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# Occurrence of cracks due to inadequate turbine shaft construction

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**Abstract** After several decades of constant exploitation of the horizontal bulb turbine, which is an integral part of the hydroelectric power unit, empirically, the formation of a crack occurs in the turbine shaft due to the influence of corrosion, erosion and cavitation.

Through experimental tests and calculations it has been determined that values of bending stresses of the turbine, which occur due to the action of fatigue and corrosion, as well as stress concentration, are bigger than 25 MPa for flanges exposed to water, and in other case bigger than 40 MPa for flanges exposed to `corrosive water` and can cause the occurrence of surface cracks on the transition radius between the cylindrical part of the shaft and the flange. It has been determined that stress values in the zone under the influence of bending stresses were bigger than allowable values, which led to the occurrence of many cracks due to fatigue corrosion. One of those cracks caused the failure of the shaft and of the whole turbine.

**Keywords** fatigue, crack, turbine shaft, stress concentration

## 1. INTRODUCTION

Turbine loads originate during the production of components and equipment assembling (residual stresses), during the process of performing functional requirements in exploitation (stationary and dynamic loads) and during the disturbed process of exploitation (non-stationary dynamic loads). It's clear that component and equipment loads can't be expressed by a simple mathematical function knowing that unpredictable influence of corrosion, erosion and cavitation during exploitation should be taken into account. That's the reason why extensive researches,

tests and inspections of hydropower plant equipment have been undertaken [1 - 7].

Turbine shaft was assembled by welding the cylindrical body of the hollow shaft (outer diameter 1200 mm, internal diameter 600 mm) to the flange, made of 20GSL steel. During welding heat-affected zone has been created, which overlapped the transition diameter. Residual stresses were accumulated at the external surface of the transition area from the shaft to the flange. Anti-corrosive protection in the shaft flange area has been exposed to water. After several hundred thousand hours of use, a huge loss of oil from the regulating system of hydroelectric generating set was noticed. After the exclusion of the hydroelectric generating set from the service and visual inspection of all suspicious locations a crack, 2100 mm long, was detected, through which the turbine oil from the servo motor leaked, on the transition radius

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R80 from the cylindrical part of the shaft toward the runner hub.

## 2. ASSESSMENT AND ANALYSIS OF STRENGTH AND FRACTURE OF THE TURBINE SHAFT RADIUS

Turbine shaft is subjected to tensile stress due to the effect of the hydraulic force on turbine runner.

Pressure of oil in the servo motor of the runner in the closing stroke and axial hydraulic force load subject the flange to bending. The weight of the runner and of the shaft itself subject the shaft to cyclic bending. Due to the transfer of the force the shaft is subjected to torsion as well.

Combination of cyclic loads and corrosive environment (leakage of water through the seal, poor execution and non-renewal of the corrosion protection) led to the occurrence of the corrosion fatigue on the transition radius (location where value of the stress concentration factor is 3). Corrosion fatigue damages, as far as stress concentration is concerned, act like cracks (stress concentration value is ranging from 3 to 6).

High-cycle fatigue conditions in a corrosive environment the initiation of cracks occurred, and their joining led to the formation of 20-30 mm long cracks, which was confirmed by the existence of corrosion products on the smooth fracture surface. When the load bearing area of the turbine shaft, under low-cycle fatigue conditions, fell under the critical value in the crack growth area the fracture occurred. That segment of the area is embossed and deprived of corrosion products.

Calculation regarding the critical cross-section of the shaft has been carried out through the use of the Theory of Elasticity and data provided by the manufacturer. Taking into account the fact that the fracture is irregularly shaped, it has been adopted that the critical cross-section is positioned at the end of the cylindrical part of the shaft.

Critical cross-section of the turbine shaft is subjected to [7]:

- Axial hydraulic force  $F_a=5.5426 \cdot 10^6$  N,
- Moment of torsion  $M_t=4.278 \cdot 10^6$  Nm,
- ( $P=28000$  kW-turbine power and  $n=1.04166$  s<sup>-1</sup> - turbine shaft number of revolutions)
- Bending moment  $M_o=337768$  Nm (occurs due to the action of the axial hydraulic force and

force that occurs due to the oil pressure in the cylinder of the servo motor of the runner),

- Bending moment  $M_o=1964943$  Nm (occurs due to the weight of the runner and weight of the part of the flange as far as the critical cross-section):

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

$$\sigma_z = \alpha_z \frac{F_a}{A_k} = 14,3 \text{ MPa} \quad (1)$$

where:  $\alpha_z=2.19$ -stress concentration factor during tension and  $A_k$  critical cross-section area.

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

$$\sigma_o = \frac{M_o}{W_o} = \frac{M_o}{h^2/6} = 22,52 \text{ MPa} \quad (2)$$

where:  $W_o$ -moment of resistance per length unit of the critical cross-section.

- Torsional stress:

$$\tau = \alpha_t \frac{M_t}{W_t} = \alpha_t \frac{M_t}{\frac{\pi \cdot D^3}{16} \left[ 1 - \left( \frac{D}{d} \right)^4 \right]} = 20,85 \text{ MPa} \quad (3)$$

where:  $W_t$ -polar moment of resistance per length unit of the critical cross-section,  $\alpha_t = 1.55$

- stress concentration factor during torsion for

$$\frac{D_p}{D} = \frac{2300}{1200} = 1,916 \quad \frac{R}{D} = \frac{80}{1200} = 0,066 \quad (4)$$

-Equivalent static stress at the critical cross-section:

$$\sigma_m = \sqrt{(\sigma_z + \sigma_o)^2 + 4\tau^2} = \sqrt{(14,3 + 22,52)^2 + 4 \cdot 20,85^2} = 55,6 \text{ MPa} \quad (5)$$

Cyclic stress at the critical cross-section occurs due to the bending moment caused by weights of the runner and of the part of the flange as far as the critical cross-section:

$$\sigma_a = \alpha_s \frac{M_s}{W_s} = \alpha_s \frac{M_s}{\frac{\pi \cdot D^3}{32} \left[ 1 - \left( \frac{D}{d} \right)^4 \right]} = 24,46 \text{ MPa} \quad (6)$$

where:  $W_s$ -moment of resistance per length unit of the critical cross-section,  $\alpha_s=1.98$ -stress concentration factor during bending for

$$\frac{D_p}{D} = \frac{2300}{1200} = 1,916 \quad \frac{R}{D} = \frac{80}{1200} = 0,066 \quad (7)$$

## 2.1 Corrosion fatigue factor of safety

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:

$$S_\sigma = \frac{\sigma_{-1} - \psi_\sigma \cdot (\sigma_m + \sigma_{MO})}{\sigma_a} \quad (8)$$

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

$$\psi_\sigma = \frac{\sigma_{-1}}{R_m} = \frac{26,5}{480} = 0,0552 \quad (9)$$

$\sigma_m$ -maximum value of static exploitation stresses at the calculated cross-section, eq. (5).

$\sigma_a$ -amplitude of cyclic stresses, eq. (6).

$\sigma_{MO}$ -residual stresses present after casting and heat treatment have not been taken into account ( $\sigma_{MO}=0$  MPa) because there were no exact values available.

Permanent corrosion fatigue strength is being determined experimentally by testing the samples in the water and correcting the obtained results through the use of dimensional factors. Factor of safety is less than  $S_\sigma=1.5$ , the value predicted by the manufacturer's designation.

## 2.2 Effect of Stress Concentration and Corrosion on Fatigue Strength

At locations of sudden changes of shape of loaded structural components local increase of stress (stress concentration) occurs. Level of stress increase is defined by the ratio of the

maximum local ( $\sigma_{max}$ ) and nominal ( $\sigma_{nom}$ ) stress, which is being called the theoretical stress concentration factor [6]:

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} \quad (10)$$

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

$$k_{kor} = \frac{\sigma_{max(-1)kor}}{\sigma_{max(-1)}} \quad (11)$$

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.

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

$$K_{fkor} = K_f + \frac{1}{k_{kor}} - 1 \quad (12)$$

where:  $K_f=1.98$ -effective stress concentration factor for testing in the air atmosphere,  $k_{kor}=0.5$  (for  $R_m=480$  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  $K_{fkor}=2.98$ .

Corrosion fatigue analysis of asymmetric cycle it has been determined that mean tensile stress unfavorably affects, or in other words significantly decreases the dynamic durability amplitude. Mean pressure stresses favorably affect the resistance to corrosion fatigue.

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:

$$K_D = \left( \frac{K_f}{k_3} + \frac{1}{k_{kor}} - 1 \right) \frac{1}{k_{po}k_A} \quad (13)$$

$K_f=1.98$ -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.

At the end,  $K_D=4.28$  for the turbine shaft. Factor  $K_D$  can be used in formulas for determination of the factor of safety.

### 2.3 Analysis of fracture mechanics parameters

To assess material behaviour and structural integrity in the presence of cracks, fracture mechanics parameters are used. Also, fracture mechanics parameters are used for better determine the tendency to crack growth, critical conditions for fracture development, material resistance to rapid crack propagation.

During 1961, Paris, Gomey, and Anderson have first proposed that the crack growth rate,  $da/dN$ , might be correlated with the stress intensity factor range,  $\Delta K$ , when the material is exposed to variable loading of constant amplitude.

Paris-Erdogan equation according to Paris' law [8,9] is the fatigue crack growth rate, which is defined as a function of the stress intensity factor range  $\Delta K$ :

$$\frac{da}{dN} = C \cdot (\Delta K)^m \quad (14)$$

where  $C$  and  $m$  are material parameters. These constants are related to the fatigue properties, material microstructure, stress, frequency, cyclic loading, and applied temperature. The constant  $m$  is usually between 2 and 4, near 4 for metallic materials. For materials with low static fracture toughness, the value of material parameter  $m$  can be as high as 10.

The diagram in Fig. 1 shows three different regimes of crack growth. The fatigue life can be divided into two parts: crack initiation phase (regime A) and propagation phase (regime B). Fatigue crack propagation behaviour is typically described in terms of crack growth rate or crack length extension per cycle of loading ( $da/dN$ ) plotted against the stress intensity factor range or the change in stress intensity factor from the maximum to the minimum load [10].

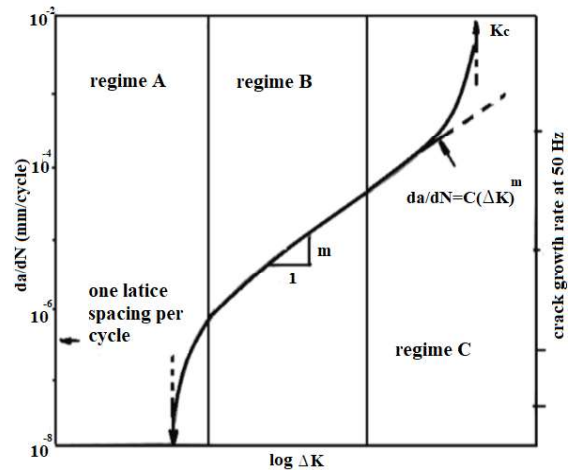


Fig. 1. Different regimes of stable fatigue crack propagation [11]

Using CRACKTRONIC pulsator, in load control conditions [12, 13], the fatigue crack growth rate  $da/dN$  and fatigue threshold  $\Delta K_{th}$  are performed.

Crack growth is monitored by measuring potential drop by strain gauge RUMUL RMF A-5, measuring 5 mm length, located on the specimen face surface, Fig. 2.

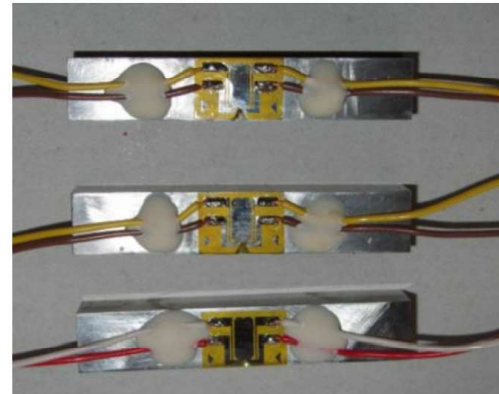


Fig. 2. Strain gauge RUMUL RMF A-5 located on the Charpy specimen (dimensions: 55x10x10 mm) face surface

Strain gauges RUMUL RMF A-5 of 5 mm length are cemented on machined specimens, allowing crack growth monitoring by FRACTOMAT device, based on electrical potential of gauge and connected with instrumentation, Fig. 3. The measuring gauge, a thin resistant measuring foil, is cemented on the specimens in the same way as classical strain gauges. As the crack

grows under the measuring foil causing it to rip, it traces the fatigue crack tip, changing foil electrical resistance linearly with the crack length [14].

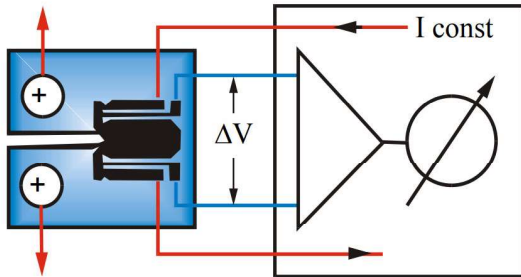


Fig. 3. Scheme of measurement foil and crack growth detection

By reducing the relationship between the fatigue crack growth rate per cycle  $da/dN$  and the range of stress intensity factor  $\Delta K$  is determined the coefficient  $C$  and the exponent  $m$ .  $\Delta K$ , which depends on the geometry of the specimen, the length of the crack, and on the range of variable force,  $\Delta P$ , should be added to the fatigue crack growth rate for current crack length  $a$  [12].

Table 1 shows values of the fatigue threshold  $\Delta K_{th}$  and material parameters-coefficient  $C$  and exponent  $m$  for fatigue crack growth, according the standard ASTM E647 [15] for determining the stress intensity factor range.

Table 1. Fracture mechanics parameters for specimens taken longitudinally and transversally from the casting

Spec.	Fatigue threshold $\Delta K_{th}$ , MPa m <sup>1/2</sup>	Coeff. C	Coeff. m	da/dN, m/cycle at $\Delta K=10$ MPa m <sup>1/2</sup>
Longitudinal	8,7	$3.0 \cdot 10^{-11}$	2	$5.62 \cdot 10^{-08}$
Transversal	7.4	$3.2 \cdot 10^{-11}$	4	$6.36 \cdot 10^{-08}$

### 3. CONCLUSION

Calculation of fatigue strength, fracture mechanics parameters and effect of the stress concentration in the corrosive operating environment it can be concluded that cracks and fracture of the turbine shaft before its predicted service lifetime passed occurred due to

the inadequate construction solution regarding the production of the shaft.

More precisely, the mentioned refers to occurrence of initial cracks in corrosive environment and insufficient moment of resistance in 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 to grow beyond acceptable values.

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