and the initial wear $Z_0 = 0$. Three potential cost levels of preventive maintenance were considered. Although it is not possible to make detailed analysis of the obtained NPV values due to use of conventional units for cash flow, the course of function NPV(t) shows when it is profitable to perform preventive actions. Positive values indicate that the repair is economically justified, which is visible for almost the entire period. The highest value of the index was achieved in the 14th year for $C_{rt} = 5$ units and in the 15th year for $C_{rt} = 15$ and 30 units. The designated time intervals are optimal for performing corrective repairs for the analyzed turbine rotor.

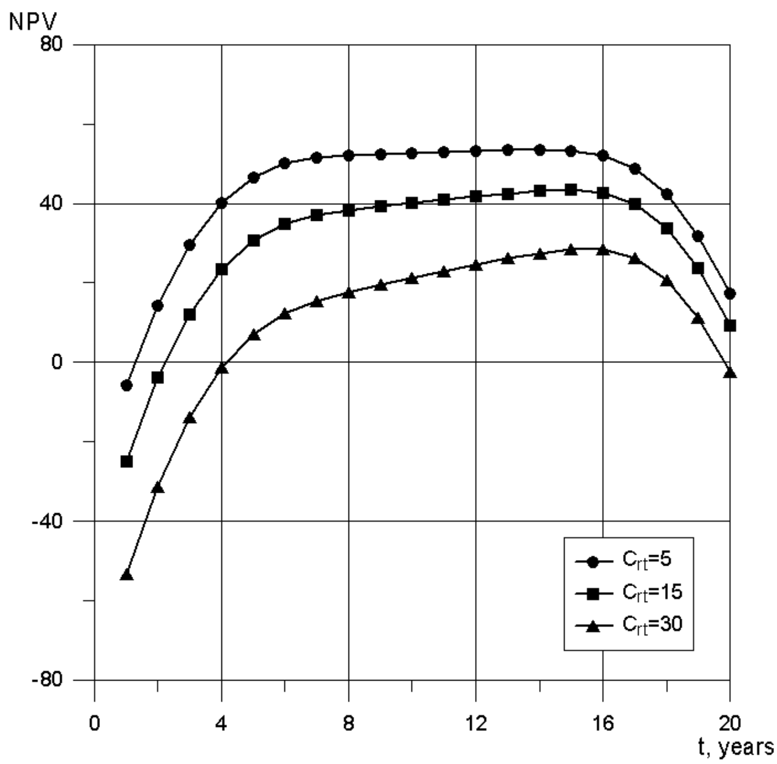

Figure 10. NPV index for $a_0 = 3$ mm and $Z_0 = 0$.

The NPV index was also calculated for the second variant, in which $a_0 = 3$ mm and $Z_0 = 0.5$ (Figure 11). The initial condition of the material in the heat grooves eliminates the turbine from the power-regulatory regime without performing a corrective repair. Such a correction is profitable only in the case of low costs ($C_{rt} = 5$), which is indicated by the positive values of the index. The optimal time for its performance is then in the 15th year. Lower values of the NPV index than those in Figure 10 result from the reduced failure risk level for crack propagation in central bore. On the basis of the analysis, it can be concluded that in the case of a high initial level of material wear, it is worth carrying out corrective repair immediately, despite the necessity to pay the costs. This will allow the turbine to be safely admitted to regulating operation, which is associated with additional profits related to fulfilling such a role in the energy system. Without this decision, preventive activities in the subsequent years will not be economically justified, and the turbine will have to be operated with low stress levels.

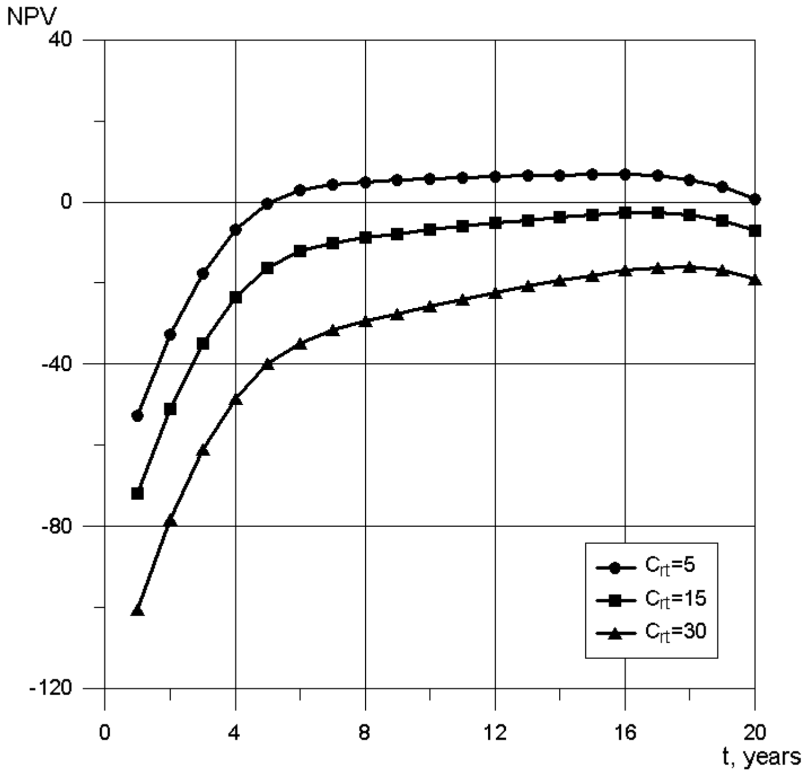

Figure 11. NPV index for $a_0 = 3$ mm and $Z_0 = 0.5$.

The article [26] by Fujiyama et al. also presents the use of risk calculations to optimize the inspection and repairs interval for steam turbines. The authors propose to analyze several separate functions in time: income from operation, maintenance cost and total risk cost. As a result, it is possible to a obtain period of time for which the implementation of preventive maintenance is justified. Using the proposed NPV indicator, the optimization is simplified by using a single function that can be determined for several elements or risk areas simultaneously.

5. Conclusions

The uncertainty of the energy resources supply observed on the global market has forced the development of alternative scenarios for the transformation of the energy generation sector. In energy systems in which the dominant sources are coal-fired plants, strategies that allow the use of these units to balance energy demand for a dozen of years must be created, especially in the face of increased flexibility of the power system with the presence of renewable sources. For this reason, it is necessary to improve a plan for their operation and maintenance, which ensures the operational safety and the required reliability of energy production. The paper presents an outline of such a strategy by focusing on one of the critical elements of the conventional power plant, which is the turbine rotor. The scenarios of the development of hazardous events leading to serious turbine failures have been presented, and the probability of their occurrence has been estimated. The online stress monitoring system for the analyzed rotor has been also described and used to create a stress control algorithm, which can be used to select the relevant heating rate during turbine start-up in real time. It is possible to use such a solution for several critical areas, which ensures, at the same time, a safe level of stress during variable loads.

Both the methodology of damage forecasting and the stress control algorithm have been used to develop a strategy for further operation of the turbine by selecting the operating conditions for the next 13 years and the assumed initial state of the material. For a longer period, it has been additionally proposed to use a dedicated NPV index, which enables optimization of the time interval for preventive maintenance. The summary of the research is the preparation of a strategy that allows effective planning of both operation and maintenance (O&M) of critical turbine elements. Similar strategies can be developed for other power unit components with different initial material consumption, planned in-service time and costs.

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