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Rigid-Flexible Coupling Fatigue Life Reliability Study on Bionic Fish Driving Shaft

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ABSTRACT

Due to the complex loads on a bionic robotic fish operating underwater, the reliability of its working mechanism has an important effect on its overall performance. By establishing a virtual prototype model for the rigid-flexible coupling of a bionic robotic fish, we obtained the instantaneous load on the caudal fin of the robotic fish based on the flapping-wing propulsion theory with MATLAB. A rigid-flexible coupled virtual prototype model for the caudal fin drive as a flexible member of the bionic robotic fish was established, and dynamic simulations were conducted on the model. The simulations revealed the weak links in the drive shaft and established a damage level indicator and fatigue reliability analysis method based on damage theory. The behavior of fatigue reliability for different stress cycles was established, and a dynamic reliability design method with great engineering application value was proposed for virtual prototypes of rigid-flexible coupled underwater bionic robots by combining the virtual prototype technology of rigid-flexible coupling with the theory of flapping wing propulsion and the theory of random load fatigue reliability.

Keywords: Bionic robotic fish; Rigid-flexible coupling; Fatigue life; Reliability

1 Introduction

Bionic robotic fish play an important role in marine development and scientific investigation. Bionic killer whales feature the characteristics of nimble movement flexibility, high transmission efficiency, and low noise. ^[1] The propulsion mechanism of the bionic killer whale—that is, the tail fin—consists of components that are susceptible to failure due to complex and variable impact loads. A recent study on the virtual prototyping of bionic robotic fish includes the design of the tail fin propulsion mechanism with two degrees of freedom and the mechanical structure of the pectoral fin with a single degree of freedom based on the bionic study of the swimming mechanism of Caranxidae model fish conducted by Zhu. ^[2] This study also presents the hydrodynamic analysis of tail and pectoral fin propulsion. ^[2] Using virtual prototype technology, Fan ^[3] conducted dynamic performance analysis of a multijointed robotic fish driven by a synchronous belt tandem epicyclic gear and obtained dynamic data of the torque and power of the drive shaft. These data served as important references in the model selection, mechanical structural optimization, and motion control of the mechanical fish. Lin ^[4] built a virtual prototype model for the swing mechanism of a three-node robotic crocodile tail with ADAMS and analyzed the relationship curves between the displacement, speed, and time for each section of the tail as well as the relationship curves between the driving speed and the speed of each section of the tail. Zhang ^[5] used ADAMS to conduct a dynamic simulation analysis of Caranxidae robotic fish. By driving the robotic fish in different modes with tandem rudders, Zhang obtained the bulk wave amplitude envelope curves of the robotic fish in each swim mode and identified the most practical drive mode of the robotic fish.

While operating underwater, the bionic robotic fish experiences nonlinear, time-varying, and strongly coupled loads. The analysis methods in previous studies cannot reliably express information on the mechanisms of robotic bionic fish. By combining rigid-flexible coupling virtual prototyping technology and the theories of flapping-wing propulsion and random loading fatigue reliability, one can successfully solve the analysis problem of fatigue life reliability for key components of bionic robotic fish and the fatigue reliability behavior of key components under random loads. This analysis has provided definitive data support and new approaches for the design and development of underwater biomimetic robots and has significant engineering application value.

2 Theoretical Background

2.1. Analysis of the tail fin propulsion motion mechanism of the bionic killer whale

The power source of the bionic killer whale is the propulsion generated by the flapping of wings. Before modeling the instantaneous load, one needs to analyze the forces and establish a mathematical model for the propulsion force. The tail fin of the bionic killer whale may be defined as a flapping wing with a single degree of freedom. ^[6-8] Figure 1 shows the analysis of the forces of the flapping wing.

Fig 1. Force analysis of flapping wing movement

Figure 1 shows a flapping wing movement mechanism, and Figure 2 shows the angle θ of the vibration, calculated as

$$\theta = \phi_t + A \sin(\omega t + \varphi_i) \quad (1)$$

where ϕ_t is the offset, A is the amplitude of the flapping wing, ω is the frequency, and φ_i is the phase angle.

Here, V_{he} is the combined velocity of the linear velocity V_w of the flapping wing and the intake velocity V_0 of the incident flow, where the linear velocity V_w is always perpendicular to the direction of motion of the flapping wing, and the lift is perpendicular to the combined velocity V_{he} as the wing flaps up and down. The lift is calculated as

$$F_{sl} = \frac{1}{2} \rho_{yt} A_{wq} C_{sl} V_{he}^2 \quad (2)$$

where ρ_{yt} is the density of the liquid, A_{wq} is the projected area of the flapping wing on the XOY coordinate plane, C_{sl} is the lift coefficient, and R_e is the Reynolds number.

The additional mass force F_{fj} can be calculated as follows:

$$F_{fj} = \frac{1}{4} \pi c^2 b \left(V_{he} \frac{d\theta}{dt} + \frac{c}{2} \frac{d^2\theta}{dt^2} \right) \quad (3)$$

where c and b are the chord length and the extension of the flapping wing, respectively. In Figure 1, the total thrust F_{zs} generated by the flapping wing is the sum of the lift force F_{sl} and the additional mass force F_{fj} .

The direction of the resistive force F_{zl} as the wing flaps up and down is the same as the direction of the combined velocity, calculated as

$$F_{zl} = \frac{1}{2} \rho_{yt} A_{wq} C_{zl} V_{he}^2 \quad (4)$$

where C_{zl} is the drag coefficient.

2.2 Fatigue reliability analysis theory

In designing the fatigue reliability of components subjected to random load, the stress–time history may be converted into frequency, and then the fatigue reliability behavior of the component can be obtained by combining the frequency with the S-N curve of the material and using the Miner linear damage theory.

When a component experiences a random stress, the miner linear damage theory states that every loading cycle consumes a certain portion of the effective life of the component. The total amount of damage when the component reaches failure is a constant. When the amount of accumulated damage reaches 1, the component fails in fatigue^[9–10] according to the following mathematical expression:

$$D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \dots + \frac{n_n}{N_n} = \sum_{i=1}^n \frac{n_i}{N_i} \quad (5)$$

$$N_i = \left(\frac{\sigma_i}{10^{a_p}} \right)^{\frac{1}{b_p}} \quad (6)$$

Here, D is the total cumulative amount of damage. When $D < 1$, the component fails in fatigue. When $D = 1$, the component fails in fatigue. Additionally, n_i is the number of cycles for the component to have worked at the stress level σ_i (n_i may be obtained from the statistical data of the random load); N_i is the number of cycles to failure on the S-N curve of the material at a stress level of σ_i ; and a_p and b_p are parameters of the S-N curve of the material.

3 Dynamic reliability analysis of the tail fin propulsion mechanism of bionic killer whales

3.1 A Establishment of mode neutral files for key components

The main modes of motion of the bionic killer whale include diving, surfacing, swimming straight, and turning. Diving and surfacing are mainly realized by the rotation of the pectoral fin. The swimming straight movement of the bionic killer whale is realized by the up and down sway of the tail fin, and the turning movement is realized by the left and right swing of the tail fin. In this paper, the bionic killer whale developed by a company is used as the project background, and the three-dimensional model of the bionic killer whale established by CREO is shown in Figure 2.

Fig 2. Bionic killer whales's 3d model

A modal neutral file for the drive shaft was established with finite element software ANSYS by meshing up the drive shaft with 185 Solid8node elements at level 6 precision to obtain 17606 nodes and 79410 elements. A finite element model of the drive shaft and the necessary external connecting points needed for simulation were established by setting up real constants and attributing properties to the material. A model neutral file needed for simulation,^[11] as shown in Figure 3, was established using ADAMS.

Fig 3. The modal neutral file of the transmission shaft

3.2 Dynamic reliability analysis of the tail fin propulsion mechanism

The forward and turning motions of the bionic killer whale are achieved by the up and down and left and right movements of the tail fin, respectively. An analysis of the propulsion mechanism of the tail fin can be conducted according to the propulsion mechanism of a flapping wing. The up and down sway of the tail fin of the bionic killer whale may be simplified to motion within the XOY plane, and the left and right swing of the tail fin may be simplified to motion within the XOZ plane.^[6] These motions are shown in Figure 4 and Figure 5.

Fig 4. Mechanism analysis of straight movement of bionic killer whales

Fig 5. Mechanism analysis of bionic orcas turning movement

The instantaneous triaxial force and moment curves for forward and turning motions of the bionic killer whale, shown in Figures 6 and 7, respectively, were obtained from the MATLAB numerical simulation of the instantaneous loads in the swimming straight and turning motions based on the analysis of the swimming straight and turning motion mechanisms of the bionic killer whale and the actual model developed by a company in China.

(a) Instantaneous three-way force curve of bionic orcas in straight motion

(b) Instantaneous three-way moment curve of bionic orca in straight motion

Fig 6. Instantaneous three-way force and three-way moment curves of bionic orcas in straight motion

(a) Instantaneous three-way force curve of bionic orca turning movement

(b) Instantaneous three-way moment curve of bionic orca turning movement

Fig 7. Curve of instantaneous three-way force and three-way moment of bionic orca turning movement

The mode neutral file (mnf) of the ANSYS-generated rocker arm chassis was introduced into ADAMS to replace the established rigid model. Due to the complexity and flexibility of the chassis component structure, we first optimized the produced mnf document with the ADAMS-Flex module to save virtual prototype simulation time in generating the optimized mnf document. In the process of replacing the rigid body with the flexible body, external connection points were used to reconstrain the components.^[11] The virtual prototype model of the rigid-flexible coupled bionic killer whale constructed is shown in Figure 8.

Fig 8. The Rigid-flexible coupled vp model of the bionic killer whale

According to the simulation step length and time setting of the generated instantaneous load text file, the simulation step length of the rigid-flexible coupling virtual prototype of the bionic killer whale was set to 0.01, and the simulation duration was set to 10 s. To ensure the simulation efficiency, the simulation integrator of the rigid-flexible coupling model that contained flexible parts and the integration format were set to WSTIFF and SI2, respectively. The stress, strain, and deformation results of the driving shaft, shown in Figures 9–13, were obtained from the postprocessing module after the simulation had succeeded.

Fig 9. Equivalent stress nephogram of the transmission shaft

Fig 10. The maximum equivalent stress force curve of the transmission shaft

Fig 11. The maximum equivalent stress force curve of the transmission shaft

Fig 12. Equivalent strain nephogram of the transmission shaft

Fig 13. The maximum equivalent strain curve of the transmission shaft

The results in Figures 9–13 show that the force experienced by the drive mechanism of the tail fin of the bionic killer whale had periodic variation, with an average value of 70.959 MPa and a maximum stress at node N1220 located at the flat keyway. The value and time of the maximum stress were 135.100 MPa and 0.16 s. The maximum stress was lower than the allowable yield strength of 205 MPa, and the safety factor of the maximum stress was 1.517. The maximum strain occurred at node N1220 located at the flat keyway, and the average value was 0.000196 mm. The magnitude and time of the maximum strain were 0.000418 mm and 0.16 s, respectively.

4 Fatigue life reliability analysis of the tail fin propulsion mechanism of bionic killer whales

Due to the asymmetric nature of the load experienced by the drive shaft during operation, random instability exists. Its reliability may be solved using the fatigue life analysis method for randomly varying stresses. Statistical analysis may be performed on the stress spectrum of the drive shaft using the rain flow counting method to obtain the stress amplitude and the stress cycle at the different stages. The material of the drive shaft of the material is 304 stainless steel. A search of the literature^[12-13] yielded the S-N fatigue curve for 304 stainless steel, as shown in Figure 14.

Fig 14. S-N fatigue curve of 304 stainless steel

When a component is subjected to random loads, its fatigue life may be computed using the miner fatigue accumulated damage theory. According to this theory, a component will experience fatigue failure when its sum of fatigue damage components reaches 1. Stress analysis of the drive shaft^[14-16] shows that within the first second of time, there are mainly five cyclic stress components in action: $[\sigma_1, \sigma_2 \cdots \sigma_5]$, as shown in Figure 15.

Fig 15. Random variable stresses at all levels

Let $[N_1, N_2 \cdots N_5]$ be the number of cycles of each stress component and N_z be the total number of cycles, where $N_z = N_1 + N_2 + N_3 + N_4 + N_5$. The amount of damage caused by each stress component is $[D_1, D_2 \cdots D_5]$, and the expression for the amount of damage is $d_i = \frac{n_i}{N_{ei}}$, where N_{ei} represents the fatigue life of the part corresponding to each stress component and can be obtained from the S-N fatigue curve in Figure 15.

The total cumulative damage D is:

$$D = \frac{n_1}{N_{e1}} + \frac{n_2}{N_{e2}} + \frac{n_3}{N_{e3}} + \frac{n_4}{N_{e4}} + \frac{n_5}{N_{e5}} \quad (7)$$

From the Miner theory of cumulative damage, when the value of the cumulative damage is greater than 1, the part is considered to have failed. The fatigue life reliability equation of the drive shaft is defined as $z(t) = 1 - D(t)$. For any given time, the fatigue life reliability of the drive shaft may be obtained. In the fatigue life reliability analysis, a log-normal distribution can successfully represent the distribution of stress, strength, and fatigue life. Assuming that the fatigue life of the drive shaft obeys a log-normal distribution, the fatigue life reliability of the given life of the drive shaft can be obtained^[17-18], as shown in Figure 16.

Fig 16. Fatigue life reliability of transmission shaft

Based on the state of motion of the bionic killer whale, we obtained the total number of cycles of the various stresses on the drive shaft in one hour to be $5 \times 60 \times 60 = 1.8 \times 10^4$. From Figure 16, when the fatigue life of the drive shaft was less than 3.132×10^5 cycles (or 17.4 h), the reliability was greater than 0.995%. When the fatigue life was greater than 5×10^5 cycles (or 27.5 h), the reliability showed a substantial drop. When the fatigue life reached 8.573×10^5 cycles (or 47.63 h), the reliability basically dropped to zero. We therefore define that when the reliability is $R \geq 0.95$, the drive shaft is operating reliably, and the fatigue life is 4.329×10^5 cycles, or 24.05 h.

5 Results

We performed a dynamic reliability simulation of the drive shaft of a bionic killer whale using virtual prototype technology with rigid–flexible coupling. We analyzed the fatigue reliability of the drive shaft based on random load fatigue reliability theory and arrived at the following conclusions:

(1) During the operation of the bionic killer whale, the load is complex and variable. The instantaneous triaxial force and triaxial moment can be obtained based on flapping wing propulsion theory. Through rigid–flexible coupled virtual prototype simulation, we obtained the position, mean value, maximum, and time of the stress and strain on the drive shaft and provided data support for the fatigue life reliability analysis.

(2) We statistically analyzed the stress on the drive shaft using the rain-flow counting method and analyzed the damage of the drive shaft based on the S–N fatigue curve of the shaft material and the miner fatigue cumulative damage theory. Based on the random loading fatigue reliability theory, we constructed an algorithm for computing fatigue reliability based on cumulative damage. We obtained the law of fatigue reliability for different numbers of stress cycles and provided the design and development of a bionic killer whale with quantitative data support and a theoretical basis.

(3) In this work, we combined the technology of rigid–flexible coupled virtual prototyping and the theory of fatigue reliability under random loading and proposed a design method for the dynamic reliability of rigid–flexible coupled virtual prototypes for bionic robots in underwater applications. This research provides definitive data support and new methods for the design and development of underwater bionic robots and possesses significant engineering application value.

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