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Estimation of Lifetime Duration for a Lever Pin of Runner Blade Operating Mechanism using a Graphic – analytic Method

In this paper are presented a graphic - analytic method that can be used to estimate the fatigue lifetime duration for an operating mechanism lever pin to a Kaplan turbine. The presented calculus algorithm is adapted from the one used by Fuji Electric to made strength calculus in order to refurbish a Romanian hydropower plant, equipped with a Kaplan turbine. The graphic part includes a 3D fatigue diagram for rotating bending stress designed by Fuji Electric specialists.

Keywords: *fatigue, pin, algorithm, Kaplan, lifetime duration*

1. Introduction

The fatigue phenomenon became a central problem in strength calculus process. From Wőhler, the first researcher that defines the fundamental lows of fatigue, until the present, the fatigue concepts register a few changes.

Wőhler was the first that shown the influence of sudden variation of section over mechanical strength of structure [1]. In present, the fatigue phenomenon is strictly related by simultaneous existence of three factors:

1. Tensile stresses;
2. Plastic deformation;
3. Variables stress.

The experts believe that, the absence of a single one of these three factors makes the stress not to be considered a fatigue one [2].

To estimate fatigue lifetime duration for any mechanical structure became a modern engineering desideratum, especially when we talking about expensive machines.

Lately, some of the mechanical structures that suffered from fatigue were the turbines, especially to the runner blade level of Kaplan turbines [6].

To repaired or change a part of a Kaplan turbine include much expensive aspects, that's way, a correct design ensure a substantial savings for beneficiaries.

The algorithm presented in this paper was applied for a Kaplan turbine which is still in operation, the operating regime being the real one, while the fatigue cycles are a theoretical one, used only to madden a simulation and a comparative study between two estimated fatigue lifetime calculus method (Figure 1).

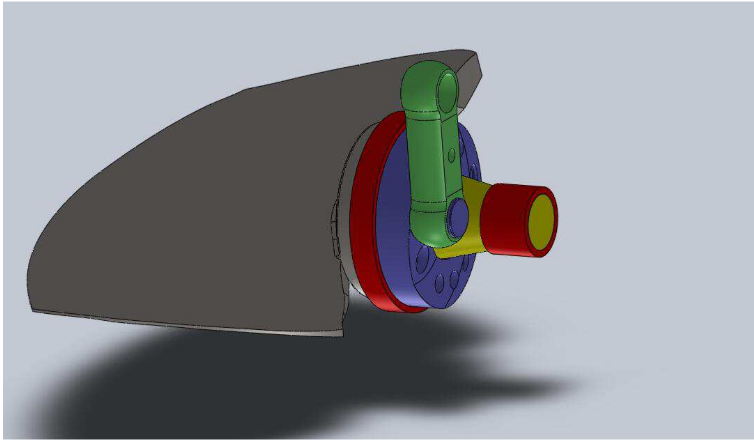

Figure 1. Runner blade – operating mechanism assembly.

2. The calculus algorithm and results

The calculus was made for three work regime:

- regime I : $\varphi = +17{,}5^\circ$ and $H = 25\,m$;
- regime II : $\varphi = +10^\circ$ and $H = 25\,m$;
- regime III : $\varphi = +10^\circ$ and $H = 31{,}4\,m$.

Where:

- H – the water operating available head;
- φ - the rotor blade angle.

The steps taken to estimate lifetime duration for lever pin are [3]:

a. the hydraulic force calculus that acting on a single blade

$$F_A = \frac{\frac{\pi}{4} \cdot \left(D_r^2 - D_h^2\right)}{z} \cdot \rho \cdot g \cdot H \qquad (1)$$

Where: D_r – runner diameter;
D_h – runner hub diameter;
g – gravitational acceleration;
z – runner blade number

b. centrifugal force calculus

$$C = \frac{G}{g} \cdot \omega^2 \cdot R_{cg} = m \cdot \left(\frac{\pi \cdot n}{30} \right)^2 \cdot R_{cg} \qquad (2)$$

Where: m – assembly weight;
n – normal and runaway speed;
R_{cg} – the assembly gravity center radius.

The centrifugal force calculus was made also for normal speed as so runaway speed. The results are given in Table 1.

Table 1

Speed n [rpm]	Centrifugal force C [kN]
71,43	4134,349
124	12459,136

c. the servo-engine force calculus to minimal oil pressure:

$$F_s = \frac{\frac{\pi}{4}\left(D_1^2 - D_2^2\right) \cdot P_{\min}}{z} \qquad (3)$$

Where: D_1 - the outer diameter of the piston;
D_2 – the rod piston diameter;
P_{min} – the minimum oil presure.

d. the force calculus that acting on lever pin

$$F_{pl} = \frac{F_s}{1{,}1} \qquad (4)$$

e. the reaction calculus from blade spindle bearings – are made in acordance with Figure 2.

$$R_A = \frac{(l_1 + l_2) \cdot F_A}{l_1} + \frac{(l_1 + l_5) \cdot G}{l_1} + \frac{l_3 \cdot F_s^{'}}{l_1 \cdot 1{,}1} \tag{5}$$

$$R_B = \frac{l_2 \cdot F_A}{l_1} + \frac{l_5 \cdot G}{l_1} + \frac{l_4 \cdot F_s^{'}}{l_1 \cdot 1{,}1} \tag{6}$$

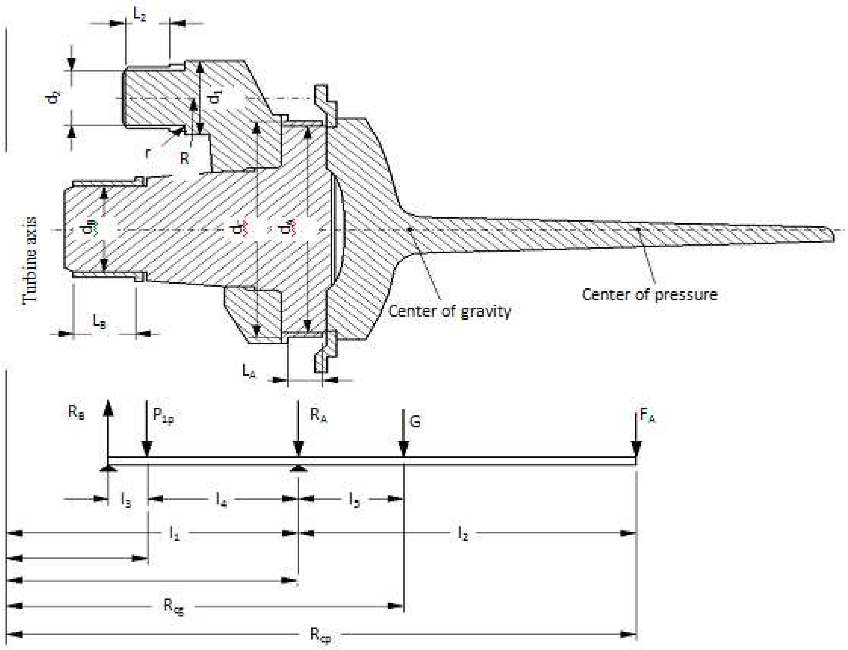

Figure 2. Load diagram of blade spindle

f. the friction moment of the shaft

$$T_f = \frac{\mu}{2}\left(R_A \cdot d_A + R_B \cdot d_B + C \cdot d_c\right) \tag{7}$$

Where: μ - the friction coeficient of material;
d_A, d_B, d_C – the interest areas diameter.

g. the calculus of maximum actuator force neded for blade closure

$$F_s^{'} \cdot R \cdot \cos\theta = T_f + T_{h\min} \quad (8)$$

After calculus we obtained:

$$F_s^{'} = F_{SC} \quad (9)$$

h. The calculus of maximum actuator force needed for blades opening are made with relation (8)

i. The servo-engine force cover rate to minim oil pressure:

$$\Delta = \frac{F_S}{F_{SO}} \quad (10)$$

j. Calculation of the two reactions from blade trunnion bearings

k. The pressure calculus in blade trunnion bearings:

$$P_A = \frac{R_A}{d_A \cdot L_A} \qquad P_B = \frac{R_B}{d_B \cdot L_B} \quad (11)$$

Where: L_A and L_B are the bushes lenghts where the forces from blade spindle bearings are distributed.

l. The pressure on lever pin:

$$P_{lp} = \frac{F_{SC}}{d_2 \cdot L_2 \cdot 1{,}1} \quad (12)$$

m. The amplitude alternating force and principal force during frequency regime (primary regime)

$$\Delta F_s = \frac{F_{sc} + F_{so}}{2} \quad (13)$$

$$F_{mean} = \frac{F_{sc} - F_{so}}{2} \quad (14)$$

<u>BENDING STRESSES</u>

n. the alternant and principal stress on lever pin

$$\sigma_a = \frac{M_a}{W} = \frac{\Delta F_s \cdot r_2}{\frac{\pi \cdot d_2^3}{32}} \tag{15}$$

Where r_2 is the coresponding radius to diameter d_2.

$$\sigma_m = \frac{M_{mean}}{W} = \frac{F_{mean} \cdot r_2}{\frac{\pi \cdot d_2^3}{32}} \tag{16}$$

<u>TORSIONAL STRESSES</u>

o. The alternant and principal stress on lever pin

$$M_{\tau a} = \mu \cdot \Delta F_s \cdot \frac{d_2}{2}$$
$$M_{\tau m} = \mu \cdot F_{mean} \cdot \frac{d_2}{2} \tag{17}$$

$$W_\tau = \frac{\pi \cdot d_2^3}{16} \tag{18}$$

$$\tau_a = \frac{M_{\tau a}}{W_\tau}$$
$$\tau_m = \frac{M_{\tau m}}{W_\tau} \tag{19}$$

p. The stress concentrator factor α:

The value of stress concentator factor rezults from calculus made after FKM Guideline [5]: $\alpha = 2{,}155$.

r. The reduction factor due fatique stresses β:

According to the Fuji Electric algorithm [3], the calculus formula is:

$$\beta = 1 + q \cdot (\alpha - 1) \tag{20}$$

Where q is the corection factor:

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} \tag{21}$$

Where: r - is the connection radius betwen lever pin and lever body;
a – Neuber constant.

s. The equivalent alternativ stress:

$$\beta \cdot \sigma_a = \sigma_{equ} \tag{22}$$

From Fuji diagram are extract the value for:

$$\beta \cdot \sigma_m$$

$$\beta \cdot \sigma_a$$

result:

$$\sigma_{ech} = \sigma_a \cdot \frac{\beta \cdot \sigma_m(\infty)}{\beta \cdot \sigma_a(\infty)} \tag{23}$$

With equivalent stress, from Wholer diagram to Figure 3 [3] are determinated the cycles number to whom should the lever resist: N = 10^6 cycles.

t. The fatigue lifetime duration:

$$\Pi = \frac{N}{\sum n_i} \tag{24}$$

Where: n_i – the total stress cycles number considered on one year.

As is shown in Table 2 the calculus was also made for four type of operating mode:

- In frequency regulation ;
- In warheads;
- Half units in frequency regulation and the other half in warheads;
- Run-of-river

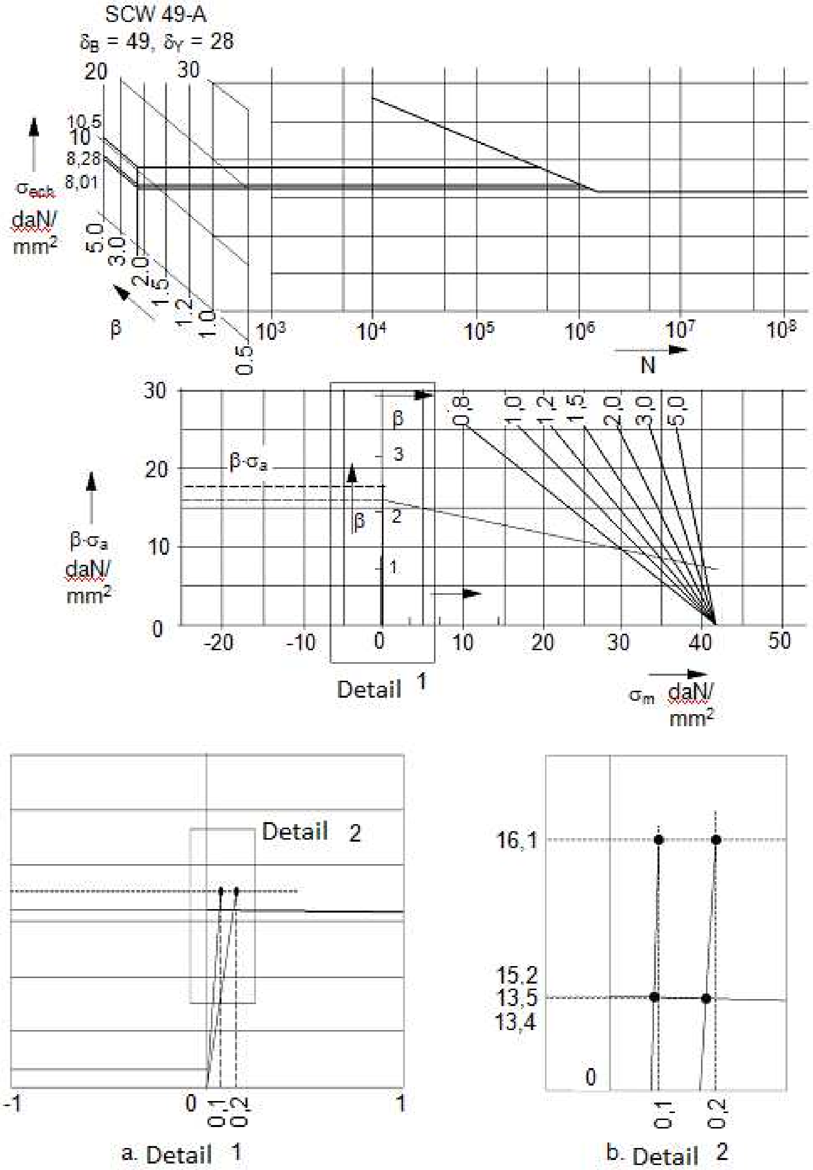

Figure 3. Fatigue diagram from rotating bending stress [3]

Table 2.

Lifetime duration Π_i [years] \ Operating regime		Regime I (H = 25 m; φ = +17,5°)	Regime II (H = 25 m; φ = +10°)	Regime III (H = 31,4 m; φ = +10°)
Π (in frequency regulation)	a) after Fuji Electric	50		17,5
	b) after internal method	16,83		5,89
Π_1 (in warheads)	a) after Fuji Electric	500		175
	b) after internal method	192,3		67,3
Π_2 (1/2 unit. in frequency regulation and 1/2 unit. in warheads)	a) after Fuji Electric	90,9		31,8
	b) after internal method	31		10,8
Π_3 (run-of-river)	a) after Fuji Electric	1000		350
	b) after internal method			

For every operating mode and every regime, the fatigue lifetime duration calculus, were made, besides Fuji algorithm, by an internal method [4] used by the Romanian hydraulic engineers from the studied objective.

3. Conclusions

After this analyze can be concluded that:

- calculation algorithm is derived from Fuji Electric company to Japan. This algorithm was already used to estimate lifetime duration for runner blade operating mechanism to an still in operating turbine from Romania;
- the measured fatigue cycles taking in consideration, by 9 cycle/hour, were assigned for normal operating regime cumulated with the power regulation regime (secondary regulation regime) for which the calculus are made;
- the estimated lifetime duration is by 5,89 years for the third operating regime and by 16,83 years for regime I and II. Is very important to say that this lifetime duration was resulted taking in consideration the fatigue diagram for rotating bending stress (for lever pin) used by Fuji Electric com-

pany. For more precise calculus is needed to determinate the lever material fatigue diagram;

The analyzed operating conditions reveal the fact that, the internal calculus method; impose a special attention for fatigue effects over a Kaplan turbine. Even if the results are obtained from a theoretical analyze, it is clearly revealed the importance for introduction of such calculus in routine strength calculus.

Acknowledgement

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