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Analytical Method to Estimate Fatigue Life Time Duration in Service for Runner Blade Mechanism of Kaplan Turbines

The paper present an analytical method that can be used to determianted fatigue life time duration in service for runner blade mechanism of Kaplan turbines. The study was made for lever button of runer blade mechanism using two analytical relation to calculate the maximum number of stress cycles whereupon the mechanism work without any damage. To estimate fatigue life time duration will be used a formula obtained from one of most comon cumulative damage methodology taking in consideration the real exploatation conditions of a specified Kapaln turbine.

***Keywords**: fatigue, runner, blade, mechanism*

1. Introduction

In the last four decades in understanding and in designing to prevent fatigue failures in mechanical ensembles have been made great strides. The concern of the design engineer focus on the overall structure as well as its components when exposed to service conditions that result in numerous fluctuating loads and attendant stress and strain histories wich may result in fatigue failure.

Previously, large factors of safety were designed into components because of lack of knowledge and understanding of interactive effects. Teoreticaly these safety factors are not longer needed since the develpment of extensive computer software packages.

The majority of these software packages calculate only the flow of loads through components and the values of stresses to a specified stress concentration. Lately, on the market cud be find some software that claim that are able to encompass large volumes and many channels of real time loading histories and combine the two for fatigue life evaluations. There's efficiency are not proved yet, especially in cases where a very large database about exactly loading histories are not available.

In cases in which the designer did not have access to one of this software for different reasons, it is steel remaining valid the analytical method. The analytical method is even simple, after we have all dates about real loads of the system.

2. Initial consideration

Usually, the real loads of the system are obtained using measured parameters which will be processed using analytical or finite element methods. After we find the real load of the system (the true strength that is developed on the interest zone) using a single equation it is really simple to determinate de maximum number of stress cycles that our structure can endure until appear fatigue failure.

In the last decade, using Wőhller diagram remains the easiest method to find these numbers of cycles. In the same time, can not be denied the fact that only specialized laboratories generate such diagrams using a lot of time, material and finally all process requiring a lot of costs.

Like an alternative, a few research worker have developed, in time many equations in different forms, trying in this way to offer designer's engineering community an analytic way for estimating fatigue life time duration. Lately, two such equations are used currently to solve on analytic way the prediction about fatigue life time duration for any mechanical system.

The two general models about we taking before are the Morrow and Smith-Watson-Topper (SWT) models [1]. The Morrow equation (1) modified the elastic term of the strain life equation for introducing the local mean stress into the strain life equation:

$$\frac{\Delta\varepsilon}{2} = \left(\frac{\sigma'_f - \sigma_m}{E}\right) \cdot \left(2N_f\right)^b + \varepsilon'_f \cdot \left(2N_f\right)^c \tag{1}$$

where: $\Delta\varepsilon$ - total strain range;

σ'_f - true fracture strength (value of σ_a at one reversal);

σ_m - the mean stress;

E – modulus of elasticity;

ε'_f - fatigue ductility coefficient;

N_f - number of cycles to failure; therefore $2N_f$ is equal to the number of reversals to failure;

b - fatigue strength exponent;

c - fatigue ductility exponent.

The SWT model assumes that by fatigue life for any condition of local mean stress depends on the product of $(\sigma_m + \sigma_a) \cdot \varepsilon_a$:

$$(\sigma_a + \sigma_m) \cdot \frac{\Delta\varepsilon}{2} = (\sigma'_f)^2 \cdot (2N_f)^{2b} + \sigma'_f \cdot \varepsilon'_f \cdot (2N_f)^{b+c} \tag{2}$$

Both the Morrow and SWT approaches are in current use by the fatigue community and no consensus exists as which one is superior to other. Overall, the belief of the fatigue community is that the SWT approach provides good results and it is a good choice for general use. However, a review of current literature suggests that the Morrow result provides better results for compressive mean stresses while SWT gives non-conservative results.

To obtain more approach results from real exploitation conditions must calculate the number of cycle for every representative regime that occur in time on service. To select these regimes is necessary to have a good loads history usually obtained from monitoring process that always exists in modern industry.

After we have all the number of cycles the last problem is to apply a cumulative damage methodology for calculating the fatigue life. The cumulative damage methodology choused for this determination are Palmgren-Miner linear aggregation criterion [3].

The formula used to estimate fatigue life time duration is:

$$\Pi = \frac{N_{mc}}{a \cdot N_{AD} + b \cdot N_{O/P} + \kappa \cdot N_e} \tag{3}$$

where: N_{mc} - common denominator of (N_1, N_2,…,N_i): N_1,…,N_i represent the number of cycles for the "i" regimes of works taking in consideration for the analyze;[cycles]

N_{AD} - cycle's number induced by device manager's blades; there are obtained from measurements made in situ;[cycles/year]

$N_{O/P}$ - cycle's number induced by stop and start operates on turbine; there are obtained from measurements made in situ;[cycles/year]

a, b - coefficients result from common denominator operation;

κ - mediate cycles coefficient result from exploitation regimes; it is de pendent on material properties and intensity and different weighting schemes in to the operating overview.

The κ coefficient cud be define like in relation (4):

$$\kappa = c \cdot y + d \cdot z \tag{4}$$

where: c, d - coefficients result from common denominator operation;

y, z – coefficients that express the different weighting schemes in real operating overview (percentage).

The expression for κ is variable function of the real operating overview. Relation (4) describes exactly the operational conditions for the case used to exemplifier the theoretical part.

To resolve equation (1) will be used specialized software capable to obtain real answers from equations with a high degree of difficulty. It is about software named Mathematica a fully integrated software environment for technical and scientific computing. As we already said without such of software actually the problem can not be solved.

The current fatigue design procedures used to design structures and components have evolved primarily from experience based on a gradual application of the new methods and followed by good correlating test results. The underlying principles for all the elements used to process the fatigue calculations, can be summarized by three necessary ingredients for problem solution[1]:

1. Structure or Component Load Histories: the external loads on a component structure flow through the material and cause the cyclic stressing of fatigue critical locations. One needs the number of cycles, directions and magnitudes of all significant external loads. Deciding which loads or magnitudes are significant for analysis may be an iterative process;

2. Geometry: it is necessary to made an analysis for finding the influence of local "hot spot" geometry under stress or strain real value. The influence of local geometry are computed from tabulations of stress concentrations factors, photo elastic experiments, or from finite element analysis results;

3. Material: Cyclic deformation and fatigue properties must be available for the component material of the structure. The cyclic deformation information is used by material models, including the newer multiaxial models to follow the stress-strain behavior at the critical location. The fatigue properties are used to predict the failure (cracking) of the critical location [1].

2. Results and comments

The study system is represented by lever's button from runner blade mechanism of Kaplan turbines from PDF I. Material of laver is 30CrNiMo8 and his mechanical properties are in accordance with current standards.

Having dates about the last two years of turbine's function we select the entry dates. The work regimes taking in considerations for this analyze are:

- $H = 25\ m\ and\ \varphi = 10^{\circ}$;
- $H = 31{,}4\ m\ and\ \varphi = 10^{\circ}$;

H are the head waterfall and φ represent the angular positions of runner blades. In accordance with work history of turbine will be consider that 70% of time the turbine work on first regime and the rest 30% she work to second regime. It is also known the medium number of stress cycles that stress the lever button over one year.

Relation (3) will be adapted for our case taking the next form:

$$\Pi = \frac{N_{mc}}{a \cdot N_{O/P} + \kappa \cdot N_e} \tag{5}$$

The value of measured stress data are [2]:

- $N_{O/P} = 365 \times 0{,}75 = 274\ cycles/year$; for turbine having a duty factor of 0,75;
- $N_e = 86724\ cycles/year$.

The entry dates necessary to apply relation (5) are:

- characteristically dates for regime one necessary to calculate the number of stress cycles N_{f1} (first work regime):
  - $\sigma_a = 262{,}3\ MPa$;
  - $\sigma_m = 229{,}6\ MPa.$
- characteristically dates for regime two necessary to calculate the number of stress cycles N_{f2} (second work regime):
  - $\sigma_a = 312{,}47\ MPa$;
  - $\sigma_m = 273{,}6\ MPa.$
- characteristically dates for stop/start operations necessary to calculate the number of stress cycles N_{f3}:
  - $\sigma_a = 124{,}84\ MPa$;
  - $\sigma_m = 124{,}84\ MPa.$

The real value of stress amplitude and main stress were obtained used measured values of power and force from a Kaplan turbine on specified operating point. To taking in consideration the influence of local geometry under the stress values are used fatigue notch concentration factor analytically defined.

The fatigue notch concentration factor was defined using FKM algorithm [4] for specified geometry and loads (K_f=1,98).

The intermediary dates, obtain after the rolling made with Mathematica software are given in Table 1.

Table 1.

No.	The rolling case	No. of stress cycles N_{fi} [cycles]
1	Regime I ($H = 25\ m\ and\ \varphi = 10^\circ$)	N_{f1}= 1,38 x 10^8
2	Regime II ($H = 31{,}4\ m\ and\ \varphi = 10^\circ$)	N_{f2}= 5,65 x 10^6
3	Stop/start operations	N_{f3}= 5,65 x 10^6

To find the estimated fatigue life duration time, first we must define the value of κ coefficient. The measurers made a period of 2 years reveals the fact that 70% from operating time the turbine function at first regime and for the other 30% we estimate that the turbine work to the highest water fall that can be occurred (one of the most solicitant regime – second regime).

For the defined operating mode the value of κ coefficient will be:

$$\kappa = \frac{1{,}38 \cdot 10^8}{1{,}38 \cdot 10^8} \cdot 0{,}7 + \frac{1{,}38 \cdot 10^8}{5{,}65 \cdot 10^6} \cdot 0{,}3 = 8{,}023$$

where: - $1{,}38 \cdot 10^8$ represent the common denominator between N_{f1}, N_{f2}, N_{f3}.

After we apply formula (5), using dates from Table 1., result an estimated fatigue life duration time by **19,81** years.

For validation the result, in Figure 1 are given the Wőhler diagram for a correspondent material having mechanical proprieties similar with the one of lever's material.

As is shown in Figure 1 the equations used to estimate fatigue life time duration for lever button of the runner blade mechanisms of Kaplan turbines are completely valid and can be used without restriction.

With orange point are marked the zone in which the lever button will break down if the system would work only to firs regime 100% of time (at a stress amplitude by 262,3 MPa).

The Wőhler diagram reveals the fact that to a level stress by 262,3 MPa the lever button can resists only $1{,}8 \times 10^7$ cycles. Knowing the one year totally stress cycles is easier to estimate the fatigue life time duration as:

$$\pi = \frac{1{,}8 \times 10^7}{0{,}86724 \times 10^6} = 20{,}75 \ ani$$

The almost one year difference occurred between life time analytical way obtained and the value result from Wőhler diagram are determinate by the 30% of time in which the system work in other regime.

The relation number (3) can be easier adapted to any kind of work regimes the only restrictive condition being the one that it must be know, as well as it can be, the real exploitation conditions.

The red and green point on the diagram represents the different exploitation regimes without significance for our study. Them represent insignificant points in entire operating picture of system (with low incidence).

It is necessary to specified that the full calculation was made considering that the analyzed structure not having any damage from manufacturing process technology. In other case the calculations are not longer available, the preexisting crack-line imperfection determined a drastically shortening of estimated fatigue life time duration. For that case is necessary to establish other start premises.

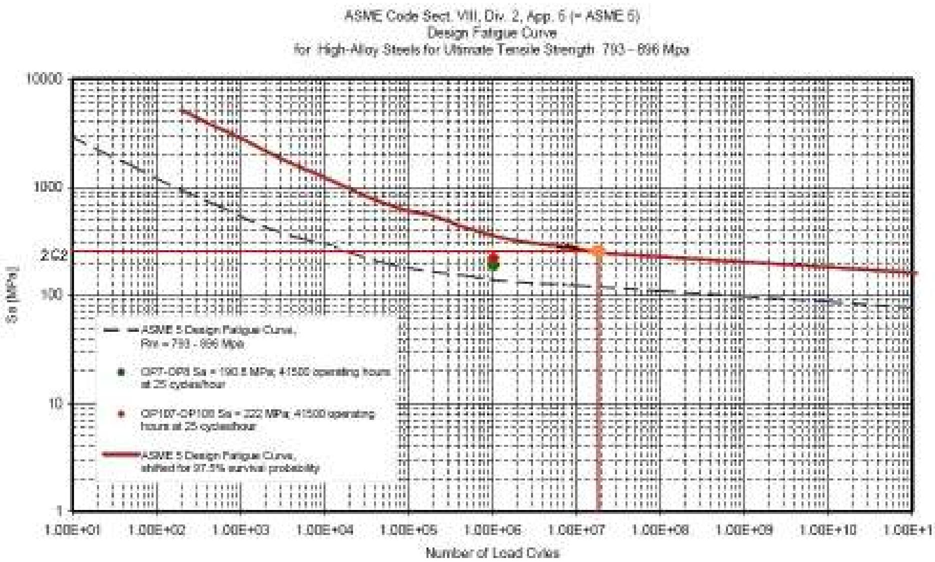

Figure 1. Design Fatigue Curve for High-Aloy Steels [2]

4. Conclusions

The paper show how without any fatigue diagram, a design engineer could estimate the fatigue lifetime duration for a mechanical system. If the real exploitation pictures are known, with minimal effort and without higher costs can be predicted the real life time duration.

The precision of results are strictly determinate by the how better are known the real operating pictures, the existence of o specialized software necessary to resolved the high difficulty equations, and in last but not the least, the experience of design engineers.

The relation (3), obtained from Palmgren-Miner linear aggregation criterion, is the result of personal point of view about the way in which a cumulative damage criterion can be adapted to resolve the problem of estimating fatigue life time duration for a mechanical system, component of Kaplan turbine.

It is necessary to specified that relation (3) was developed to be used only for Kaplan turbines components, being adapted for characteristically exploitation system. For any other mechanical system it is necessary to be developed different relation, having like start point, their specific operation conditions.

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