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METHODOLOGICAL APPROACH TO INTEGRITY ASSESSMENT AND SERVICE LIFE OF ROTATING EQUIPMENT AT HYDROPOWER PLANT – TURBINE SHAFT

METODOLOŠKI PRISTUP PROCENI INTEGRITETA I VEKA ROTACIONE OPREME U HIDROELEKTRANI – TURBINSKO VRATILO

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Keywords
- turbine shaft
- stress concentration
- corrosion
- fatigue
- crack

Abstract

Horizontal Kaplan turbines, were installed in 10 hydroelectric generating units at "DJERDAP 2", total power of 28 MW. Taking into account the fact that fatigue started to occur in many components, the research is conducted regarding the influence of loading regime, work condition and type of damage, on the service life of equipment components. Analytical calculation established values of bending stresses acting on the turbine shaft due to fatigue, corrosion, and stress concentration, that are above 25 MPa in flanges exposed to water, and over 40 MPa for flanges exposed to "corrosive water", which is also protected by epoxy fibres, possibly causing the occurrence of surface cracks on the transition radius between the cylindrical part of the shaft and the flange.

Ključne reči
- turbinsko vratilo
- koncentracije napona
- korozija
- zamor
- prslina

Rezime

Horizontalne Kaplanove turbine su instalirane u hidroelektrani „DJERDAP 2“, 10 jedinica ukupne snage 28 MW. Uzimajući u obzir činjenicu da se zamor pojavio u mnogim komponentama, sprovedeno je istraživanje uticaja režima opterećenja, radnih uslova i vrste oštećenja na njihov radni vek. Korišćenjem analitičkih izraza, izračunato je da savojni naponi koji deluju na turbinsko vratilo, a pod dejstvom zamora, korozije i koncentracije napona, prelaze vrednost 25 MPa na prirubnici koja je izložena dejstvu vode, odnosno, 40 MPa na prirubnici koja je izložena "korozivnoj vodi“, a zaštićena je epoksi premazom, što može da izazove pojavu površinskih prslina u prelaznom radijusu između cilindričnog dela vratila i prirubnice.

INTRODUCTION

Turbines and hydro-mechanical equipment of hydropower plants are subjected to loads that occur during the manufacture of parts and assembly of equipment (residual stresses), during regular exploitation (stationary and non-stationary loading) and during non-regular exploitation (non-stationary dynamic loading). By taking into account influences of exploitation (corrosion, erosion, cavitation), it is clear that loading of certain components and equipment as a whole, cannot be expressed by a simple mathematical function; i.e. it cannot be fully presented by a model in which the parameters uniformly change with the change of operational conditions. Therefore, only the testing of components and structures enables the estimation of their state. This is how the necessary data for determination of the causes of material degradation, are obtained in the case of horizontal Kaplan turbines, 10 units installed at the hydropower plant "DJERDAP 2", Fig. 1, /1/, with a nominal power of 28 MW.

Taking into account that fatigue occurs in many components of turbine and hydromechanical equipment, research regarding influence of loading regime, working environment and type of damage, on the integrity has been conducted. This research is initiated since the fracture of the hydroelectric generating set A6 turbine shaft in the area of high stress concentration, at the transition radius R80, between the flange and the runner hub, Fig. 2. During exploitation, under high-cycle fatigue loading in the corrosive environment, initial cracks have occurred and merged, forming 20-30 mm long cracks, which is confirmed by the presence of corrosion products at the smooth fracture surface. When the size of the turbine shaft cross-sectional area has reduced below the critical value, the fracture occurred. This part of the fractured surface is wrinkled and free of corrosion products.

Non-destructive testing methods (visual testing - VT, dye-penetrant testing - PT, magnetic particle testing - MT, ultrasonic testing - UT) provided the information that there is a large number of surface cracks in all shafts along the circumference, at the transition radius area. Crack propaga

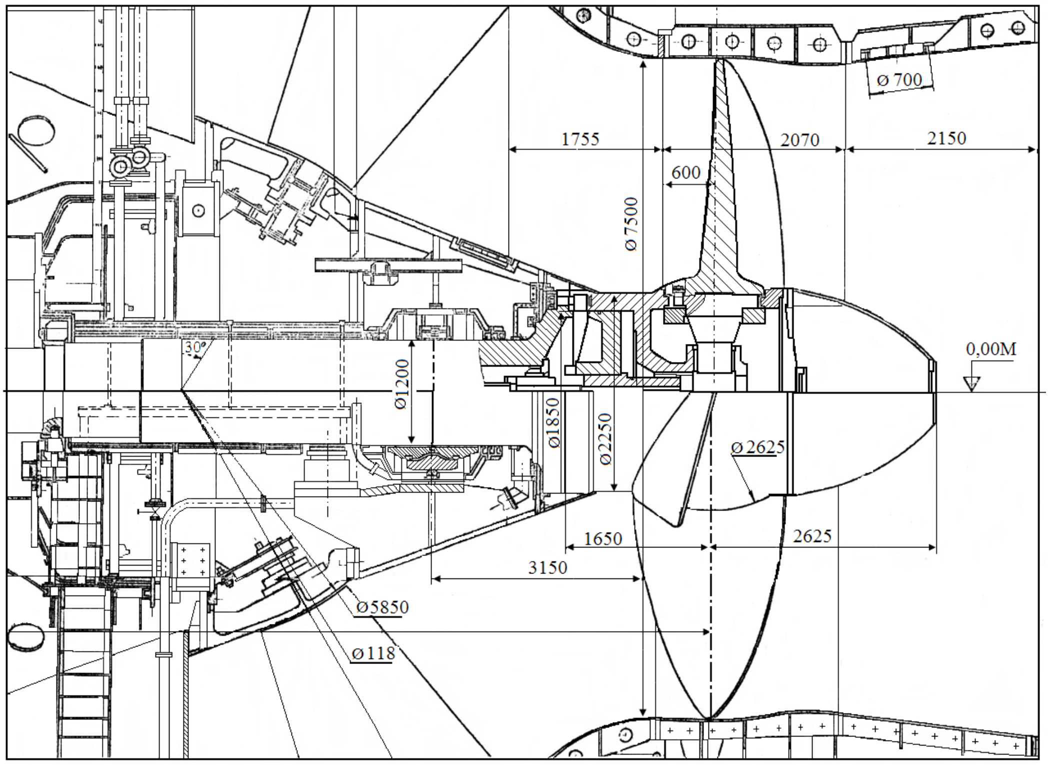


Figure 1. Appearance of the Kaplan turbine, nominal power 28 MW.
Slika 1. Shema Kaplanove turbine, nominalne snage 28 MW

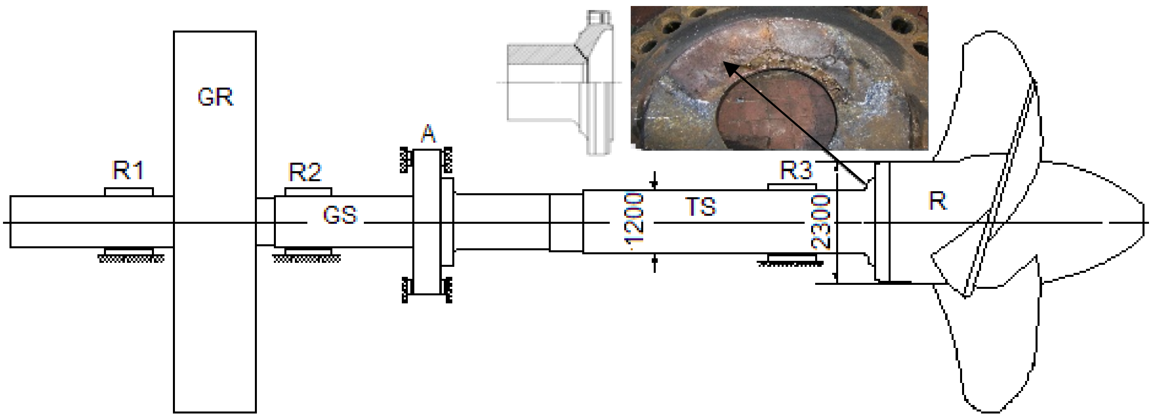


Figure 2. Appearance of basic components of the generator, appearance of the fracture surface
Slika 2. Shema osnovnih komponenata generatora i izgled prelomne površine

tion at the transition radius of the turbine shaft of the hydroelectric generating set A8, detected through visual, penetrant and magnetic particle testing is shown in Fig. 3.

Basic parameters of installed generating sets PL-15/826-G-750 are: maximum head, 12.75 m; engineering head, 7.45 m; minimum head, 5 m; turbine power at the engineering head, 28 MW; diameter of runner, 7500 mm; number of revolutions, 62.5 min^{-1}; number of runner blades, 4. Basic components are: GR-generator rotor, GS-generator shaft, TS-turbine shaft, R-runner, R1, R2 and R3-radial bearings, A-axial bearing, Fig. 2. Hollow turbine shafts have been assembled by welding together three separate parts. Cylindrical parts are made of steel 20GS (GOST), while the flange area and flange itself are made of steel 20GSL.

Fatigue under corrosive conditions is a process of metal damage due to the occurrence of initial cracks and their growth caused by variable loads. One of its components could be more influential than other (fatigue or corrosion), but generally, both are significant. Influence of loading frequency is closely connected to dependency of corrosion to time, while the effect of geometric stress concentration should also be taken into account. The corrosive medium reduces the service life in comparison to the situation when the medium is air, because it is not only that the notch acts as a stress concentrator, but the effect of geometric stress concentration is harmful as well.

A contemporary concept of determination of service life for components and structures, apart from standard Wöhler curves, also requires the knowledge regarding the process of degradation due to fatigue. The fatigue degradation process is very complex and still insufficiently explored. This multi-step process (submicroscopic, microscopic and macroscopic) unfolds in 3 stages: the first is the period until the occurrence of an initial crack, followed by crack growth and fracture. Thus, nowadays various hypotheses regarding the fatigue process are being used for practical calculations that refer to the estimation of service life, that are based on assumptions and on the calculation of damage accumulation. These hypotheses in the simplest way define the rate of damage progression during one cycle of operating stress with the specified amplitude, as well as the measure of damage accumulation, being caused by a certain number of alterations of operating stresses with various amplitude levels, /2/.

On the basis of monitoring damage progression, all hypotheses could be divided into those based on fatigue damage accumulation and those based on laws of crack growth rate. Damage accumulation hypotheses could be divided into fundamental and phenomenal. Fundamental hypotheses embrace the whole fatigue process in a complex way, and on the basis of established criteria regarding the fatigue mechanism, predict the dependency of stress on the number of cycles, $\sigma = \sigma(N)$, for components subjected to variable loads. Phenomenal (cumulative) hypotheses are based on fatigue life for a specific component and on the measure of damage (D). Phenomenological hypotheses take into account the initial crack that already exists and define the process of material damaging through crack growth. Expressions for crack length (a) and its critical value when fracture occurs (a_k) are based on fracture mechanics theory. Nevertheless, crack growth rate depends on crack growth increment per one stress cycle (da/dN). This rate is being obtained on the basis of experimental data gathered during the monitoring of the fatigue process.

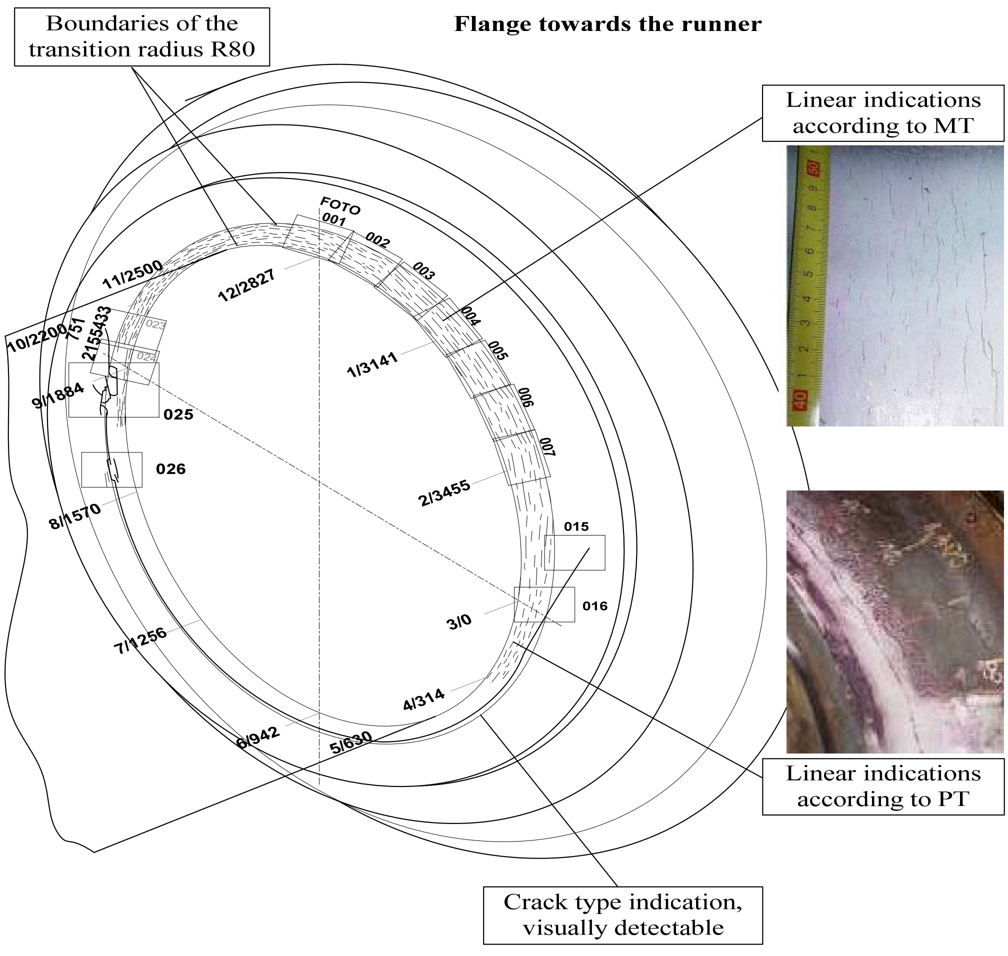


Figure 3. Turbine shaft of the hydroelectric generating set A8, with VT, PT and MT results.
Slika 3. Turbinska osovina hidroelektričnog sklopa generatora A8, sa rezultatima VT, PT i MT

METHODOLOGY OF FAILURE ANALYSIS

Important information for the improvement of the design methods for bearing parts and elements of bearing structures, as well as for the improvement of properties of existing materials and technologies for their processing and development of new materials, have enabled the creation of damage and failure analyses of parts and elements of bearing structures, /3/. Damage and failure analyses enable the development of new technical solutions and testing methods in the prototype phase. In order to determine and prevent the causes of damage and failure, adequate analyses are being made, and that is a process that requires a systematic approach, Fig. 4.

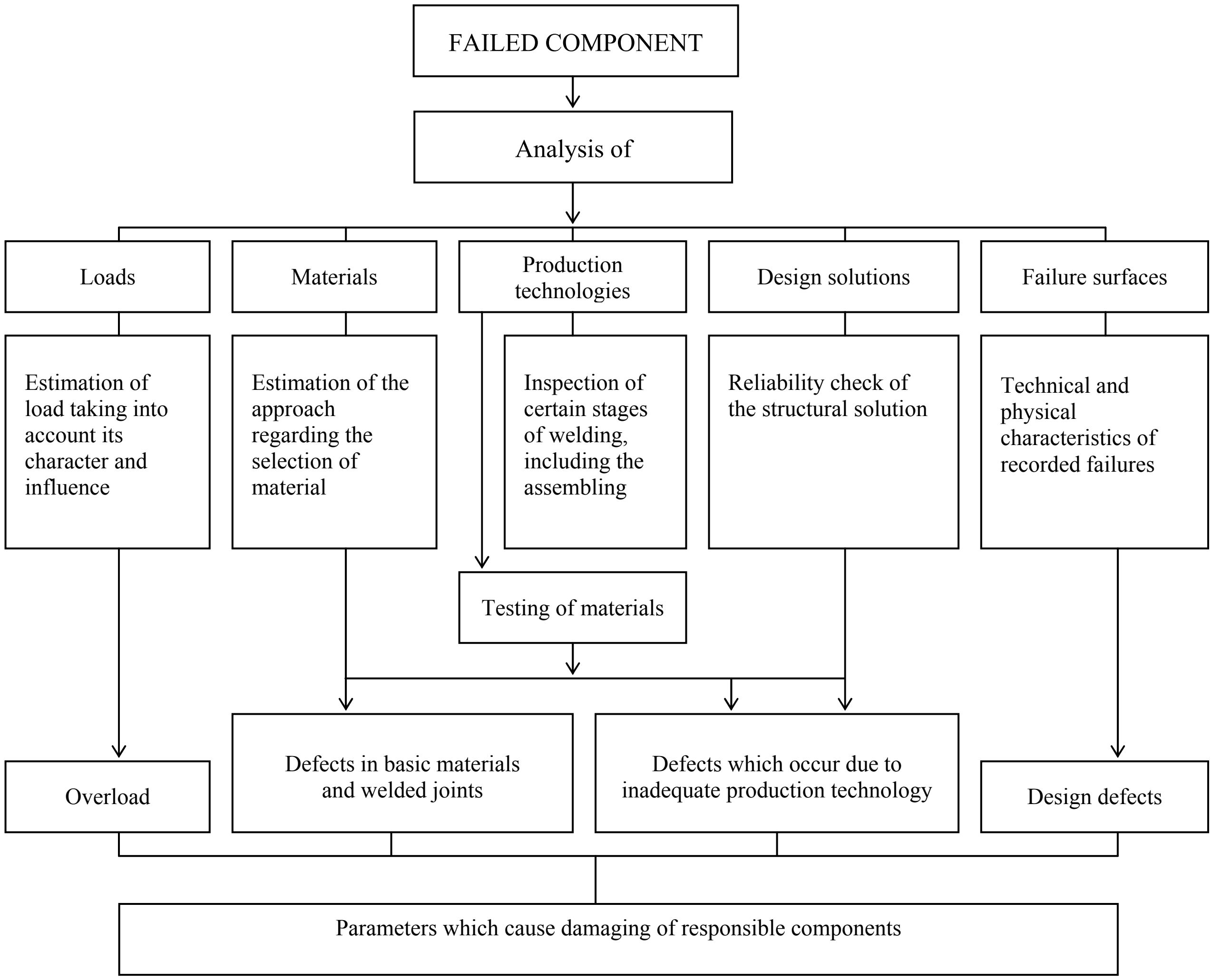
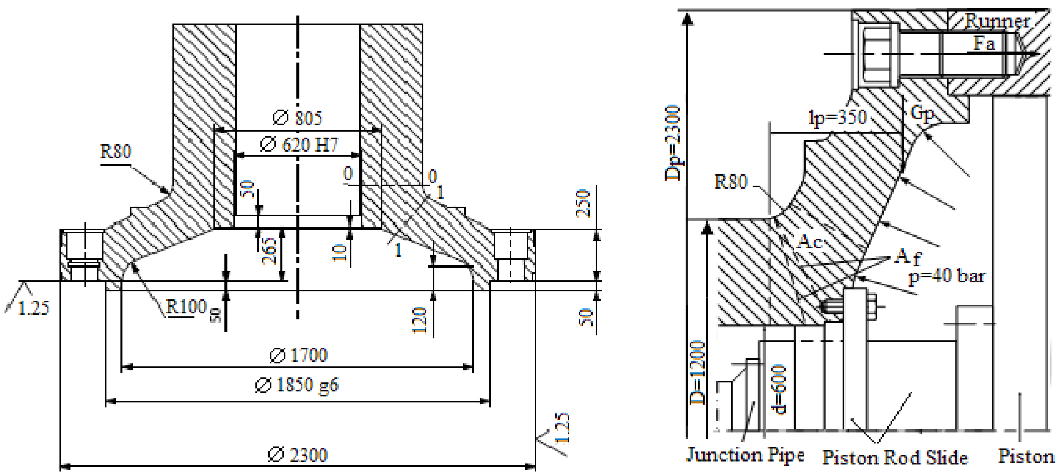


Figure 4. Process analysis of damaging and failure of turbine components.
Slika 4. Proces analize oštećenja i loma turbinskih komponenata

CALCULATION OF TURBINE SHAFT STRESSES

The turbine shaft shown in Fig. 5 (F-flange), besides bending, is subjected to tensile stress due to the effect of the hydraulic force on the turbine runner. Pressure of oil in the servo motor of the runner in the closing stroke and axial hydraulic force loads tend to subject the flange to bending. Due to the transfer of the force, the shaft is subjected to torsion as well.



a) Flange section of the shaft, critical cross-sections 0-0 and 1-1   b) Critical cross-section

Figure 5. Appearance of critical cross-sections at the flange section of the turbine shaft.
Slika 5. Shema kritičnih poprečnih preseka u preseku spojnice turbinske osovine

Cyclic loads, to which the turbine shaft is subjected, in combination with the corrosive environment (leakage of water through the seal, poor execution and non-renewal of the corrosion protection) had caused the occurrence of the corrosion fatigue on the transition radius (location where the value of the stress concentration factor is 3).

Corrosion fatigue damages, as far as stress concentration is concerned, act practically like cracks (stress concentration value is ranging from 3 to 6), /4/. Critical cross-section of the turbine shaft is subjected to, /5, 6, 7/:

− Axial hydraulic force $F_a = 5.5426 \cdot 10^6$ N
− Moment of torsion M_t:

$$M_t = \frac{P}{2\pi n} = 4.278 \cdot 10^6 \text{ Nm} \qquad (1)$$

where: $P =$ 28 000 kW, turbine power; $n =$ 62.5 min^{-1}, number of revolutions of the turbine shaft.

− Bending moment that occurs due to the action of axial hydraulic force and the force that occurs due to the oil pressure in the cylinder of the servo motor of the runner $M_{ob} =$ 337 768 Nm, according to data provided by the turbine manufacturer,
− Bending moment that occurs due to the weight of the runner and the weight of the part of the flange as far as the critical cross-section:

$$M_b = G_{rk} \cdot l_{rk} + G_p \cdot l_p = 1964943 \text{ Nm} \qquad (2)$$

where: $G_{rk} =$ 941760 N, weight of the runner; $l_{rk} =$ 2050 mm, distance from the centre of gravity of the runner to the critical cross-section; $G_p =$ 98100 N, weight of the flange with bolts as far as the critical cross-section; $l_p =$ 350 mm, distance from the centre of gravity of the flange to the critical cross-section.

Analytical Calculation of Stresses in the Critical Cross-Section of the Turbine Shaft

- Static stresses in the critical cross-section

Components of static stresses are as follows, /5, 6, 7/:

− Tensile stress due to the action of axial hydraulic force:

$$\sigma_t = k_{tens} \frac{F_a}{A_c} = 14.3 \text{ MPa} \qquad (3)$$

where: $k_{tens} = 2.19$, stress concentration factor for

$$\frac{D_f}{D} = \frac{2300}{1200} = 1.916\,; \quad \frac{R}{D} = \frac{80}{1200} = 0.066 \qquad (4)$$

$$A_c = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(1{,}2^2 - 0{,}6^2) = 0.848 \text{ m}^2 \qquad (5)$$

where: A_c critical cross-section area (Fig. 5a).

− Bending stress due to the action of axial force and force that occurs due to pressure in the servo motor of the runner

$$\sigma_0 = \frac{M_0}{W_0} = \frac{M_0}{h^2/6} = 22.52 \text{ MPa} \qquad (6)$$

where: W_0-moment of resistance per unit length of critical cross-section.

− Torsional stress:

$$\tau_t = k_{tor} \frac{M_t}{W_t} = k_{tor} \frac{M_t}{\frac{\pi D^3}{16}\left[1 - \left(\frac{D}{d}\right)^4\right]} = 20.85 \text{ MPa} \qquad (7)$$

where: W_t-polar moment of resistance per length unit of the critical cross-section; $k_{tor} =$ 1.55-stress concentration factor during torsion for ratios given by Eq.(4).

− Equivalent static stress at the critical cross-section:

$$\sigma_{eq} = \sqrt{(\sigma_z + \sigma_0)^2 + 4\tau_t^2} = \\ = \sqrt{(14.3 + 22.52)^2 + 4 \cdot 20.85^2} = 55.6 \text{ MPa}$$

- Cyclic stress at the critical cross-section

The cyclic stress at the critical cross-section occurs due to the bending moment caused by weights of the runner and of the part of flange as far as the critical cross-section, /5, 6, 7/:

$$\sigma_{-1} = k_{ten} \frac{M_t}{W_{ccs}} = k_{ten} \frac{M_t}{\frac{\pi D^3}{32}\left[1 - \left(\frac{D}{d}\right)^4\right]} = 24.46 \text{ MPa} \qquad (8)$$

where: W_{ccs}-moment of resistance per unit length of critical cross-section; $k_{ten} =$ 1.98–stress concentration factor during bending for ratios given by Eq.(4).

- Factor of safety in relation to corrosion fatigue

Factor of safety of the turbine shaft in relation to corrosion fatigue and, in conditions of cyclic loading of amplitude $\sigma_a =$ 24.4 MPa and corrosion, is obtained through the use of the following equation, /5, 6/:

$$S_\sigma = \frac{\sigma_{-1} - \psi_\sigma(\sigma_m + \sigma_{M0})}{\sigma_a} \qquad (9)$$

where: $\sigma_{-1} =$ 26.5 MPa-permanent corrosion fatigue strength of cast steel 20 GSL during the action of the alternating load in a corrosive environment; ψ_σ–coefficient that takes into account the asymmetry of the cycle and is equal to the ratio of the corrosion fatigue strength and tensile strength.

$$\psi_\sigma = \frac{\sigma_{-1}}{\sigma_{UTS}} = \frac{26.5}{480} = 0.0552 \qquad (10)$$

where: σ_m–maximum value of static exploitation stresses at the calculated cross-section; σ_a–amplitude of cyclic stresses σ_{-1}; σ_{M0}–residual stresses present after casting and heat treatment have not been taken into account ($\sigma_{M0} =$ 0 MPa) because there were no exact values available.

Permanent corrosion fatigue strength is being determined experimentally by inspecting the samples in water and correcting the obtained results through the use of dimensional factors. For this analysis, experimental results regarding the testing of 20 GSL material during the calculation process for the cover of the runner for HPP "DJERD–AP I" have been used. The obtained equation contains the correction factors, /5, 6, 7/:

$$\log \sigma_{-1} = A - B \log N$$

$$\log \sigma_{-1} = 2.787 - 0.155 \cdot \log N \qquad (11)$$

where: N–true number of loading cycles.

$$N = nT = 62.5\cdot 60\cdot 163411 = 0.63\cdot 10^9 \text{ cycles} \quad (12)$$

where: n = 62.5 min^{-1}–number of revolutions per minute of the shaft; T = 163411 h–service life of the hydro-electric generating set until breakdown. After inserting appropriate values into the equation, the following is obtained:

$$\log\sigma_{-1} = 2.787 - 0.155\cdot\log(0.63\cdot 10^9)$$
$$\log\sigma_{-1} = 1.423 \quad (13)$$
$$\sigma_{-1} = 26.5 \text{ MPa}$$

Factor of safety in relation to corrosion fatigue:

$$S_\sigma = \frac{\sigma_{-1} - \psi_\sigma(\sigma_m + \sigma_{M0})}{\sigma_a} = 0.96 \quad (14)$$

Factor of safety is less than S_σ = 1.1, the value predicted by manufacturer's designation.

- Effect of stress concentration and corrosion on fatigue strength

At locations of sudden changes of shape of loaded structural components, the local increase of stress (stress concentration) occurs. Level of stress increase is defined by the ratio of the maximum local ($\sigma_{\max}$) and nominal (σ_{nom}) stress, called the theoretical stress concentration factor, /8/:

$$k_f = \frac{\sigma_{\max}}{\sigma_{nom}} \quad (15)$$

Values of the theoretical stress concentration factor for various shapes of components and various loading type are presented in Peterson's paper, /8/.

Fatigue strength due to the action of the corrosive environment is defined by the following factor:

$$k_{kor} = \frac{\sigma_{\max(-1)kor}}{\sigma_{\max(-1)}} \quad (16)$$

where: $\sigma_{\max(-1)kor}$, $\sigma_{\max(-1)}$–fatigue strengths of smooth specimens in the corrosive environment and in the air atmosphere, respectively.

Fatigue strength in the corrosive environment depends on the number of cycles, but also on the length of exposure period of elements to the corrosive environment. Thus the effect of stress change frequency is significant. In relation to that, a Wöhler corrosion fatigue curve is a constantly descending line, therefore, permanent fatigue strength practically does not exist, /5/.

The joint effect of corrosion and stress concentration can be expressed by the following coefficient:

$$k_{fkor} = k_f + \frac{1}{k_{kor}} - 1 \quad (17)$$

where: k_f = 1.98–effective stress concentration factor for testing in the air atmosphere, k_{kor} = 0.5 (for σ_{UTS} = 480 MPa)–coefficient of the effect of corrosion for smooth specimens. As far as the turbine shaft is concerned, value of stress concentration coefficient, including the effect of corrosion, is k_{fkor} = 2.98.

Through the analysis of the corrosion fatigue during the asymmetric cycle it has been determined that mean tensile stress unfavourably affects, or in other words, significantly decreases the dynamic durability amplitude. Mean pressure stresses favourably affect the resistance to corrosion fatigue. This effect is commonly used for method of surface strengthening of elements that operate in the corrosive environment.

Decrease of fatigue strength of an element with respect to fatigue strength of the smooth specimen is calculated through the use of the overall fatigue strength reduction factor:

$$k_{rf} = \left(\frac{k_f}{k_3} + \frac{1}{k_{kor}} - 1\right)\frac{1}{k_{po}k_A} \quad (18)$$

Coefficients for the calculation of k_{rf} are:
k_f = 1.98 – stress concentration coefficient,
k_3 = 0.6 – cross-section coefficient,
k_{po} = 1 – coefficient that takes into account technological methods of surface strengthening,
k_A = 1 – coefficient of anisotropy for steel castings,
k_{kor} = 0.5 – corrosion coefficient.

By putting values of influential coefficients into Eq.(18) one gets k_{rf} = 4.28 for the turbine shaft.

Numerical Calculation of the Stress State in the Critical Cross-Section of the Shaft

Calculation of the stress state of the turbine shaft with the transition radius R80 between the cylindrical and flange area of the shaft has been carried out by the turbine manufacturer LMZ (Russia), UCM Resita (Romania), Institute for Materials Testing and LOLA Institute (Serbia). All calculations show that the values of bending stresses, which occur due to the action of the load and corrosion fatigue, including stress concentration factors, are higher than 25 MPa for flanges subjected to "corrosive water" and can cause the occurrence of surface cracks on the transition radius between the cylindrical and flange area of the shaft.

FRACTURE MECHANICS PARAMETERS

Fatigue Crack Growth Rate Tests

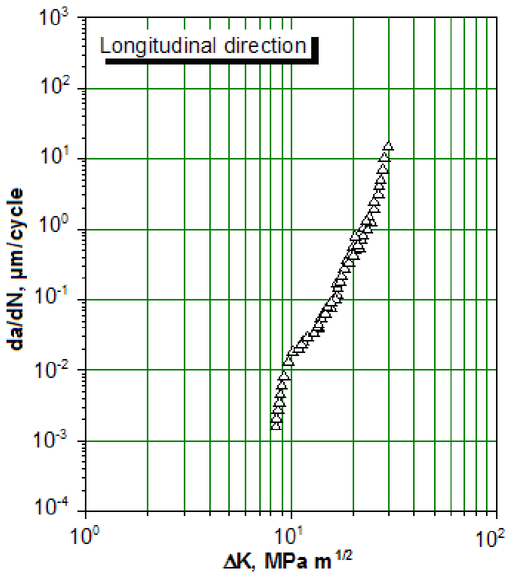
Behaviour of the material subjected to alternating loading in the presence of a crack is defined by the Paris-Erdogan equation. Results of tests performed in order to determine crack growth rate (da/dN) in relation to stress intensity factor (ΔK) are shown in Fig. 6 and Table 1.

Obtained results for fatigue threshold ΔK_{th} indicate that the casting of the flange area of the turbine shaft in the longitudinal direction (crack is transversely oriented) is more resistant to the propagation of the existing crack, which confirms the conclusions reached through the analysis of the fracture area of the turbine shaft. Effect of notch orientation on fatigue crack growth rate da/dN is directly related to parameters in the Paris equation, coefficient C and exponent m. For the analysis, the stress intensity factor range ΔK = 10 MPa·m$^{1/2}$ has been taken, because that value resides in the area of stable crack growth in da/dN versus ΔK graph, for which the Paris law /9, 10/ is applicable:

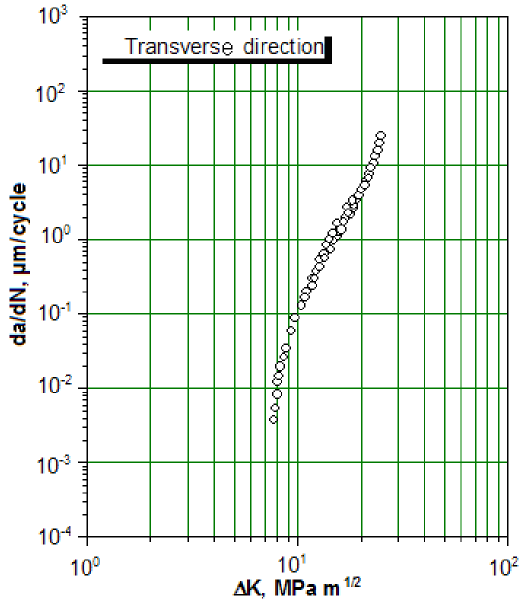
$$\frac{da}{dN} = C\cdot\Delta K^m \quad (19)$$

Table 1. Fracture mechanics parameters for specimens taken from the casting in different directions.
Tabela 1. Parametri mehanike loma za epruvete uzete sa odlivka, u različitim pravcima

Specimen direction	Fatigue threshold ΔK_{th}, MPa·m$^{1/2}$	Coefficient C, (m/cycle)/(MPa√m)^{mp}	Coefficient m_p	da/dN, m/cycle at ΔK = 10 MPa·m$^{1/2}$
Longitudinal	8.7	$3.3 \cdot 10^{-11}$	3.02	$5.62 \cdot 10^{-08}$
Transversal	7.4	$3.2 \cdot 10^{-11}$	3.10	$6.36 \cdot 10^{-08}$
Minimum allowed value of K_{IC} for 20GSL at $T \leq 0$°C is K_{IC} = 41–44 [MPa·m$^{1/2}$], GOST 977 - 88				



a) Longitudinal direction of the shaft flange casting



b) Transverse direction of the shaft flange casting

Figure 6. Diagram of the dependence between crack growth rate and stress intensity factor range.
Slika 6. Dijagram zavisnosti brzine rasta prsline i raspona faktora intenziteta napona

Service Life Estimation on the Basis of Fracture Mechanics Test Results

On the basis of test results, through the use of Paris-Erdogan equation, the number of loading cycles until the occurrence of the crack with the critical length could be obtained, in the longitudinal and transverse direction of the flange section of the turbine shaft.

$$\Delta K = \Delta\sigma f\left(\frac{a}{W}\right)\sqrt{\pi a} \tag{20}$$

where: ΔK–stress intensity factor range; $\Delta\sigma$–stress range,

$$\Delta\sigma = \frac{\sigma_{\max}}{2} = \frac{\sigma_{eq}+\sigma_a}{2} = \frac{55.6+24.46}{2} = 40.03 \text{ MPa} \tag{21}$$

$f(a/W)$–factor depending on the length of initial crack and thickness of the test specimen,

$$f(a/W) = 1.12 \tag{22}$$

Number of loading cycles needed in order to reach the critical crack length a_c, when as the initial crack length a_0 the value of required surface roughness after sand blasting (a_0 = 2.5 μm) is adopted:

$$N = \frac{1}{\left(\frac{m-2}{2} C f^m \pi^{\frac{m}{2}} \Delta\sigma^m\right)} \left[\frac{1}{a_0^{\frac{m-2}{2}}} - \frac{1}{a_c^{\frac{m-2}{2}}}\right] \tag{23}$$

- Service life estimation for the flange section of the shaft in the longitudinal direction

Data for the flange section of the turbine shaft in the longitudinal direction are as follows:

- $a_0 = 0.0000025$ m,
- $C = 3.3 \times 10^{-11}$ (m/cycle)/(MPa√m)^{mp},
- $f = 1.12$,
- $m_p = 3.02$.

According to standard GOST 977-88, for the minimum allowed value of critical stress intensity factor K_{IC} = 41–44 (MPa·m$^{1/2}$) the critical crack length is, /5/:

$$a_c = \frac{1}{\pi}\left(\frac{K_{Ic}}{\sigma_{\max}}\right)^2 = \frac{1}{\pi}\left(\frac{41 \text{ MPa}\times\text{m}^{1/2}}{80.06 \text{ MPa}\cdot 1.12}\right)^2 = 0.0666841 \text{ m} \tag{24}$$

The number of loading cycles until the crack reaches the critical length in the longitudinal direction:

$$N_L = 6.28 \cdot 10^8 \text{ cycles}$$

Number of loading cycles during a one-year period:

$$N_{1y} = n_h \cdot 60 \cdot 7000 = 62.5 \cdot 60 \cdot 7000 = 2.625 \cdot 10^7 \text{ cycles} \tag{25}$$

Number of loading cycles during a twenty-two-year period:

$$N_{22y} = 22 \cdot 2.625 \cdot 10^7 = 5.775 \cdot 10^8 \text{ cycles} \tag{26}$$

where: n_h–number of revolutions of turbine shaft; n_h = 62.5 min^{-1}; 60–number of minutes in an hour; 7000–overall number of operating hours of the hydroelectric generating set during one year; 22–number of years of operation of the hydroelectric generating set.

Estimated service life of the turbine shaft:

$$n = \frac{N_L}{N_1} = \frac{6.28 \cdot 10^8}{2.625 \cdot 10^7} = 23.95 \text{ years} \tag{27}$$

- Service life estimation for the flange section of the shaft in the transverse direction

Data for the flange section of the turbine shaft in the transverse direction are as follows:
- $a_0 = 0.0000025$ m
- $a_c = 0.0666841$ m
- $C = 3.2 \times 10^{-11}$ (m/cycle)/(MPa√m)^{m_p}
- $f = 1.12$
- $m_p = 3.10$.

Number of loading cycles until the crack reaches the critical length in the transverse direction:

$$N_T = 8.81 \cdot 10^7 \text{ cycles}$$

Estimated service life of the turbine shaft:

$$n = \frac{N_T}{N_1} = \frac{8.81 \cdot 10^7}{2.625 \cdot 10^7} = 3.357 \text{ years}$$

On the basis of fracture mechanics test results and service life estimation for the flange section of the turbine shaft in the longitudinal and transverse direction, taking into account the growth of the crack from the initial (a_0) to critical length (a_c), the following can be concluded:
- from the dependency curve between the crack growth rate (da/dN) and range of stress intensity factor (ΔK), it can be concluded that the number of loading cycles required for the initiation of cracks is within the suitable range for the above mentioned type of material, as well as that crack growth rate increases afterwards, which could be explained by the non-homogeneity of the casting and lower values of critical stress intensity factor (K_{IC}),
- experimentally obtained values of fatigue threshold (ΔK_{th}) are within the range for the above mentioned type of material,
- explanation for 163 411 operating hours (22 years) of the shaft with a frequency of 1.04 s^{-1} and a number of loading cycles $N_{22y} = 5.775 \cdot 10^8$, in relation to the obtained number of loading cycles until the critical length of the crack is reached in the transverse direction at the flange section of the turbine shaft, $N_T = 8.81 \cdot 10^7$ cycles, is based on the existence of anti-corrosion protection, which slowed down the process of propagation of initial cracks until the seal was damaged, causing the water to leak and exposing the turbine shaft to accelerated corrosion.

CONCLUSIONS

On the basis of theoretical considerations, test results regarding the fatigue strength, fracture mechanics parameters and effect of stress concentration in the corrosive operating environment, it can be concluded that cracks and fracture of the turbine shaft of the hydro-electric generating set 6, occurred due to the following:
- high stress concentration on the transition radius of the flange area of the turbine shaft (R = 80 mm), and operating in a corrosive environment, which caused the occurrence of initial cracks in that area. Additional analytical and numerical calculations showed that, in order for the stress concentration at transition radiuses of flange sections of turbine shafts to be reduced, the value of transition radius has to be R = 100 mm, which has been applied during repair welding/surface welding of other damaged shafts, /11/.
- insufficient area of the critical cross-section of the transition radius and shaft operation in the corrosive environment, as well as exposure to the leaking water, which caused the bending stress and other components of stress to grow beyond acceptable values,
- inadequate structural solution concerning the seal caused the exposure of the flange area of the shaft to the corrosive environment (leaking water),
- corrosion protection poorly executed and not renewed,
- absence of periodic non-destructive testing.

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