Vibration Characteristics Analysis of Propulsion Shafting of a 48,000 tons Bulk Carrier

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Abstract: The vibration and noise of the marine propulsion shafting is a fairly common problem. As the important composition of the ship propulsion system, marine propulsion shafting is the main noise source of the ship propulsion system. The finite element analysis and harmonic response analysis of the marine propulsion shafting is carried on, and the local natural frequency are calculated. In order to analyze the vibration characteristics of the marine propulsion shafting, the harmonic response analysis was adopted to find out the relationship between the resonant frequency and the related vibration data, which lays the foundation for the reliability design and optimization of the marine propulsion shafting. The study lays the foundation for the reliability design and optimization of marine propulsion shafting.

Keywords: Modal analysis, Propulsion Shafting, Vibration Characteristics Analysis, harmonic analysis, Bulk Carrier.

1. INTRODUCTION

The vibration and noise of the marine propulsion system is a fairly common problem. The marine propulsion system consists of five parts, propeller, stern shaft, intermediate shaft, thrust shaft and main engine. As the transmission systems of the marine propulsion system, the reliability of the marine propulsion shafting affects the safety of the ship. The vibration noise of the propulsion is prone to a variety of failures, which can reduce the reliability of equipment, result marine propulsion system failure, and even ship fault. Therefore, this paper takes marine propulsion shafting the 48,000 tons bulk carrier as an example, carries on the finite element analysis to the marine propulsion shafting, calculates its local natural frequency, and carries on the harmonic response analysis. To find the vibration reason of the system, this paper uses the finite element method to establish the three-dimensional model of the marine propulsion shafting, calculate the natural frequency of the marine propulsion shafting. The vibration and noise analysis of the marine propulsion shafting is the basis of vibration and noise testing and equipment status monitoring, which provide theoretical basis to the optimal design for the marine propulsion system.

2. 3D MODELING AND MESHING

The structure of the marine propulsion shafting of the 48,000 tons bulk carrier consists of propeller, stern shaft, intermediate shaft, thrust shaft and main engine. The diagrammatic sketch of the marine propulsion system is showed in Fig.1. In order to carry out vibration characteristics analysis of the marine propulsion shafting more accurately, the 3D model of the marine propulsion shafting is established. The three-dimensional model is established in Pro/E according to the design paper. The established 3D model is imported into ANSYS for meshing. The finite element model of the marine propulsion shafting is showed in Fig.2. The finite element models of the whole propulsion shafting include: propeller, stern shaft, intermediate shaft, gear box shaft, sleeve hydraulic coupling and all kinds of bearings. The propeller modeling is divided into two parts: the hub and the blade. The hub is acted as a three-dimensional beam element, and the blade is centralized mass unit. The stern shaft, intermediate shaft, gear box shaft and sleeve coupling are all treated as three dimensional beam elements. The propeller is placed on the equivalent point of the propeller shaft end as a lumped mass unit, and the thrust bearing is taken as an axial spring unit, which is applied to the right end of the shafting. The finite element model of the shafting beam consists of 65 nodes, of which the beam element contains 56 nodes, and the other 9 nodes are the support of the shafting. The BEAM188 unit is used to divide the shafting structure. The bearing is represented by COMBIN14 spring element, and the shafting model consists of 309 beam units and 14 spring elements.

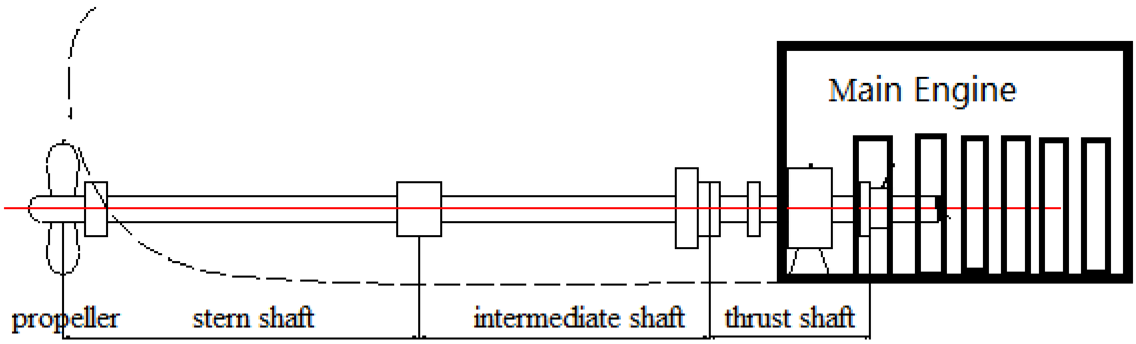


Fig. 1 The diagrammatic sketch of the marine propulsion system

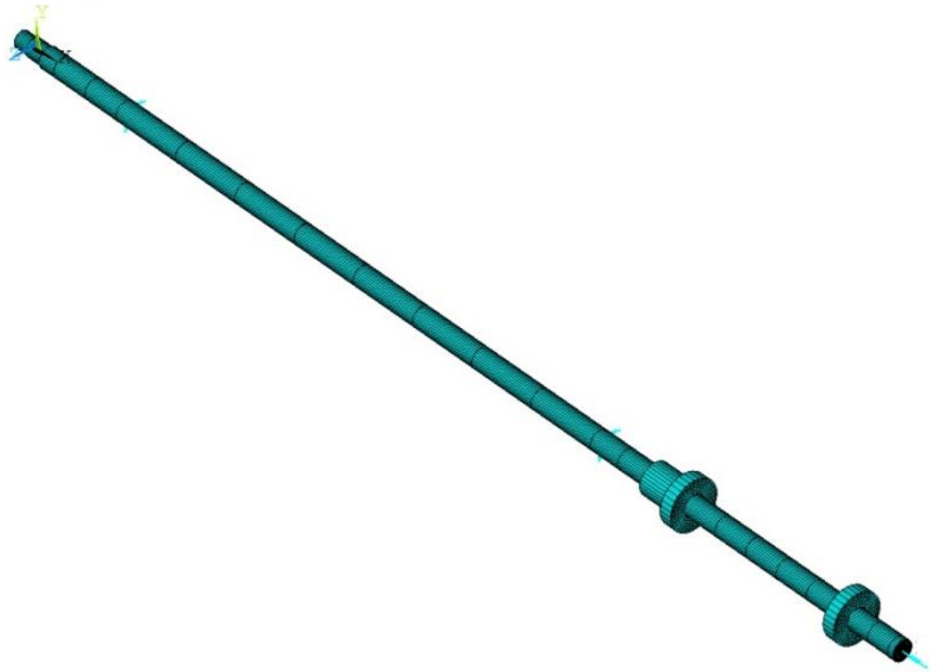


Fig.2 Finite element model of the marine propulsion shafting

3. CALCULATION RESULTS ANALYSIS

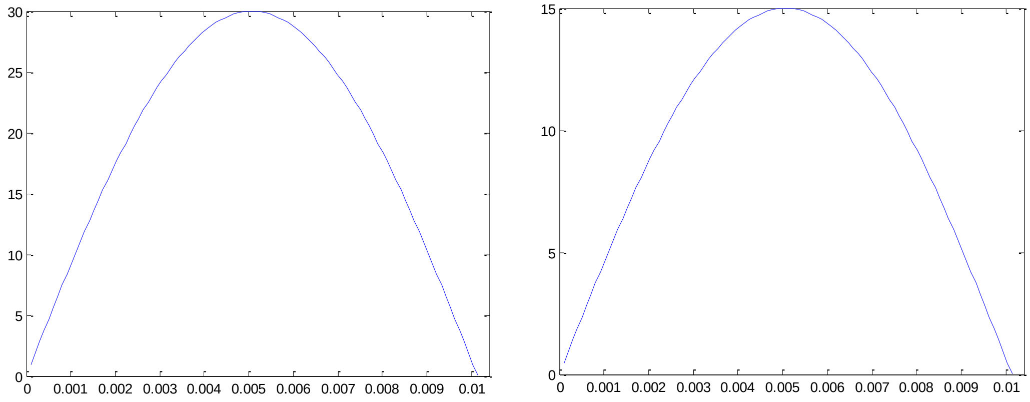
In this paper, the modal analysis of the marine propulsion shafting is obtained without imposing any boundary conditions, that is, free modal analysis. The natural frequency of the marine propulsion shafting is calculated in ANSYS. Through the simulation analysis, the first 10-order mode of the marine propulsion shafting are calculated, as shown in Table 1. According to the vibration theory, the low order mode plays an important role in the process of structural vibration. The contribution of the high order mode to the response is small and the vibration decays faster. Therefore, the paper only need to consider the low-order mode.

Table 1 natural frequency of the marine propulsion shafting

Order	vertical vibration	horizontal vibration	longitudinal vibration
1	9.6	9.6	22.3
2	12.3	12.3	82.6
3	13.7	13.7	125.1

The input load of the harmonic response analysis changes along sinusoidal with the dimensionless time. The load can be in the form of force, pressure and displacement. The parameters are mainly frequency and amplitude. The calculation result is usually expressed by the displacement, stress and strain of the nodes. As the basis for the structural vibration mechanism analysis and vibration reduction design, the peak response frequency and the response amplitude can be obtained by analyzing the curve of the output value versus frequency.

The transient impact response of shafting is calculated, and the dynamic shear and bending moments of each section are obtained. Accordingly, the corresponding dynamic shear stress and dynamic bending stress are calculated. The equivalent stress value of dangerous section is obtained by integrating it with static stress. The vertical impact load is 30g(the acceleration of gravity), and the horizontal impact load is 0.5 times the vertical impact load, which is 15g.The acceleration shock wave and the time history curve are showed in Fig.3.



(1) vertical impact acceleration (2) horizontal impact acceleration

Fig. 3 The acceleration shock wave and time history curve of the propulsion shafting

The bearing stress curve under impact load is shown in Fig.3. The maximum impact load is 1730 N. After 0.06 S, the impact load is reduced to less than 400 N. After 0.3 S, the vibration energy can be completely attenuated.

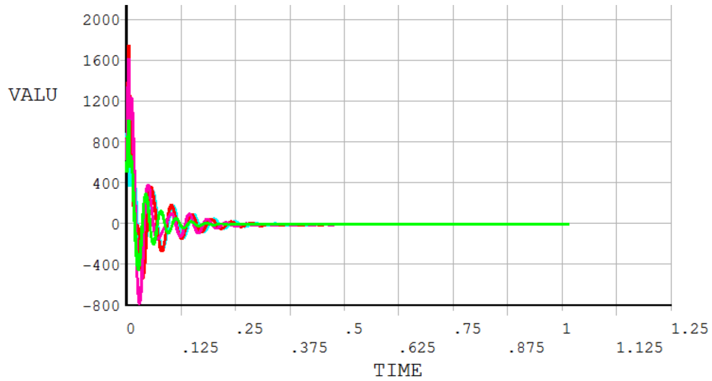
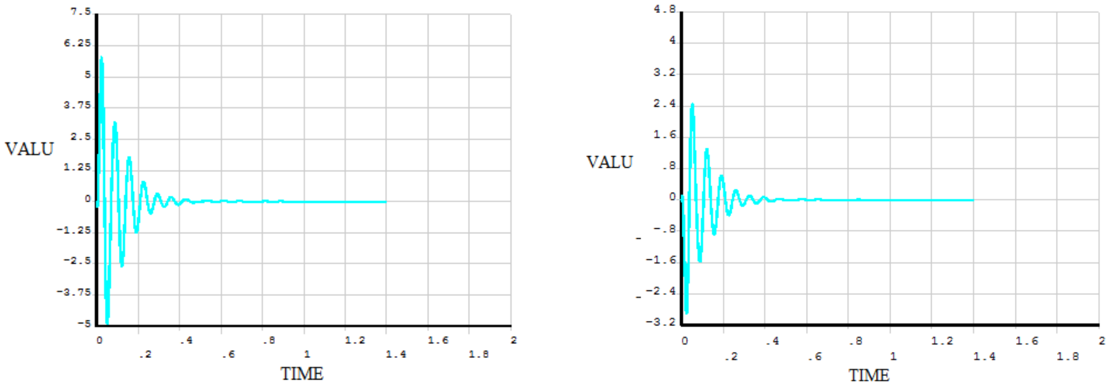


Fig. 4 The attenuation curve of the bearing reaction force under the impact load

Through the calculation of the maximum displacement of different impact direction, it can be seen that the maximum displacement response occurs at the position of the stern tube bearing when the vertical and lateral impact is impacted, and the maximum displacement response occurs at the rear stern bearing. The vertical impact displacement is larger than the horizontal and longitudinal impact. The displacement response curve of the 1# node under vertical and horizontal acceleration impact is showed in Fig.5.



(1)vertical (2) horizontal

Fig.5 The displacement response curve under the impact acceleration

The vertical acceleration attenuation curve under impact load is shown in Fig.6. The maximum response position in the shaft is located in the middle of the bearing, and the maximum response speed of the maximum response point on the intermediate shaft is 0.4 m/s^2. After 0.3 s, the shock vibration energy all attenuates.

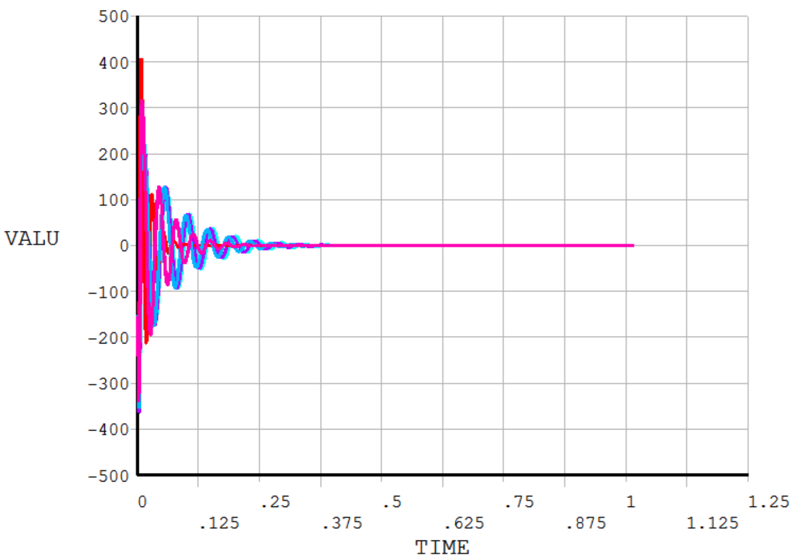


Fig.6 The vertical acceleration attenuation curve under impact load

4. SUMMARY

The harmonic response analysis was adopted to find out the relationship between the resonant frequency and the related vibration data, which lays the foundation for the reliability design and optimization of the marine propulsion shafting. This study lays the foundation for the reliability design and optimization of marine propulsion shafting. The main conclusions drawn from the results of this study are listed as follows:

(1) The first three natural frequencies in the longitudinal of the propulsion shafting are 22.3Hz, 82.6Hz and 125.1Hz.

(2) The maximum displacement response occurs at the position of the stern tube bearing when the vertical and lateral impact is impacted, and the maximum displacement response occurs at the rear stern bearing.

(3) The maximum response position in the shaft is located in the middle of the bearing, and the maximum response speed of the maximum response point on the intermediate shaft is 0.4 m/s^2.

REFERENCES

[1]. Murawski L. Axial vibrations of a marine shaft lines. Calculations-measurements comparison[A]. In 4th International Conference on Marine Technology, Wessex Institute of Technology, UK, WIT[C], 2001; 299-308.

[2]. Murawski L. Axial vibrations of a propulsion system taking into account the couplings and the boundary conditions [J]. Journal of marine science and technology, 2004, 9(4): 171-181

[3]. Vanem E, Bitner-Gregersen E M. Stochastic modelling of long-term trends in the wave climate and its potential impact on ship structural loads[J]. Applied Ocean Research, 2012, 37(8):235-248.

[4]. Cui Zhiqin，Yang Ruifeng． Parametric modeling and modal analysis for the complex mechanical structure. Chinese Journal of Mechanical Engineering, 2008, 44(2) : 234-237.

[5]. Gong C M, Shen H, Yao Z Q. Theoretical and Numerical Modal Analysis for Nuclear Coolant Pump and its Scaled Rotor System[J]. Applied Mechanics & Materials, 2011, 44-47:985-991.

[6]. Zhou Z H, Guang-Yi L I, Wang Y B. Modal analysis of the main structure and assembly of self-priming centrifugal pump based on Ansys[J]. Ship Science & Technology, 2016.

[7]. Tanrikulu O, Kuran B, Ozguven H N, et al. Forced harmonic response analysis of nonlinear structures using describing functions[J]. Aiaa Journal, 1993, 31(7)(7): 1313-1320.

[8]. Pindado S, Cubas J. Aerodynamic analysis of cup anemometers performance: the stationary harmonic response.[J]. The scientific world journal, 2013, 2013(403):197325.

[9]. Li L, Hu Y. Generalized mode acceleration and modal truncation augmentation methods for the harmonic response analysis of nonviscously damped systems[J]. Mechanical Systems & Signal Processing, 2015, s 52–53(February):46-59.