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ESTIMATED CONSUMPTION OF WORKING LIFE OF HIGH PRESSURE STEAM TURBINE ROTOR IN STATIONARY WORK

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Abstract. *Strategy of Yugoslavia energetic development forecasts the increase of electrical energy production covering its future consumption. In order to satisfy that consumption it's necessary to build new capacities (plants) or/and to prolong working life of existing capacities by revitalisation. Because of large prices of building new capacities (1200 $/kW), the aim is to prolong working life for, at least, 15 years. Costs of revitalisation of thermoenergetic plants are 20% of unit costs of building new one.*
Getting reliable estimation about spent of work life components is one of the demands that follows quality repair preparation.
The methodology, for estimated consumption of working life of steam turbine rotor, has been introduced. There have been taken into account creep, due to stationary thermal stresses, centrifugal forces and steam pressure. For estimated consumption of working life, Palmgren-Miner hypothesis based on the theory of linear damage accumulation, has been used.
The procedure is applied to the high-pressure turbine rotor (MAN-339 MW) in the power plant "Kosovo-B", unit 1, in Obilić.

1. INTRODUCTION

If we want to estimate consumption of working life of steam turbine rotor, it's necessary to study in details changes of operation parameters in operarion condition, factors that influence working life and all the matters linked to exploitation of rotor.

Steam temperature is a parameter that influences working life the most. Under steady operating conditions and by influence of complete stress field, and high values of temperature, in parts of turbine the residual working life is constantly dropping.

For estimated consumption of working life, there's a whole spectrum of hypotheses. In this work, a Palmgren-Miner's hypotheses has been used. Mathematic formulation of this hypothesis is in the form [1]:

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$$Z = \sum_{i}^{k} \frac{t_i(T_i, \sigma_i)}{t_L(T_i, \sigma_i)} + \sum_{i}^{k} \frac{n_i}{N_i} \tag{1}$$

– $t_i\,(T_i,\sigma_i)$ - the duration time of loading at temperature T_i and stress σ_i;
– $t_L(T_i,\sigma_i)$ - the time of the formation of crack at temperature T_i and stress σ_i;
– n_i - the number of load changes of the same type;
– N_i - the number of cycles of the observed load changes until the appearance of crack.

For calculation, the input data are necessary: process parameters for power plant (from diagrams in [2]), which are being statistically treated (in Fig.1.), as well as the geometrical, physical and mechanical characteristics of high steam pressure turbine rotor.

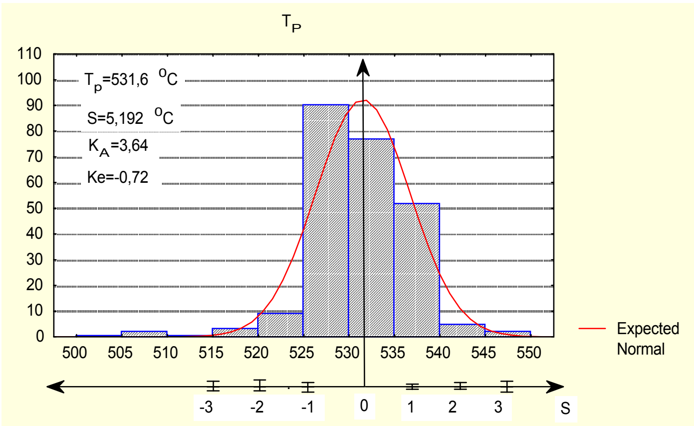

Fig. 1-a. Distribution of steam temperature in high-pressure turbine rotor

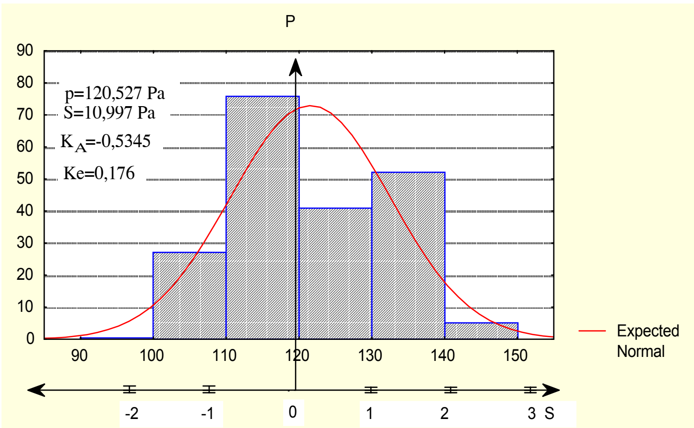

Fig. 1-b. Distribution of steam pressure in high-pressure turbine rotor

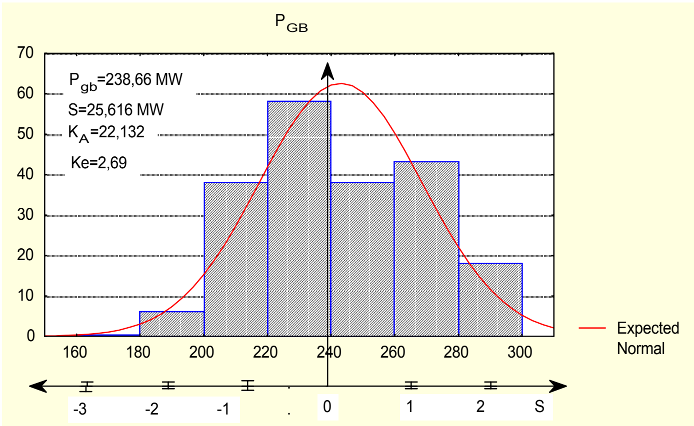

Fig. 1-c. Distribution of power in high-pressure turbine

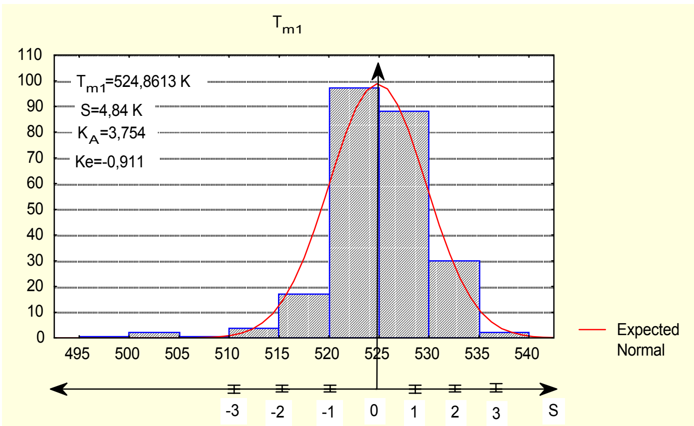

Fig. 1-d. Distribution of case temperature in high-pressure turbine

2. CALCULATION OF STRESS IN HIGH-PRESSURE TURBINE ROTOR

In the calculation of stress in high-pressure turbine rotor, it also can be treated as ideal body (Fig. 2). Calculation of stress on the rotor surface in the first disc zone.

During working, rotor is exposed to the effect of the steam pressure, thermal stress due to the differences of temperature and to the effect of centrifugal force. Stresses in polar coordinates, assuming rotor as rotation cylinder, can be written as [3]:

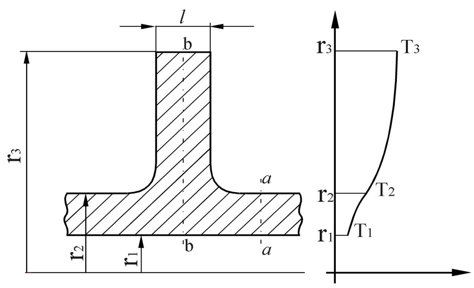

Fig. 2. Schematic layout of a rotor disk and temperature distribution

– due to effect of pressure:

$$\sigma_{rp} = -\left(\frac{r_1}{r}\right)^2 \cdot \left(\frac{r_2^2 - r^2}{r_2^2 - r_1^2}\right) \cdot (p_i - p_a) - p_a \tag{2}$$

$$\sigma_{\theta p} = \left(\frac{r_1}{r}\right)^2 \cdot \left(\frac{r_2^2 + r^2}{r_2^2 - r_1^2}\right) \cdot (p_i - p_a) - p_a \tag{3}$$

$$\sigma_{zp} = \frac{r_1^2 \cdot p_i - r_2^2 \cdot p_a}{r_2^2 - r_1^2},$$

– due to effect of centrifugal force:

$$\sigma_{r\omega} = \sigma_{r\omega_a} - \frac{(r_1/r_2)^2}{(r/r_2)^2} \cdot \left(\frac{1-(r/r_2)^2}{1-(r_1/r_2)^2}\right) \cdot (\sigma_{r\omega_a} - \sigma_{r\omega_i}) +$$
$$+ \frac{1}{8}\left(3 + \frac{\nu}{1-\nu}\right) \cdot (1-(r/r_2)^2) \cdot \left(1 - \frac{(r_1/r_2)^2}{(r/r_2)^2}\right) \cdot \rho(r_2\,\omega)^2$$

$$\sigma_{\theta\omega} = \sigma_{r\omega_a} + \frac{(r_1/r_2)^2}{(r/r_2)^2} \cdot \left(\frac{1+(r/r_2)^2}{1-(r_1/r_2)^2}\right) \cdot (\sigma_{r\omega_a} - \sigma_{r\omega_i}) +$$
$$+ \left\{\frac{1}{8}\left(3 + \frac{\nu}{1-\nu}\right) \cdot \left[1 + (r_1/r_2)^2 + \frac{(r_1/r_2)^2}{(r/r_2)^2}\right] - \frac{1}{8}\left(1 + \frac{3\nu}{1-\nu}\right) \cdot (r/r_2)^2\right\} \cdot \rho(r_a\omega)^2 \tag{4}$$

– due to temperature differences:

$$\sigma_{rT} = \frac{\beta \cdot E}{1-\nu} \cdot \left[\frac{1}{2} \cdot \left(\frac{1}{r_1^2} - \frac{1}{r^2}\right) \int_{r_1}^{r_2} r^2 \frac{dT}{dr} dr - \int_{r_1}^{r} \frac{1}{r^3} \left(\int_{r_1}^{r} r^2 \frac{dT}{dr} dr\right) dr\right]$$

$$\sigma_{\theta T} = r \cdot \sigma'_{rT} + \sigma_{rT} \tag{5}$$

$$\sigma'_{rT} = \frac{\beta \cdot E}{(1-\nu) \cdot r^3} \cdot \int_{r}^{r_2} r^2 \frac{dT}{dr} dr$$

Final expressions for calculation of stresses can be written:

$$\sigma_r = \frac{p}{y^2}\left(\frac{Y^2 - y^2}{Y^2 - 1}\right) - p + \rho\, r_2^2\, \omega^2 \frac{1}{8} \frac{3-2\nu}{1-\nu}\left(1 - \frac{y^2}{Y^2}\right)\left(1 - \frac{1}{y^2}\right) -$$
$$- \frac{\beta E \Delta T}{2(1-\nu)}\left[\frac{\ln(Y/y)}{\ln Y} - \frac{(Y/y)^2 - 1}{Y^2 - 1}\right]$$

$$\sigma_\theta = -\frac{p}{y^2}\left(\frac{Y^2 + y^2}{Y^2 - 1}\right) - p + \rho\, r_2^2\, \omega^2\left[\frac{1}{8}\frac{3-2\nu}{1-\nu}\left(1 + \frac{1}{Y^2} + \frac{1}{y^2}\right) - \frac{1}{8}\frac{1+2\nu}{1-\nu}\frac{y^2}{Y^2}\right] - \tag{6}$$
$$- \frac{\beta\ E \Delta T}{2(1-\nu)}\left[\frac{\ln(Y/y) - 1}{\ln Y} + \frac{(Y/y)^2 + 1}{Y^2 - 1}\right]$$

$$\sigma_z = -\frac{p}{Y^2 - 1} + \nu(\sigma_\theta + \sigma_r) - E\beta\ T - \frac{2}{r_2^2 - r_1^2} \int_{r_1}^{r_{22}} \left[\nu(\sigma_\theta - \sigma_{r)}) - E\beta\ T\right] \cdot r dr$$

where:

- $y = d/d_1$; $Y = d_2/d_1$,
- $d_1 = 0.1$ (m) $d_2 = 0.4$ (m) - rotor diameters,
- $\rho = 7.82 \cdot 10^{-3}$ (kg/cm^3) - rotor density,
- $\nu = 0.3$ - Poisson coefficient,
- $\omega = \frac{2 \cdot n \cdot \pi}{60}$ - angular velocity,
- $\beta = 13.7 \cdot 10^{-6}$ (1/C^0) - coefficient of linear temperature dilatation,
- $E = 17.658 \cdot 10^4$ (N/mm^2) - modulus of elasticity,
- $\Delta T = T_1 - T_2$ – temperature difference.

Equivalent stress is defined according to Misses criterion:

$$\sigma_i = \frac{1}{\sqrt{2}}\sqrt{(\sigma_\theta - \sigma_z)^2 + (\sigma_z - \sigma_r)^2 + (\sigma_r - \sigma_\theta)^2} \qquad (7)$$

Temperature of the case interior surface may be obtained from over all heat transfer balance between steam and case, and the assumption that the temperature of rotor surface T_2 is almost equivalent to the temperature of the internal case surface:

$$T_{1K} = \frac{\alpha_{pk} d_{1k} T_p + \frac{2\lambda_k}{\ln Y} T_M}{\alpha_{pk} d_{1k} + \frac{2\lambda_k}{\ln Y_K}} = T_2 \qquad (8)$$

where:

- $y_K = d_K/d_{1K}$; $Y_K = d_{2K}/d_{1K}$,
- $\lambda_K = 30$ (W/mK) – case conductivity coefficient,
- $d_{1K} = 1.38$ (m), $d_{2K} = 1.66$ (m) – case dimeter.

Temperature of the internal rotor surface is defined from the over all transfer balance between steam and rotor.

$$T_1 = T_2 - \frac{\alpha_{PR}}{2\lambda_R} d_2 (T_p - T_2) \ln Y \qquad (9)$$

λ_R=30 (W/mK) – rotor conductivity coefficient

Heat transfer coefficients are determined on basis of empirical correlations. They can be obtained by expressions [4;5]:

- from steam to case:

$$\alpha_{pK} = 360 + 6540 \cdot (P^*_{Gb})^{0,8} \;\; [W/m^2K] \qquad (10)$$

taking into account:

$$P^*_{Gb} = \frac{P_{Gb}}{P_{Gb\,max}}$$

$$P_{Gb}\ \bar{P}_{Gb} + k_i \cdot S_{P_{Gb}}$$

Where k_i – is middle position of a strip in the normal distribution diagram (Fig.3), and s – standard deviation, and so we can get:

$$\alpha_{pK} = 360 + \frac{6540}{P_{Gb\,max}}(\bar{P}_{Gb} + k_i \cdot S_{P_{Gb}})^{0,8} \tag{11}$$

– from steam to rotor:

$$\alpha_{PR} = \frac{\lambda_p \cdot N_u}{d_2} \tag{12}$$

where:

$$N_u = 0,038\left(\frac{R_e}{2}\sqrt{\frac{r_k - r_2}{r_2}}\right)^{0,8} \cdot P_r^{\,0,33} \tag{13}$$

$$R_e = \frac{W_o\, d_2}{\nu_p} = \frac{d_2\,\pi\, n}{60}\frac{d_2}{\nu_p} = \frac{d_2\,\pi\, n}{60}\frac{d_2}{\mu_p}\rho \tag{14}$$

$$P_r = \frac{\nu_p}{a} = \frac{\mu_p \cdot c_p}{\lambda} \tag{15}$$

Thermo-physical properties of steam are calculated by expressions:

– dynamical viscosity:

$$\mu = \mu_1 + 353 \cdot \rho + 676,5 \cdot \rho^2 + 102,1 \cdot \rho^3 \tag{16}$$

$$\mu_1 = 80,4 + 0,407 \cdot t$$

– conductivity coefficient:

$$\lambda = \lambda_1 + (103,51 + 0,4198 \cdot t - 2,771 \cdot 10^{-5} \cdot t^2) \cdot \rho + 2,1482 \cdot 10^{14} \cdot \rho^2 \cdot t^{-4,2} \tag{17}$$

$$\lambda_1 = 17,6 + 5,87 \cdot 10^{-2} \cdot t + 1,04 \cdot 10^{-4} \cdot t^2 + 4,51 \cdot 10^{-8} \cdot t^3$$

– Heat capacity:

$$p = 80 \div 160 \text{ bar}$$

$$(480 < t < 520)^0\text{C}$$

$$c_p = 1,1655 \cdot 10^{-10} \cdot p^4 - 2,5641 \cdot 10^{-8} \cdot p^3 + 1,2314 \cdot 10^{-5} \cdot p^2 + 3,3128 \cdot 10^{-3} \cdot p + 2,1284$$

$$(520 < t < 560)^0\text{C} \tag{18}$$

$$c_p = -1,7483 \cdot 10^{-10} \cdot p^4 + 8,3916 \cdot 10^{-8} \cdot p^3 - 7,3718 \cdot 10^{-6} \cdot p^2 + 3,7524 \cdot 10^{-3} \cdot p + 2,1342$$

– specific densities are assumed from tables for mean values of steam pressure and temperature, because of small changes in domain of steam temperature and pressure changes during stationary work.

3. Consumption of Working Life

Based on loading estimation, considering balance of heat, we can estimate consumption of working life in stationary operating conditions. Degree of fatigue of material for the mean value of quantity is proportional to both the surface area of strip A_i, below the distribution curve (Fig.3.), and the time period, can be displayed as:

$$\Delta Z_i = \frac{A_i t_p}{t_L(T_i, \sigma_i)} \tag{19}$$

– t_p - complete working time which fits the above given distribution (7267 h)

– $t_L(T_i, \sigma_i)$ - working life function, for T_i and σ_i (h) [1].

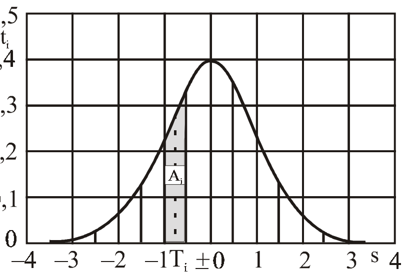

Fig. 3. Normal parameter distribution

$$t_L(T, \sigma) = \exp\left[\frac{2.3 \cdot 10^3}{bT} \ln \frac{\sigma}{a} - 46\right] \quad (h) \tag{20}$$

where:

– a and b – constants of rotor material [1].

Total degree of fatigue is:

$$Z = \sum_{k_i=-3,75}^{k_i=+3,75} \Delta Z_i \tag{21}$$

Values of fatigue of material are given in table 1:

Table 1.

k_i	tp	T_2	T_1	p(N/mm^2)	t_L	σ_{i2}	A_i	ΔZi
0.25	506.385	505.998	489.634	9.484	2.02E+14	24.533	0.192	6.89939E-12
–0.25	504.154	503.776	487.792	9.355	3.12E+14	24.045	0.192	4.4687E-12
0.75	508.615	508.221	491.479	9.613	1.32E+14	25.017	0.149	8.20235E-12
–0.75	501.924	501.554	485.954	9.226	4.86E+14	23.555	0.149	2.22797E-12
1.25	510.846	510.443	493.327	9.742	8.68E+13	25.499	0.092	7.70067E-12
–1.25	499.693	499.332	484.120	9.097	7.63E+14	23.061	0.092	8.76343E-13
1.75	513.076	512.666	495.178	9.871	5.75E+13	25.977	0.045	5.6861E-12
–1.75	497.463	497.110	482.289	8.968	1.21E+15	22.565	0.045	2.7066E-13
2.25	515.307	514.888	497.032	10.000	3.84E+13	26.452	0.016	3.03092E-12
–2.25	495.232	494.888	480.462	8.839	1.93E+15	22.065	0.016	6.02076E-14
2.75	517.537	517.111	498.890	10.129	2.58E+13	26.925	0.0047	1.32589E-12
–2.75	493.002	492.666	478.639	8.710	3.12E+15	21.562	0.0047	1.09588E-14
3.25	519.768	519.334	500.750	10.258	1.74E+13	27.395	0.0011	4.59162E-13
–3.25	490.771	490.445	476.820	8.581	5.08E+15	21.056	0.0011	1.57331E-15
3.75	521.998	521.557	502.784	10.387	1.28E+13	27.643	0.0002	1.13325E-13
–3.75	488.541	488.223	475.005	8.452	8.37E+15	20.547	0.0002	1.73631E-16
							ΣZi =	4.13344E-11

6. Conclusion

The purpose of this work is to develop a method for estimated consumption of working life of some steam turbine parts, under stationary working conditions, based on available records in form of diagrams.

So, there is a possibility of introducing an estimated consumption of working life as a new reliability parameter of turbine, to the existing power plants. It can be useful as one of the criteria for needs of turbo-unit components maintenance, according to working status. The purpose of this work is to provide basics for continuous calculation of estimated consumption of working life for steam turbine parts, by using computers techniques for acquisition and treating of found results.

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PROCENA UTROŠENOG RADNOG VEKA ROTORA TURBINE VISOKOG PRITISKA PRI STACIONARNOM RADU

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Strategija razvoja energetike Jugoslavije, predviđa porast proizvodnje električne energije radi zadovoljenja njene buduće potražnje. Za zadovoljenje te potražnje nužno je da se izgrade novi kapaciteti ili/i da se postojećim kapacitetima produžI radni vek njihovom revitalizacijom. Zbog visoke cene gradnje novih kapaciteta (1200 $/kW) težnja je da se revitalizacijom postojećih produži njihovo korišćenje za bar 15 godina. Troškovi revitalizacije termoelektrana su 20% jediničnih troškova novih.

Dobijanje pouzdanih procena o utrošenom radnom veku komponente, jedan je od zahteva koji prate kvalitetnu pripremu remonta.

U radu je prikazana metoda za proračun utrošenog radnog veka rotora turbine visokog pritiska. U obzir je uzeto puzanje, kao posledica stacionarnih termičkih napona, centrifugalnih sila i pritiska pare. Za procenu utrošenog radnog veka korišćena je Palmgren-Minerova hipoteza zasnovana na teoriji o linearnoj akumulaciji oštećenja.

Navedena procedura primenjena je na rotoru turbine visokog pritiska (MAN-339 MW) u termoelektrani "Kosovo-B", blok 1, u Obiliću.