Experimental Study on the Performance of Ship Pipeline Coated Vibration Isolator

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Abstract. In order to improve the efficiency of ship pipeline vibration isolation, an optimized design scheme of coated rubber vibration isolator was proposed. The natural frequency was calculated and a test bench was established for vibration isolation performance test. The results show that the dynamic and static stiffness of the optimized vibration isolator is significantly lower than the traditional design value at the natural frequency. The natural frequency of the vibration isolator is reduced from about 35Hz to about 9.7Hz from the initial design scheme, and the vibration isolation efficiency is increased from about 20dB to about 33dB as well.

1. Introduction
Ship piping systems have fixed and reliable requirements for supports. The constraints are not only vertical, but also work in the radial direction of the pipeline cross-section 360°, the vibration isolation effect is the same as that of the shock absorber, which causes the piping system to vibrate. Isolate and attenuate the transmission to the hull. In addition, the limited internal space of the ship requires a compact support structure[1-2]. Therefore, the support of modern large-scale ship system pipelines is mainly covered[3]. Because of the compact design structure and high support and fixation reliability, this form is also widely used in various applications[4-5].

On the basis of the existing support structure, full-covering design scheme is adopted here with the main design purpose as vibration reduction and isolation. And as well the requirements of reliable fixation, compact structure, convenient disassembly and assembly, etc were taken into account. The structural composition of the pipeline isolator can be decomposed into elastic elements and support structures. The main function of the elastic elements is to reduce vibration. The function of the support structure is maintaining the shape of the elastic element and its uniform deformation, connecting the pipeline and the basic structure, and realizing the pipe alignment.

2. Design scheme and experiment

2.1. Object pipeline
A verification example of the calculation method of the vibration isolator is carried out on a pipeline with a diameter of 200mm. The outer diameter of the pipeline is 220mm, copper, the wall thickness is 10mm, and the density of the same material is 8900kg/m³. The medium in the pipeline is seawater with a density of 1025kg/m³. The quality of the road plus the quality of the medium in the pipe with current construction process requirements: pipes with a diameter of 76mm or more, separated by 1500mm~1800mm, the total mass is about 200kg.
2.2. Vibration isolator design
Using the pressure-bearing/pull type pipeline vibration isolator as shown in Figure 1, the elastic element structure design principle prototype, nitrile rubber is selected as the vibration isolation material, and the material damping ratio is assumed to be 0.12. For medium hardness rubber, the approximate relationship between the shear modulus and its Shore hardness is \( G = 0.0224 \sqrt{H} \), the relationship between the tensile and compression elastic modulus and the shear modulus. The prototype uses nitrile rubber with a hardness of 55, and its Young's modulus The amount can be set temporarily as \( 6.5 \times 10^6 \, N/m^2 \). The supporting structure is made of Q235 steel plate, which is formed by welding.

![Figure 1 Overall scheme of pipeline vibration isolator](image1)

The prototype of the vibration isolation element has a width of 30mm and a thickness of 40mm, and the support angle of the full-coverage is 90°. Static stiffness of the element is \( 2.12 \sim 2.22 \times 10^6 \, N/m \), the natural frequency of the element under rated load is \( 34.43 \sim 35.22 \, Hz \).

2.3. Experiment method
This test refers to GB/T15168-2013 vibration and shock isolator static and dynamic performance test method for static and dynamic performance test of pipeline vibration isolator. The pipeline isolators are arranged in the form of two vibration isolators evenly supporting the pipeline as Figure 2. Because the weight of the test pipe section is less than the rated load, concentrated masses are evenly arranged on both sides of the two vibration isolators during the dynamic performance test, so that the total mass of the pipeline and the masses is equal to the sum of the rated loads of the two vibration isolators.

![Figure 2 Schematic diagram of test bench](image2)
2.3.1. Static performance test
Carry out the measurement of the static deformation of the rated load and the static stiffness of the rated load, and the calculation of the relationship between the static load and the static deformation. Carry out two slow preloading of the vibration isolator, the load range is from zero to 1.25 times the rated load; after that, two formal loadings are carried out, with a total of 7 static load points, respectively 0.5 times, 0.7 times, 0.9 times, rated, 1.1 times, 1.2 times and 1.25 times. The method of linear regression to the data of each load point is used to calculate the static stiffness and the relationship between static load and deformation.

2.3.2. Dynamic performance test
Perform natural frequency test under rated load, static stiffness calculation and vibration isolation measurement under dynamic excitation conditions. The natural frequency calculation adopts the hammer method to measure the resonance frequency of the bench system.

According to the test method, the static stiffness of the rated load is calculated by the following formula:

\[
K = \frac{1.1P_0 - 0.9P_0}{X_{1.1} - X_{0.9}}
\]

Where \(P_0\) is the rated static load of the vibration isolator, \(X_{1.1}\) is the static deformation value of the vibration isolator at 1.1 times the load, and \(X_{0.9}\) is the static deformation value of the vibration isolator at 0.9 times the load.

Due to the limited conditions of this test, the deformation of the vibration isolator after the load is applied is indirectly measured by the caliper method, and the measurement results will have large errors. In order to reduce the error as much as possible, and at the same time according to the small deformation and linear assumption, the data processing adopts the method of linear difference of the least square method, and all other measurement points except the zero point participate in the calculation. Three measurement points are selected for data collection. The measurement points are on both sides of the upper half ring of the pipeline isolator and the back of the lower half ring (strong support part). In the calculation, the measurement point data with uniform data distribution is selected as the final result.

3. Results and discussion
The static stiffness of the vibration isolation element prototype measured by the static load method is approximately \(2.26 ~ 2.63 \times 10^6 N/m\), and the static stiffness of a single vibration isolation element calculated by the natural frequency of the system is \(\sim 2.36 \times 10^6 N/m^2\). The static stiffness results obtained by the two methods are basically consistent with the results of the theoretical method. The natural frequency of the vibration isolation element obtained by the system test is consistent with the result of the theoretical method, and is closer to the result of using the shape factor calculation formula of the Japanese Society of Mechanical Engineers. The static stiffness and natural frequency test results are close to the theoretical calculation results, which proves the validity of the test method in this paper.

The vibration acceleration level curve obtained by the test is shown in Figure 3. According to the vibration acceleration levels of different measuring points under external excitation, the vibration reduction effect between measuring point at the pipeline and measuring point at the base is calculated, the vibration level drop can reach above 30dB.
Figure 3 vibration isolating performance

4. Conclusion
In view of the compact space of the ship, an optimization plan for the coated pipeline vibration isolator is proposed. By increasing the free area of the rubber body, increasing the shape factor and reducing the stiffness of the vibration isolator, a hollow rubber body vibration isolator structure is designed. A test bench was established to carry out performance tests to verify the design optimization effect. The results show that after the optimized design, the natural frequency is reduced to 27.6% of the initial scheme, and the vibration isolation efficiency is increased by 65%, from 20dB to 33dB.

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