Fatigue Life Prediction of a Pump Turbine Runner

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Abstract. The working environment of the runner of a hydro-turbine unit is complex, and it is subjected to relatively large and variable water pressure. Fatigue cracks may occur after a long period of work, and even cause the runner to lose its working ability. The fatigue crack problem of hydraulic turbines should be paid attention to. This paper introduces the nominal stress method which is often used to analyse the fatigue life of a hydraulic turbine unit. Modelling and numerical calculation are carried out for a pump turbine unit. Stress concentration points and load spectra are calculated for different working conditions of the turbine. The load spectra are processed by rain flow counting method. The fatigue life of the runner under different working conditions is carried out by Miner criterion. The results show that the fatigue life of the unit tends to be infinite under steady working conditions, and the fatigue life under variable working conditions is far less than that under steady working conditions. Therefore, from the point of view of fatigue life, the working process under variable working conditions should be minimized to improve the service life of the unit.

1. Introduction

Fatigue failure is a problem faced by hydraulic turbines nowadays. The runner works in complex turbulence. Large and variable water pressure will cause damage to the blade, and fatigue cracks may occur after a long period of work. If unexpected cracks occur outside the maintenance cycle, the unit will have to shut down for maintenance, resulting in losses and even safety accidents. Therefore, the fatigue life prediction of units is of great significance to the maintenance cycle design and work safety of hydropower stations.

This paper introduces a common method of calculating fatigue life: nominal stress method, and describes the process of applying nominal stress method to fatigue calculation. The fatigue life of the runner of a Pump Turbine Unit is calculated by the nominal stress method. The results can provide reference for the scheduling and maintenance cycle of the hydropower station[1].

2. Fatigue Life Prediction: Nominal Stress Method

Due to the simplicity and effectiveness of the nominal stress method, the nominal stress method is widely used to predict the fatigue life of hydraulic turbines. Nominal stress method is a classical and commonly used method in fatigue analysis. Based on the S-N curve of material, after obtaining the load spectrum of the structure under test, the damage is calculated, and the fatigue life is calculated by the fatigue damage accumulation theory. The calculation steps are shown as follow[3].
2.1. Obtaining the Load Spectrum
According to the nominal stress method, as long as the stress concentration factor is the same and the load spectrum is the same, the fatigue life of the two structures is the same. In the face of complex practical structures, the stress concentration factor can be used to equivalence it with simple experimental specimens, so as to obtain the maximum stress spectrum corresponding to the dangerous points.

With the development of finite element analysis technology, the stress diagram and stress-time spectrum of complex geometric structure and load-bearing structure can be obtained directly by finite element analysis. It is convenient to find the load spectrum of dangerous point and dangerous point, so as to carry out the next calculation.

2.2. Compilation of Nominal Stress Spectrum
The stress spectrum needs to be transformed into a cyclic load spectrum that can be used to calculate life with S-N curve. Therefore, two treatments are needed: multiaxial fatigue treatment and counting treatment. For multiaxial fatigue problems, the commonly used method is equivalent stress method. Equivalent stress method transforms multiaxial stress into Von Mises equivalent stress, which is described on the basis of deformation energy theory. Von Mises equivalent stress formula is

$$\sigma_{eq} = \frac{1}{\sqrt{2}} \left[ (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right]^\frac{1}{2}$$

(1)

$\sigma_1, \sigma_2, \sigma_3$ are three-dimensional stress and $\sigma_{eq}$ is equivalent stress.

By counting the complex and irregular load spectrum, the load spectrum can be transformed into several regular stress cycles, and the next step of fatigue calculation can be carried out, which is called counting process.

Rainfall counting method is widely used in engineering. It ignores the frequency and time series of loads, discretizes the load spectrum, reorders it into multiple micro-stress cycles that can calculate damage, and then calculates fatigue life. The actual load spectrum cycle can be treated as multiple stress cycles and stress half cycles by rain flow counting. Based on these, the next analysis can be carried out.

2.3. Modification of Nominal Stress Spectrum
The actual life is considered to be related to amplitude stress $\sigma_a$ and mean stress $\sigma_m$. When the influence of load sequence and time history is neglected, the function form of actual life $N_f$ is as follows:

$$N_f = f(\sigma_a, \sigma_m)$$

(2)

The life curve equation described by S-N curve diagram in material test is mostly under the condition of fixed stress ratio. Different groups of data can be obtained by changing the stress amplitude. Its function is as follows:

$$N_f = f(\sigma_a, R = -1)$$

(3)

In practice $\sigma_m \neq 0$, as the effect of stress mean value on material life is weaker than that of stress amplitude, the crack closes more easily under compressive stress and expands more easily under tensile stress, which is called "average stress effect". Therefore, it is necessary to take stress mean value into account in life calculation. When $\sigma_m \neq 0$ is used in practice, the actual stress amplitude needs to be corrected. The stress amplitude is converted to the corresponding amplitude stress when $\sigma_m = 0$ through the correction of the lifeline. The stress amplitude is recorded as $\sigma_{ar}$ after revision.

Classical modified models include Gerber model, Goodman model and Soderberg model. Generally speaking, the Goodman model has the best correction effect and the widest application range. Gerber model and hyperbolic model are also close to the actual situation. Soderberg model is far from the actual situation and less used.

2.4. Calculation of the Damage Degree and Fatigue Life
For the $\sigma_{ar}$ value of each cycle, the corresponding number of cycles $N_f$ is calculated from the S-N curve, and the damage degree of this cycle is defined as
For a micro-cycle with corresponding fatigue life of $N_i$, if $n_i$ times are applied, the total damage degree is as follows.

$$D_i = \frac{1}{N_i} \tag{4}$$

Miner’s rule is based on the following two assumptions:

(1) If the stress amplitude and the mean stress of the two cycles are the same, the damage caused by the two cycles is the same and can be linearly superposed.

(2) The damage of all cycles is linearly superposed, and fatigue fracture occurs when the total damage reaches 1.

Therefore, the condition of fracture is

$$\sum \frac{n_i}{N_i} = 1 \tag{6}$$

The load spectrum of a given time is called load block. In a load block, the damage caused by each stress cycle is respectively

$$d_i = \frac{n_i}{N_i} \tag{7}$$

Then the total damage degree corresponding to this load block is

$$\sum d_i = \sum \frac{n_i}{N_i} \tag{8}$$

Assuming that the material will break after experiencing a load block of lambda, the Miner’s linear rule is satisfied as follows

$$\lambda \sum \frac{n_i}{N_i} = 1 \tag{9}$$

Then the life of the structure under a load block can be calculated, that is to say, the fatigue life of the structure will be designed and calculated after the cycle of a load block[4].

### 3. Modelling the Turbine Runner

The dynamic stress of pump turbine runner is calculated by fluid-solid coupling method. Pump turbine has 16 fixed guide vanes, 16 movable guide vanes, 5 long blades and 5 short blades, as shown in the figure. When a pump turbine is in normal operation, there is water flowing through the runner's outer runner passage (within the clearance). The resulting pressure will affect the stress of the runner, so this part of the influence will be taken into account in the calculation. The runner part and clearance grid are shown as Figure. 1 and Figure. 2.

![Figure 1. prototype pump turbine Drawing](image-url)
The finite element model of the runner is shown in the figure. The connecting position between the upper crown of the runner and the spindle is set as a fixed constraint. The runner is subjected to gravity and centrifugal force. The water pressure calculated by CFD is loaded into the finite element model as a boundary condition.

The material properties of the runner are as follows:

- Density $\rho = 7850 \text{ kg/m}^3$;
- Young's modulus $E = 2.00 \times 10^{11} \text{ Pa}$;
- Poisson's ratio $\mu = 0.3$;
- Ultimate Tensile Strength (UTS) $\sigma_b = 800$ MPa;
- Yield strength (YS) $\sigma_s = 550$ MPa;
- The rotation period of runner under steady condition: $T = 0.12s$;

The S-N curve of the runner material is provided by the manufacturer. The S-N curve equation is as follows:

$$\sigma_a = 1000.6N^{-0.1918}$$  \hspace{1cm} (10)

It is regarded as the fatigue characteristic curve of the blade, and the surface coefficient is no longer modified. The load spectrum counting method is chosen as rain flow counting method, the average stress effect correction method is Goodman model, and the fatigue damage accumulation criterion is Miner's linear rule.
The following is life prediction for six different working conditions of pump turbine[5]: start-up condition, stop condition and 100% load stable condition; start-up and shut-down condition, rated head condition and minimum head condition of pump.

4.2. Fatigue life calculation under different working conditions

4.2.1. Turbine start-up condition. Stress diagram and danger point and load spectrum of start-up process are calculated as follows:

![Stress Spectrum](image)

**Figure 4.** Hot Points and Stress Spectrum of Runners under Start-up Conditions

The stress spectrum is processed by rain-flow counting method. After eliminating the non-peak and non-Valley values, six stress cycles are obtained. The average stress effect of each cycle is modified according to Goodman model, and the damage degree of each cycle is calculated as follows:

**Table 1.** Processing Tables by Rainfall Counting Method in Start-up Process

| Cycle Number | Peak stress /MPa | Valley stress /MPa | Correction stress /MPa | cycle life N/time | Damage Degree D |
|--------------|------------------|--------------------|------------------------|------------------|-----------------|
| 1            | 410.83           | 268.52             | 123.66                 | 5.42 * 10^4      | 1.84 * 10^-5    |
| 2            | 236.54           | 218.56             | 12.56                  | 8.17 * 10^9      | 1.22 * 10^-10   |
| 3            | 255              | 209.64             | 31.96                  | 6.28 * 10^7      | 1.59 * 10^-8    |
| 4            | 274.93           | 206.32             | 49.06                  | 6.72 * 10^6      | 1.49 * 10^-7    |
| 5            | 309.64           | 196.23             | 82.92                  | 4.36 * 10^5      | 2.30 * 10^-6    |
| 6            | 435.57           | 168.78             | 214.36                 | 3.08 * 10^3      | 3.25 * 10^-4    |

Calculating the total damage degree of one start-up process:

\[ D = \sum_{i=1}^{6} D_i = 3.46 * 10^{-4} \quad (11) \]

The total fatigue life of the start-up process is as follows:

\[ \lambda = \frac{1}{D} = 2890 \text{ times} \quad (12) \]

4.2.2. Turbine shutdown condition. Stress diagram and danger point and load spectrum of shutdown process are calculated as follows:
Figure 5. Hot Points and Stress Spectrum of Runners under Shutdown Condition

The rain flow counting method is used to deal with the load spectrum. The method is the same as the above. The damage degree and life obtained are as follows.

Calculating the total damage degree of one start-up process:

\[ D = \sum_{i=1}^{2} D_i = 1.20 \times 10^{-7} \]  \hspace{1cm} (13)

The total fatigue life of the start-up process is as follows:

\[ \lambda = \frac{1}{D} = 8.34 \times 10^6 \text{ time} \]  \hspace{1cm} (14)

4.2.3. Turbine rated head 100% load condition. Stress diagram and danger point and load spectrum of rated head 100% load condition are calculated as follows:

Figure 6. Hot Points and Stress Spectrum of Runners under rated head 100% load condition

The stress spectrum is discretized into 32 cycles by rainflow method, and the total damage degree in each runner cycle (T=0.12s) is calculated as follows.

\[ D = \sum_{i=1}^{32} D_i / 2 = 5.30 \times 10^{-9} \]  \hspace{1cm} (15)

The total fatigue life is

\[ \lambda = 1 / D = 8.34 \times 10^6 \text{ times} \]  \hspace{1cm} (16)

Calculate the hours when the runner can work at 100% load at rated condition:

\[ t = 6300 \text{ h} \]  \hspace{1cm} (17)

4.2.4. Pump Start-up and Shut-down Conditions. Considering the start-up and shutdown conditions of the pump, the total load spectrum of the pump is obtained as follows:
Calculating the total damage degree of one start-up process:
\[ D = \sum_{i=1}^{2} D_i = 3.31 \times 10^{-7} \]  
(18)

The total fatigue life of the start-up process is as follows:
\[ \lambda = \frac{1}{D} = 3.02 \times 10^6 \text{ time} \]  
(19)

4.2.5. **Pump rated head Condition.** Within a runner cycle (T=0.12s), the stress spectrum of dangerous points at rated pump head is as follows:

Calculating the total damage degree of one start-up process:
\[ D = \sum_{i=1}^{21} D_i = 1.16 \times 10^{-12} \]  
(20)

The total fatigue life of the start-up process is as follows:
\[ \lambda = \frac{1}{D} = 8.64 \times 10^{11} \text{ time} \]  
\[ t = 2.9 \times 10^7 \text{ h} \]  
(21) (22)

4.2.6. **Pump Minimum Head Conditions.** Within two runner cycles (T=0.24s), the stress spectrum of dangerous points at the lowest pump head is as follows:
Figure 9. Stress Spectrum of Runner Dangerous Points in Pump Minimum Head Conditions

Calculating the total damage degree of one start-up process:
\[ D = \sum_{i=1}^{30} D_i / 2 = 2.07 \times 10^{-8} \]  (23)
The total fatigue life of the start-up process is as follows:
\[ \lambda = 1/D = 4.84 \times 10^7 \text{ time} \]  (24)
\[ t = 1610 \text{ h} \]  (25)

4.3. Summary
The results of fatigue life calculation under different working conditions are compared in the following table:

**Table 2. Working Life of Turbine Condition**

| Turbine                  | Statistical time/s | Cycle Number | Total Damage Degree D | cycle fatigue life \(\lambda/\text{time}\) | total fatigue life/h |
|--------------------------|--------------------|--------------|-----------------------|---------------------------------------------|----------------------|
| start-up condition       | 250                | 6            | 3.46 \times 10^{-4}   | 2890                                        | -                    |
| rated head 100% load     | 0.12               | 16           | 5.30 \times 10^{-9}   | 1.89 \times 10^8                            | 6300                 |
| Shut-down Condition      | 100                | 2            | 1.20 \times 10^{-7}   | 8.34 \times 10^6                            | -                    |

**Table 3. Working Life of Pump Condition**

| Pump                     | Statistical time/s | Cycle Number | Total Damage Degree D | cycle fatigue life \(\lambda/\text{time}\) | total fatigue life/h |
|--------------------------|--------------------|--------------|-----------------------|---------------------------------------------|----------------------|
| Start-up and Shut-down Conditions | 84                 | 6            | 3.31 \times 10^{-7}   | 3.02 \times 10^6                            | -                    |
| rated head Condition     | 0.12               | 21           | 1.16 \times 10^{-12}  | 8.64 \times 10^{11}                         | 2.9 \times 10^7      |
| Minimum Head Condition   | 0.12               | 15           | 2.07 \times 10^{-8}   | 4.84 \times 10^7                            | 1610                 |

The result shows that the damage degree is the greatest during the start-up of the turbine in the transitional condition, and the life of the lowest head of the pump in the stable condition is the shortest. The life design of other working conditions is far more than 50 years of safe working goal, which can be regarded as infinite life design.

From the point of view of load spectrum, the stress changes most dramatically during the start-up of hydraulic turbine, and the maximum stress amplitude at the dangerous point changes from 435 MPa to
168 MPa. The single cycle damage almost determines the life of the turbine. The lowest working life of pump is determined by the whole stress cycle. From the analysis of material properties, the survival rate of S-N curve of given material is 99.9%, relatively conservative; and the fatigue limit of material is only 45 MPa (corresponding to $10^7$ cycles). Compared with similar steel (CA6NM, about 200 MPa), the fatigue curve of given material is lower, so the fatigue life calculation results are relatively conservative[6,7].

5. Conclusion

In this paper, the nominal stress method is introduced to predict the fatigue life of a pump turbine runner. The results show that the fatigue damage caused by transient processes such as start-up process is much greater than that caused by steady-state process, and the life of transition process is much shorter than that of steady-state process. The lowest head is not the common working condition of the pump. The runner working in the pump mode is damaged greatly, and the life of the other steady-state working conditions are in line with the design expectations. The results of fatigue life prediction of runners can be used for reference in the dispatching of hydropower stations.

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