The experimental modal analysis of water-lubricated batten

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Abstract. With the widespread use of water lubricated stern bearing, researchers around non-metallic materials are becoming hot points in recent years and researches on dynamic characteristic especially modal analysis are particularly important. The dynamic characteristics of water lubricated batten depend mainly on its modality parameters. This paper presented experimental modal analysis of water lubricated batten, four material, they are PTFE, ACM, dragon, cylon material, by hammering method, analysed the batten based on linear modal analysis theory. Due to the nonlinearity of batten material, although the natural frequency range of water lubricated batten layer could be gotten, it’s hard to get the precise modal shapes of it.

1. Introduction

Water-lubricated rubber stern bearing is an important part of the ship. The commonly used material is rubber. The rubber has excellent vibration damping performance, which can effectively reduce the stern shaft vibration, reduce the bearing noise radiation, improve the working environment of the stern shaft, reduce the ship's sonar, and improve discover distance and ship survivability.

However, under the special working conditions of poor lubrication such as start-stop, low speed, heavy load and high temperature, the water-lubricated rubber stern bearing is prone to abnormal vibration and sound, which seriously affects the ship's concealment [1-6]. Modal analysis is the main means of dynamic characteristic parameter identification [7-8]. Wu Xiaojin et al [9-10] of Chongqing University analyzed the dynamic characteristics of the stern bearing but only studied the influence of the bushing material on the dynamic characteristics of the bearing. There is no systematic consideration of the change of the rubber bearing structure and the rubber material. Jin Yong et al use finite element modal analysis method on water-lubricated rubber crucible bearings and their linings and bushings, and research the influence law and level of different structural forms and material properties on the dynamic characteristics of water-lubricated rubber crucible bearings[11]. LAO Kunsheng (2017) use the 3D Modeling Software Solidworks to model the water lubricated rubber stern bearing, and the finite element model was established in ANSYS Workbench. Ten modals of the water lubricated rubber stern bearings were analyzed. The conclusion shows that the natural frequency distribution at every stage of the bearings is concentrated and the low modal mainly depends on the inherent characteristics of the lining. In order to obtain the vibration of bearing in a certain frequency range, the harmonic response analysis was carried out knowing the frequency and stress amplitude of resonance of the bearings, which provides a theoretical basis for the further design optimization and the use of the water lubricated rubber stern bearings[12]. In this paper, the test
modal analysis of water-lubricated slats is carried out on the LMS platform by hammering method, and the test results were analyzed.

2. Experimental modal analysis

Modal analysis is a process of identifying the dynamic characteristics (natural frequency, mode shape, damping, stiffness, mass, etc.) of a mechanical system and determining its dynamic response characteristics. It is the basic step of other dynamic analysis such as harmonic response analysis and spectral analysis. Modal analysis can be divided into theoretical modal analysis and experimental modal analysis according to different methods. The current theoretical modal analysis is mainly solved by finite element technology, such as ANSYS, ABAQUS, MARC, PATRAN, DYTRAN and so on. The experimental modal analysis is further divided into frequency domain identification and time domain identification. The frequency domain identification method is used in this paper. The advantage of experimental modal analysis is that the dynamic information of the structure can be accurately obtained, especially the natural frequency and damping of the low-order mode, but the information of the vibration mode (especially the rotational mode) is not accurate enough, and the analysis ability for higher-order modes is limited. By combining the finite element modal analysis method and the experimental modal analysis technique, the modal analysis is a very effective structural dynamic analysis method. First, the finite element modal analysis is used to perform preliminary calculations and view the structure. The natural frequency and mode shape provide guidance for the test of the experimental modal analysis; then the experimental modal analysis results are used to test, supplement and correct the original finite element dynamic model. Finally, the modified finite element model is used to calculate the dynamic characteristics and response of the structure, and the optimal design of the structure is carried out.

3. Modal analysis of water lubricated batten

3.1 Experimental modal analysis object

The test object is 4 non-metal water lubricated batten, and the battens are divided into 18 measuring points. Equally divided by 5 in the length direction and 2 equal in the width direction, as shown in Figure.1.

![Figure 1. Four material water lubricated battens](image)

Batten geometry parameters are as follows:
Length 500mm, width 50mm, height 15mm
Batten material parameters, as listed in Table 1.

| Material | Elastic Modulus/GPa | Poisson's ratio | Density/kg·m⁻³ |
|----------|---------------------|-----------------|----------------|
| PTFE     | 0.80                | 0.40            | 2200           |
| ACM      | 3.40                | 0.45            | 1100           |
| Dragon   | 0.30                | 0.48            | 1320           |
| Cylon    | 0.49                | 0.45            | 1200           |

3.2 Test device

The system used in this modal test is the vibration, noise and modal test analysis system of SC310 manufactured by LMS of Belgium. The hammer is Klaster's 9724A (2000) with a sensitivity of 2.2 m V/N and maximum excitation force. Up to 2 000 N, the highest frequency can reach 6 000 Hz. The acceleration
sensor uses Kistler's Model 8791A250 piezoelectric three-way accelerometer with a maximum range of 250 g, sensitivity of 20 mV/g in all three directions, and a maximum frequency response of up to 5000 Hz.

3.3 Selection of support methods

The support method is equivalent to the boundary condition of the tested part. Under different boundary conditions, the dynamic characteristics of the test piece may be completely different. Therefore, in order to test the results accurately and effectively, the boundary conditions of the tested parts must be correctly simulated. According to the similarity principle, in the experimental modal analysis, the most important influence on the dynamic characteristics of the system is the geometric boundary condition, that is, the supporting condition of the tested component.

In this test, the free support is selected as the support method of the water-lubricated slats. In order to achieve the completely free restraint state of the test piece, the support with low support stiffness and low damping is selected as much as possible, so that the elastic mode of the test piece is not greatly affected. If the free support point is selected near the test piece concerned with the modal node, the support effect is better. Therefore, in order to make the bearing as completely free as possible, the test uses a rubber band that is as soft as possible to suspend the bearing on the bracket. Considering the actual test conditions, this test uses the method of winding the cable around the outer ring of the bearing, as shown in Figure 2.

![Figure 2. Experimental modal test system](image)

1-Steel Support; 2- Hammer; 3- Measure Point(Output); 4- Hit Points(Input); 5- PC; 6- LMS Platform; 7- Rubber rope.

3.4 Excitation points, incentives, test points and test ranges

The choice of the incentive mode is not only related to the structural characteristics of the tested component, but also the requirements of the different parameter identification methods for the frequency response function test. According to the characteristics of the tail bearing slat structure and the SIMO parameter identification method, this test uses a single point transient impact excitation. The excitation point should be selected at the position where the slat stiffness is large, and the signal attenuation is small. In the test, 18 points (Hit Points(4)) of the slats are selected as fixed excitation points; point 3 is selected as the signal output point. As shown in Figure 2.

The selection of the test frequency band should take into account the frequency range of the excitation during the actual working process of the slat. Usually, the water-lubricated tail bearing slats are directly excited by the tail shaft, and the excitation frequency is mainly the axial frequency and its multiplication frequency, generally within 100 Hz. Therefore, the highest analysis frequency is not more than 2000 Hz.

4. Results and analysis

4.1 Modal parameter identification

First, in the LMS, the measured frequency response function is applied to the Polymax algorithm (for large damping structure analysis) to extract the poles (corresponding to the system eigenvalues) within the tolerance. The principle of extraction is to extract the frequency as much as possible. Damping, the pole where the vector is stable within the tolerance range and the number of poles must be considered to reflect the true dynamic characteristics of the system. In order to avoid missing valuable poles, the correlation between the integrated frequency response function and the measured frequency response function and the error amount can be calculated for the identified modal
parameters. The identified parameters are accepted only if the correlation is greater than 90% and the error is less than 10%.

4.2 Modal verification

In order to judge the correctness of the experimental modal analysis results, this paper uses the modal confidence criterion (MAC value) for verification. Through the MAC value of the overall modal model of the water-lubricated battens in the free state, it is found that the MAC value of the non-diagonal line exceeds 10%, the four water lubricated battens’ MAC values shown in Table 2-5, but the overall effect is good, which leads to the following possibilities:

| Table 2. MAC values of PTFE |
|-----------------------------|
| AutoMAC          | 111.959 | 310.737 | 437.283 | 614.030 | 884.484 |
| 111.959          | 1.000   | 0.010   | 0.114   | 0.063   | 0.111   |
| 310.737          | 0.010   | 1.000   | 0.231   | 0.209   | 0.255   |
| 437.283          | 0.114   | 0.231   | 1.000   | 0.762   | 0.990   |
| 614.030          | 0.063   | 0.209   | 0.762   | 1.000   | 0.763   |
| 884.484          | 0.111   | 0.255   | 0.990   | 0.763   | 1.000   |

| Table 3. MAC values of ACM |
|-----------------------------|
| AutoMAC          | 111.383 | 309.044 | 612.131 | 996.255 |
| 111.383          | 1.000   | 0.022   | 0.069   | 0.097   |
| 309.044          | 0.022   | 1.000   | 0.175   | 0.155   |
| 612.131          | 0.069   | 0.175   | 1.000   | 0.655   |
| 996.255          | 0.097   | 0.155   | 0.655   | 1.000   |

| Table 4. MAC values of dragon |
|-----------------------------|
| AutoMAC          | 84.057  | 669.595 |
| 84.057           | 1.000   | 0.592   |
| 669.595          | 0.592   | 1.000   |

| Table 5. MAC values of cylon |
|-----------------------------|
| AutoMAC          | 41.559  | 117.702 | 233.881 | 382.791 | 572.172 |
| 41.559           | 1.000   | 0.380   | 0.402   | 0.448   | 0.530   |
| 117.702          | 0.380   | 1.000   | 0.445   | 0.388   | 0.586   |
| 233.881          | 0.402   | 0.445   | 1.000   | 0.632   | 0.786   |
| 382.791          | 0.448   | 0.388   | 0.632   | 1.000   | 0.762   |
| 572.172          | 0.530   | 0.586   | 0.786   | 0.762   | 1.000   |

For a mode with a similar two-order frequency, the MAC value is greater than 10%, which indicates that the test piece may have such a two-order mode with similarities; however, the shape information is missing due to the arrangement of the measuring points, some The deformation of important parts or important directions was not reflected in the test; only the translational freedom of each point in the x, y, and z directions was measured during the test, and the rotational degrees of freedom in these three directions were lost. Therefore, the existing vibration mode information cannot fully reflect the overall appearance of the mode. Subtle frequency shifts during measurement result in two modes when parameter identification occurs.

For modals with large difference in two-order frequency, the MAC value is greater than 10%. It may be due to insufficient number of measuring points or improper positioning. The movement of important parts or important directions in the system may be deformed differently in the unmeasured part, resulting in different deformations. Two similar modalities are produced.

These characteristics have a certain influence on the observational performance of the hammer modal test, resulting in a large deviation of the individual MAC values, but overall, the test modal test results have higher credibility.
4.3 Analysis of modal test results

The mode vibration and frequency diagrams of the four slats are shown in the Figure 3-6. According to the judgment of the modal confidence criterion and the observation of the vibration mode diagram, the first-order mode of the PTFE and the ACM at 111 Hz and the second-order mode of the 309 Hz are more authentic, and the third mode is 612 Hz. The order mode may be due to the force hammer being excited only in the Y direction, resulting in an inconspicuous effect of the bending and torsion coupling mode. The slab 1 mode at 437 Hz and 884 Hz should be a coupled mode, and the mode shape is similar to the 612 Hz mode, so it is removed.

![Figure 3. The modal vibration and frequency diagrams of PTFE](image)
d) Fourth-order mode (996.25537Hz)

Figure 4. The modal vibration and frequency diagrams of ACM

Figure 5. The modal vibration and frequency diagrams of dragon

Figure 6. The modal vibration and frequency diagrams of cylon
The first-order bending mode of the cylon is 41 Hz in accordance with the battens PTFE, ACM and dragon, and the second-order bending mode 117 Hz is identical to the battens of PTFE and ACM, and the third-order bending mode The other 3 blocks of 233 Hz and the fourth-order bending mode of 382 Hz are not excited, and may be related to the position of the suspension. The 572Hz mode is the same as the 669Hz mode of the dragon, and the deformation is already small.

5. Conclusion
In the four types of battens, the magnitude of the deformation is in a range, and the first-order bending deformation modes are basically the same. The battens PTFE and ACM have the highest frequency of first-order bending resonance, indicating that the battens PTFE and ACM have the highest stiffness. In the first-order bending deformation, the deformation of the batten ACM is the largest, and the deformation of the batten dragon is the smallest, indicating that the batten dragon has the highest flexural modulus. In addition, it should be noted that in the first-order bending mode, the damping ratio of the slab cylon reaches 16.9, and the damping ratio of the second-order bending mode 117 Hz also reaches 9.06, which is much better than the damping properties of the other three materials.

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References
[1] Tian Yuzhong, Liu Zhenglin, Jin Yong, et al. Mechanism analy-sis of squeal of water lubrication stern tube rubber bearingbased on experimental study. Journal of Wuhan Universityof Technology.33(2011) 1 : 130 - 133.
[2] Jin Yong, Liu Zhenglin, Tian Yuzhong, et al. Vibration monito-ring of ship stern bearing based on pulse system. Journal of Wuhan University of Technology.32(2010) 6 : 84 -88.
[3] Yao Shiwei, Wang Juan, Wang Jun, et al. Vibration and noise generation theory and experiment of water-lubricated rubber bearings. Ship Science and Technology.12 (2009) 31: 32-35.
[4] Yao Shiwei, Hu Zongcheng, Ma Bin, et al. The new developmentof rubber bearing and its application in warships. Ship Science and Technology.27(2005)( S1) : 27-30.
[5] Zhou Chunliang, Liu Shunlong. Numerical simulation of ship propeller shaft bearing fluid field. Lubrication Engineering.30(2005) 6 : 75-78.
[6] Yu Jiangbo, Wang Jiaxu, Xiao Ke, et al. The effect of elastic modulus to water lubrication bearing. Lubrication Engineering.31(2006) 11): 69-70.
[7] Sinou Jean-Jacques, Jezequel Louis. Mode coupling instability in friction-induced vibrations and its dependency on system parameters including damping. European Journal of Mechan-ics A: Solids.26(2007): 106-122.
[8] Gent A N. Engineerin with rubber: how to design rubber components. USA: Hanser Gardner Publications,1992.
[9] Wu Xiaojin, Wang Jiaxu, Xiao Ke, et al. Dynamic simulation of water lubricated bearing. Journal of System Simulation. 21(2009)(13) : 4167-4170.
[10] Wu Xiaojin, Wang Jiaxu, Xiao Ke, et al. Dynamics simulationof water lubricated bearing . Lubrication Engineering.33(2008)(2): 21-25.
[11] JIN Yong, TIAN Yuzhong, LIU Zhenglin. Analysis of Influencing Factors of Water-lubricated Rubber Stern Bearing Modal.Lubrication Engineering.36(2011) 9 : 10-13.
[12] LAO Kunsheng, JIN Yong, LU Yayun. Modal and harmonic response analysis of water-lubricated rubber stern bearing based on ANSYS.China Shiprepair.,30(2017) 2:37-44.