Simulation of shock test for an AUV propulsion motor based on DDAM

Lei Liu¹,*, Zengwu Zhao², Juan Wei³,⁴, Xiaobei Li⁴, Liwei Yuan⁵, ⁶

¹Beijing Institute of Precision Mechatronics and Controls, Beijing, China
²Beijing Institute of Precision Mechatronics and Controls, Beijing, China
³Beijing Institute of Precision Mechatronics and Controls, Beijing, China
⁴Beijing Institute of Precision Mechatronics and Controls, Beijing, China
⁵Beijing Institute of Precision Mechatronics and Controls, Beijing, China

*hlldsyl@126.com
b247474954@qq.com
cweijuanpanda@126.com
dhitzll@163.com
e13439310023@163.com

Abstract—The paper studies the simulation of shock test for a propulsion motor equipped on an AUV. According to DDAM, the modal parameters are calculated using finite element analysis. Then the input spectrum of impacting acceleration is given based on modal analysis. The response after shocking is analyzed at last. Distribution of the stress in different directions are given, showing that the shock resistance performance of the propulsion motor satisfied the shock test.

1. INTRODUCTION
Countries all over the world have listed ocean exploration as one of state’s development projects. A lot of efforts have been made to research underwater vehicles, among which autonomous underwater vehicle (AUV) is a hot topic. AUV is equipped with power-supply system and can accomplish multiple tasks like navigation, communication and development of marine resource. Due to its advantage in autonomous control, AUV has been widely used in different areas [1]. However, the environment that AUV needs to face is complex during its voyage under water. There are dangerous geography circumstances like hills and ravines that AUV would run into. Additionally, AUV would suffer from non-contact shock like the blast wave caused by explosion, which would cause serious damage to the body of AUV. Thus the capacity of AUV to resist shock is of vital importance. Some relevant standards stipulate that shock test needs to be done to validate the shock resistance performance of underwater devices, so that AUV can continue working even shocked by external impact.

Experiment for shock test is quite expensive and difficult to manipulate. As a more economic and efficient method, numerical simulation is becoming an important way to evaluate the shock resistance performance of machine [2]. During the phase of initial design, technique staff can find out the weak spot of the structure by simulation as a base for further optimum design. This paper studies the
simulation of the shock resistance performance of an AUV propulsion motor based on dynamic design analysis method (DDAM). The modal parameters are first calculated using finite element analysis. Then the input spectrum of impacting acceleration is given based on modal analysis. The response after shocking is analyzed at last, illustrating the distribution of the stress in different directions.

2. DYNAMIC DESIGN ANALYSIS METHOD

DDAM is a way to assess shock response by modal superposition. Based on dynamic theory, it simplifies the object as a multiple-freedom system and combines each individual modal to calculate the maximum linear response after stimulation. Usually, the stimulation is uncertain in actual application. It is often defined in the form of spectrum between frequencies and acceleration, taking modal parameters and impacting orientation into consideration. Thus DDAM can overcome the disadvantage of static analysis that the response of higher frequencies is neglected [3].

The differential equation of motion for a multiple-freedom system excited by acceleration $\ddot{a}(t)$ is

$$\mathbf{M}\ddot{x} + \mathbf{K}x = -\mathbf{M}\ddot{\omega}_n^2\mathbf{1} \ddot{a} \quad (1)$$

In which $\mathbf{M}$ is the mass matrix of the system, $\mathbf{K}$ is the stiffness matrix of the system, $x$ is the displacement of each single freedom.

After the regularization transformation, the equation can be decoupled as following,

$$q_n = -\frac{P_n}{\omega_n} \int_0^\infty \ddot{a}(\tau) \sin \omega_n (t-\tau) d\tau \quad (2)$$

In which $P_n$ is the modal participation factor, indicating the level of importance of modal to the system. $\omega_n$ is the modal frequency of $n$th modal, which is the key point in the spectrum of impacting.

The spectrum of velocity defined by convolution is

$$V_n = \int_0^\infty \ddot{a}(t) \sin \omega_n (t-\tau) d\tau_{\text{max}} \quad (3)$$

Then $q_{n\text{max}}$, which is the modal coefficient, is

$$q_{n\text{max}} = P_n V_n / \omega_n \quad (4)$$

And $x_n$, the maximum displacement corresponding to the $n$th modal, can be expressed as

$$\{x_n\} = q_{n\text{max}} \cdot \{\varphi_n\} \quad (5)$$

where $\{\varphi_n\}$ is the transformation matrix.

There are several methods to finish the modal superposition, such as complete quadratic combination, square root of the sum of the square and so on. NRL combination is recommended in DDAM to evaluate the peak value of the response. According to the NRL combination method, the displacement of a single freedom is

$$x_{\text{NRL}} = \sqrt{\sum_n x_n^2 - x_b^2 + x_b} \quad (6)$$

where $x_b = \max(x_n)$, $n = 1, 2, 3…$

To evaluate the stress of the structure after shocking, Von Mises Failure Theory is widely used. Effective stress can be expressed as

$$\sigma_{\text{shock}} = |\sigma_{n\text{max}}| + \sqrt{\sum_n \sigma_n^2 - \sigma_{n\text{max}}^2} \quad (7)$$

In which $\sigma_n$ is the largest stress value of the $n$th modal, $\sigma_{n\text{max}}$ is the largest stress value of all modal and $\sigma_{\text{shock}}$ is the effective stress.
Compare $\sigma_{\text{shock}}$ with the mechanical performance of the materials used on the body of AUV. The dynamic property of the structure satisfies the demand for shock test when

$$\sigma_{\text{shock}} < \sigma_b$$

where $\sigma_b$ is the yield limit of material.

3. SIMULATION OF SHOCK TEST

In order to simulate the shock test on the propulsion motor of AUV, the modal parameters of the system need to be specified through finite element analysis firstly. Then the input spectrum between impacting acceleration and some crucial frequencies needs to be determined, according to the result of modal analysis. On the base of the former two steps, the response of the impacting acceleration can be calculated according to DDAM.

3.1 Modal Analysis

Modal is the intrinsic dynamic property of structure that determines how the structure deforms. Modals in different order have its corresponding frequency. When the external force on the structure has the same frequency as the modal frequency, resonance that causes large deformation would happen [4]. By modal analysis, properties of modals in different order can be obtained to predict the structure’s actual response after the stimulation by the external force. Thus modal frequencies are important parameters in dynamic design and usually included in the input spectrum of impacting acceleration.

The differential equation of motion for a system can be expressed as

$$M\ddot{x} + C\dot{x} + Kx = F$$

(9)

Where $C$ is the damping matrix of the system, $F$ is the external force on the system [5]. Usually, modal analysis is based on undamped free vibration, thus (9) can be simplified as

$$M\ddot{x} + Kx = 0$$

(10)

Solution of the above equation is

$$x = b \sin(\omega t)$$

(11)

Where $b$ is the amplitude of vibration displacement. The characteristic equation of (10) is

$$\det(M - \omega^2K) = 0$$

(12)

Solve the above equation and $\omega$ is the modal frequency of the system.

FEA is an efficient and effective method to obtain the modal parameters of the system. The finite element model of the propulsion motor is illustrates in Fig. 1. Some details like chamfer and rounded corners have very small influence on the dynamic property of the structure, so that these details are usually neglected to improve the efficiency of the calculation. The eight bolt holes on the flange of the propulsion motor are fixed as boundary condition. Connections between different parts are defined according to the assembly relationship between them.
Figure 1. FEA model of the propulsion motor

Table 1 illustrates the necessary mechanical properties of the material used on the propulsion motor.

| material          | density (kg/m$^3$) | poisson ratio | elastic modulus (N/m) |
|-------------------|--------------------|---------------|-----------------------|
| structural steel  | 7850               | 0.30          | 2x10$^{11}$           |
| alloy aluminum    | 2630               | 0.33          | 7x10$^{10}$           |
| copper            | 5810               | 0.34          | 1.19x10$^{11}$        |
| titanium alloy    | 4500               | 0.34          | 1.13x10$^{11}$        |

Underwater devices should be able to bear impact from different directions, thus the modal parameters in three directions are all calculated. Table 2 shows the result of modal analysis in three directions. Modals of which the mass is up to 10% of the total mass are all included. These frequencies make up the key points for the input spectrum of the impacting acceleration.

| direction     | frequency/Hz | ratio/% |
|---------------|--------------|---------|
| vertical      | 109.15       | 11      |
|               | 148.74       | 14      |
|               | 150.11       | 16      |
| broad-wise    | 109.15       | 11      |
|               | 148.74       | 17      |
|               | 150.11       | 13      |
| end-wise      | 550.8        | 96      |
3.2 The Input Spectrum of Impacting Acceleration

According to the national standard GJB 1060.1-91 General Requirement for Environment Conditions of Naval Ships Mechanical Environment, the impacting acceleration is related to a reference value. For underwater devices, the reference acceleration $A_0$ and the reference velocity $V_0$ can be obtained by the following equations

$$A_0 = 98.1 \times \frac{19.05 + m_a}{2.72 + m_a}$$

(13)

$$V_0 = 1.52 \times \frac{5.44 + m_a}{2.72 + m_a}$$

(14)

where $A_0$ is the reference acceleration, $V_0$ is the reference velocity, $m_a$ is the modal mass which approximates 80% of the total weight. Then the design value $A_a$ and $V_a$ can be calculated as Table 3 [7].

| direction      | equation                      | $A_a/(m/s^2)$ | $V_a/(m/s)$ |
|----------------|-------------------------------|---------------|-------------|
| vertical       | $A_a=A_0$, $V_a=0.5V_0$       | 652.3         | 1.5         |
| broad-wise     | $A_a=0.4A_0$, $V_a=0.2V_0$    | 260.9         | 0.6         |
| end-wise       | $A_a=0.2A_0$, $V_a=0.2V_0$    | 130.5         | 0.6         |

According to the national standard, the amplitude of the impacting acceleration for the system should be the smaller one between $A_a$ and $V_0\omega_0$, where $\omega_0$ is the natural circular frequency [8]. Finally, the input spectrum between the impacting acceleration and frequencies for the simulation of shock test is defined in Table 4.

| impacting direction | frequency/Hz | acceleration/(m/s^2) |
|---------------------|--------------|----------------------|
| vertical            | 109.15       | 652.3                |
|                     | 148.74       | 652.3                |
|                     | 150.11       | 652.3                |
| broad-wise          | 109.15       | 260.9                |
|                     | 148.74       | 260.9                |
|                     | 150.11       | 260.9                |
| end-wise            | 550.8        | 130.5                |

3.3 Simulation of shock test

Input the spectrum between the impacting acceleration and frequencies derived from Table IV into the dynamic model of the propulsion motor to calculate the response and identify the dangerous zone in different impacting directions after shocking [9]. The distribution of the equivalent stress is illustrated in Fig. 2.
As can be seen in Fig. 2, stress on the shaft is the largest in the vertical and broad-wise direction, of which the corresponding amplitude are 313.5MPa and 124.7MPa. The largest stress happens at the spot where the bearing condition for the shaft changes, which would cause the concentration of stress. Stress near the bolt hole is the largest in the end-wise direction and the corresponding stress is 23.9MPa. That’s because of the smaller thickness of the flange and the intense appearance change near the bolt hole.

The largest stress in all three directions after shocking is smaller than the yield limit of the material. According to the national standard, the shock resistance performance satisfies the shock test [10]. However, the stress on the vertical direction is relatively larger than the other two directions. If shocked for enough times, failure would happen on the dangerous zone, which would affect the safety and reliability of the AUV.
4. CONCLUSION
The paper studies the numerical simulation for shock test on a propulsion motor equipped on an AUV using DDAM. Details about the distribution of equivalent stress in different impacting direction is illustrated, showing that the mechanical property of the structure is good enough to bear the shock. However, the stress caused by the vertical impacting on the shaft is larger than the other two directions, due to the sudden change of the bearing condition. Further optimal design of the bearing configuration can be considered to achieve a more reasonable structure.

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