Academic Education Publishing House

Industry Science and Engineering Vol. 3 No. 8, 2026

Vibration Characteristics Analysis of the Propeller Shaft in a Deep-Sea Thruster

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Abstract: To gain the vibration characteristics of the propeller shaft of a certain type of deep-sea thruster in an oil medium and to understand the effect of oil viscosity on the structural modes, the simulation models of the propeller shaft and the oil are established separately. In addition to the dry modal vibration characteristics calculation of the propeller shaft on the ANSYS Workbench platform, the wet modal vibration characteristics calculation of the propeller shaft under the influence of oil viscosity is also carried out by using the fluid-structure interaction method. Through analysis of the simulation results, it is concluded that the oil has an obvious suppression effect on the vibration of the propeller shaft. In addition, by changing the viscosity of the oil, various wet modal vibration characteristics are obtained. The analysis results indicate that the greater the viscosity of the oil, the more prominent the suppression effect of the oil on the vibration of the propeller shaft is. This provides an important reference for the follow-up structural design of the propeller shaft and the selection of the oil type.

Keywords: Deep-Sea Thruster; Modal Analysis; Natural Frequency; Fluid-Structure Interaction

1. Introduction

As the device with the largest mass and the most frequently used in an underwater vehicle, the thruster generates vibration during operation, which is the main vibration source of the underwater vehicle [1,2]. The underwater thruster consists primarily of a motor, housing, propeller shaft and a propeller. The noise sources of the underwater thruster in its operation can be basically divided into two types: hydrodynamic noise and mechanical vibration noise, based on their formation mechanisms. The mechanical vibration noise is generated primarily due to the vibration of the propeller shaft at its natural frequency after departure from the equilibrium position during rotation and the transmission of the vibration to the housing through the bearing, thus generating the radiated noise. Therefore, how to take measures to reduce the vibration that is transmitted to the housing through the bearing has become the focus of discussion.

Many Chinese researchers have currently come up with effective solutions to the problem by reducing transmitted vibration. Zhang et al. [3] investigated a vibration reduction design method of the shafting based on finite element static simulation. They found that the stiffness of the shafting for the static case in all directions was determined by finite element simulation and the theoretical guidance on the vibration response of structure materials was obtained by the modal superposition method. Duan et al. [4] researched the metal rubber vibration isolation device which could be used to isolate the transmission of vibration from the shafting to the housing. Through their study, they found that the lower the stiffness of the metal rubber isolation ring, the better the vibration isolation effect, and the higher the speed of the shafting, the greater the vibration isolation effect. Zhao et al. [5] investigated the co-optimization of vibration control by a damped dynamic vibration absorber (DVA) and the multi-order modes of the longitudinal vibration of a hollow shafting. Through the finite element simulation, they also optimized the multi-stage vibration absorber, further improving its effect, which is expected to be a reference for future design methods of multi-order vibration absorbers. Zhao et al. [6] and Lu et al. [7] investigated the propulsion shafting with the vibration isolation equipment. By optimizing the vibration isolation treatment at multiple locations using the finite element simulation, their study gave theoretical

suggestions for future vibration isolator design methods. There are few studies on shafting vibration reduction starting from the shafting structure itself. Dan et al. [8] and Tang et al. [9] performed dry modal analysis of the rotor without prestress based on the ANSYS Workbench platform, and found that the vibration displacement of the shafting is larger at higher orders. Based on the dry modal analysis of the shafting, Zhu et al. [10], Wan and Wang [11], Wang and Cao [12] imposed rotational prestress and got the direction of the structural optimization by the first 8-order modal analysis using the Block Lanczos method. As for the research of wet modal analysis, Li et al. [13], Liang et al. [14] and Shi et al. [15] have studied the wet modal analysis of pipes with seawater outside and discovered that the seawater damping has a suppression effect on the vibration frequency and amplitude of the pipe.

In view of the above, as this paper deals with a deep-sea thruster, in which the housing is filled with oil to balance the inner and outer pressure so it can function normally, the vibration reduction study of shafting cannot be carried out only by analyzing the vibration isolation. It is also required to introduce oil as the medium for dry and wet modal analysis, and to change the viscosity of the oil in the medium of the propeller shaft to analyze the frequency and amplitude changes of the propeller shaft, and to obtain the relationship between viscosity and suppression coefficient. This paper is based on ANSYS Workbench, a new generation multi-physics collaborative CAE (computer aided engineering) simulation environment, which not only has all the functions of ANSYS Classic finite element simulation analysis software, but also has strong geometric modeling functions of CAD software like UG and PRO/E software [16,17]. In this paper, the structure of the propeller shaft of the underwater thruster is simulated by using finite element simulation platform ANSYS Workbench. With the related parameters adjusted, both dry and wet modal analyses are performed, which can provide theoretical support for engineering design and experiments.

2. Model Construction and Finite Element Simulation

2.1 Geometric Model Construction and Description

The deep-sea thruster developed by our research team is selected as the research object. The schematic diagram of the shafting transmission is shown in Figure 1. The shafting transmission system is composed of a propeller, self-lubricating bearings, a propeller shaft and angular contact ball bearings. For the optimized thruster, the propeller shaft structure is modified based on the standard model.

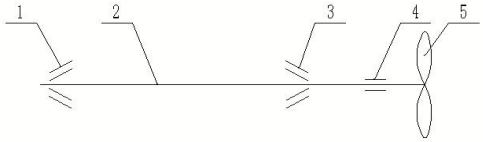


Figure 1. Schematic Diagram of the Underwater Thruster Structure

1. Angular contact ball bearing 2. Propeller shaft 3. Angular contact ball bearing 4. Self-lubricating bearing 5. Propeller

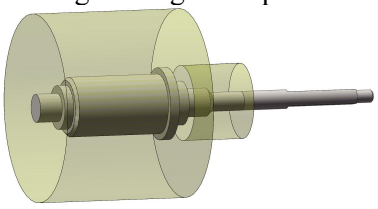


Figure 2. Schematic Diagram of the Coupling between Oil and the Propeller Shaft in the Deep-Sea Thruster

This kind of deep-sea thruster adopts a stepped and weight-reduction propeller shaft structure. This shaft structure can not only reduce the weight of the rotating parts of the overall components, but also ensure the coaxiality of the overall parts during assembly. These benefits include a lighter overall weight of the thruster without reducing its strength, a lower moment of inertia and the flexible rotation of the propeller shaft. The oil in the cabin of the deep-sea thruster is coupled with the propeller shaft as illustrated in Figure 2. Putting oil in the cabin can help to balance the internal and external pressures of the deep-sea thruster, and also can improve the heat dissipation and insulation of the circuits in the cabin.

2.2 Finite Element Simulation

First, SolidWorks is used to create models of the propeller shaft and the oil in the housing. Then, the modal analysis (Modal) module in the finite element simulation platform ANSYS Workbench is called to perform dry modal simulation analysis of the propeller shaft, and the fluid-structure interaction (FSI) module is

used to perform wet modal simulation analysis of the coupled body of the propeller shaft and oil. SolidWorks is a good choice for modeling due to its high modeling efficiency and convenient interface with ANSYS Workbench, so the analysis module can automatically create a data connection to share the geometric model with ANSYS Workbench. So, the model to be analyzed by modal simulation is imported into the ANSYS Workbench platform through SolidWorks and appropriate finite element analysis processing is performed on the model. Then the frequency and amplitude of the propeller shaft in dry and wet modes are compared and analyzed [18-20].

3. Establishment of the Computational Model

3.1 Introduction to Modal Theory

Through the modal analysis of the propeller shaft of this kind of deep-sea thruster, the natural frequency, mode shape and amplitude of the shaft can be obtained, which are important parameters of the structure under the action of dynamic load. Thus, modal analysis is an important approach for dynamic design and equipment fault diagnosis. Modal analysis is based on multi-degree-of-freedom dynamics. The governing dynamic equilibrium equation [21] is as follows:

$$M_S\ddot{u} + C_S\dot{u} + K_S u = T_S \quad (1)$$

When $T_S = 0$ and $C_S = 0$, the influence of oil damping is not considered, and it is free vibration. When solving the modes of the propeller shaft in air, the influence of damping can also be ignored, and the vibration equation [21] of the propeller shaft in air is obtained as:

$$[M]\{\ddot{u}\} + [K]\{u\} = 0 \quad (2)$$

where [M] is the structural mass matrix; [K] is the structural stiffness matrix; and {u} is the structural displacement.

3.2 Definition of Elements and Materials

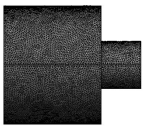
The propeller shaft of this type of deep-sea thruster is made of 0Cr17Ni12Mo2 stainless steel, with an elastic modulus of E=193 GPa, a Poisson's ratio of μ=0.3, and a density of ρ=7980 kg/m³. Other physical parameters are shown in Table 1.

Table 1. Dimensional parameters of the propeller shaft of the deep-sea thruster

Material	Maximum diameter D/mm	Length L/mm	Weight /kg
0Cr17Ni12Mo2	23	113.2	0.094

3.3 Mesh Generation

The propeller shaft of this type of deep-sea thruster is a simple stepped shaft. If the built-in meshing tool of ANSYS Workbench is used for meshing, this can meet the calculation requirements. There are convenient and high-quality meshing capabilities in ANSYS Workbench. Extension meshing, mapped meshing, free meshing and adaptive meshing are the methods involved [22-24]. As shown in Figure 3, the fluid domain of oil and the propeller shaft are meshed. Among them, the number of mesh elements in the fluid domain is 255,730 and the number of mesh elements in the propeller shaft structure is 176,263.



(a) Fluid domain mesh



(b) Structural mesh of the propeller shaft

Figure 3. Mesh Generation of the Fluid Domain and Structure

4. Modal Analysis of the Propeller Shaft

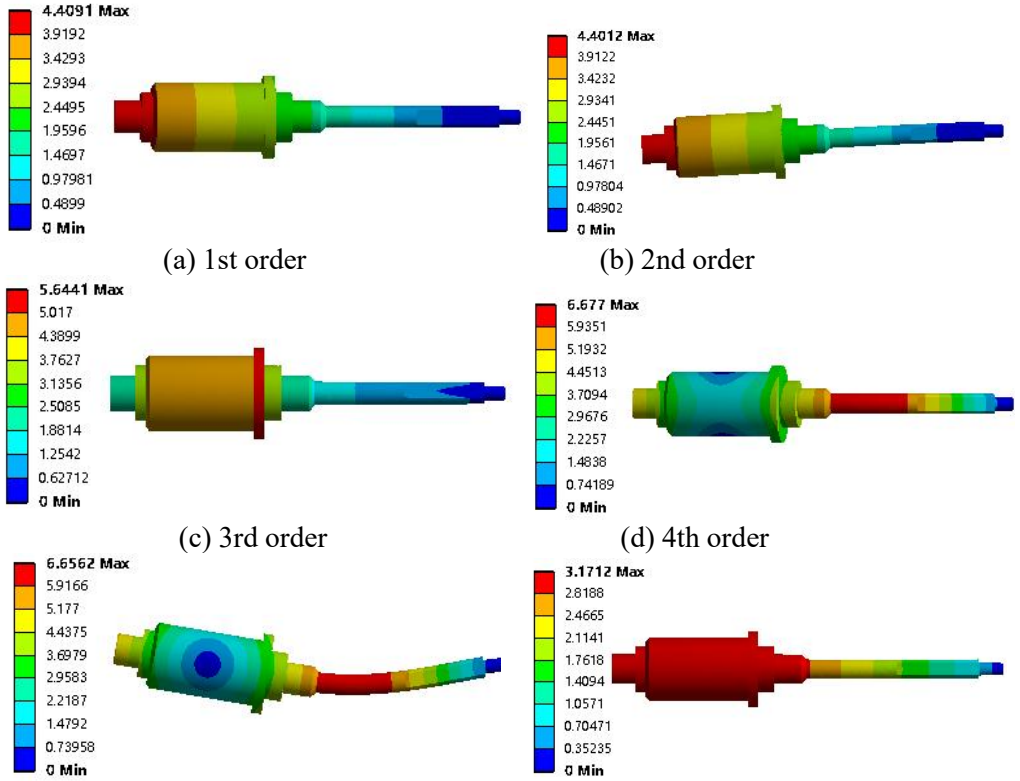
4.1 Dry Modal Analysis of the Propeller Shaft

To determine the vibration characteristics of a designed structure or machine part, modal analysis is used. Dry modal analysis is the study of natural frequencies and mode shapes of a structure under vacuum conditions. The first 6 orders of natural frequencies of the vibrating shaft [25] were computed using the Block Lanczos method. Let $f\alpha$ be the natural frequency of the propeller shaft and Lα be the maximum value of each mode shape of the shaft. The natural frequencies of each mode shape of the propeller shaft are given in Table 2 and the mode shape diagrams of the propeller shaft are shown in Figure 4.

Table 2. Analysis Results of the First 6 Orders of Natural Frequencies of the Propeller Shaft

Order	Natural frequency f0/Hz	Maximum value L0/mm	Location of maximum value
1	79.296	4.4091	Front end of the propeller shaft

2	85.689	4.4012	Front end of the propeller shaft
3	769.82	5.6441	Middle part of the propeller shaft
4	1319.5	6.677	Middle-rear part of the propeller shaft
5	1342.8	6.6562	Middle-rear part of the propeller shaft



(a) 1st order (b) 2nd order

(c) 3rd order (d) 4th order

(e) 5th order (f) 6th order

Figure 4. Dry Modal Mode Shape Diagrams of the Propeller Shaft

4.2 Comparative Analysis of Dry and Wet Modes of the Propeller Shaft

A wet modal analysis is the analysis of natural frequencies and mode shapes of a structure under a certain medium environment [26,27]. The rotation of the propeller shaft makes the oil rotate by friction between the propeller shaft and the oil as shown in the flow field in the oil region in Figure 5, and the oil produces a reaction force on the surface of the propeller shaft. So when the propeller shaft rotates in the housing of the deep-sea thruster, its modal results will inevitably be affected by the oil in the housing.

No. 46 oil in this type of deep-sea thruster is set as the working medium of the propeller shaft. Through the fluid-structure interaction (FSI) module in the ANSYS Workbench platform, the first 6 orders of wet modal analysis of the propeller shaft are performed, and the mode shape diagrams are shown in Figure 6.

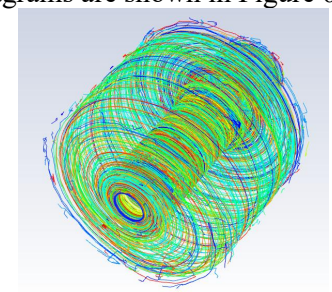
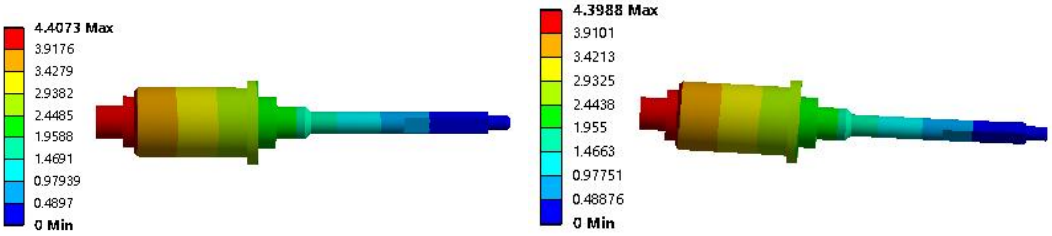
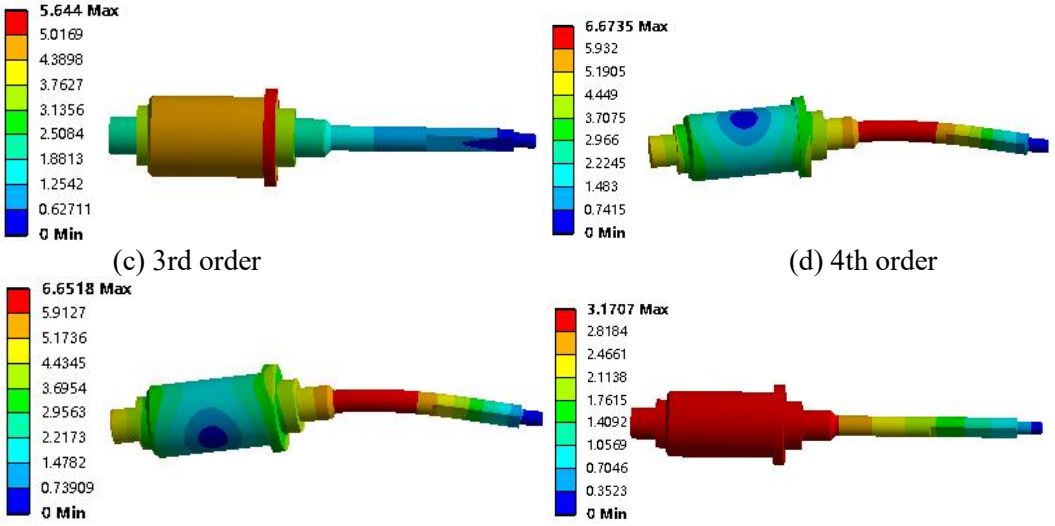


Figure 5. Flow Field Diagram of the Oil Region



(a) 1st order (b) 2nd order



(c) 3rd order (d) 4th order

(e) 5th order (f) 6th order

Figure 6. Wet Modal Mode Shape Diagrams of the Propeller Shaft

By comparing the wet modal data with the dry modal data, the first 6 orders of vibration frequencies (Table 3) and the vibration frequency curves (Figure 7) of the two are obtained. Let $f\beta$ be the wet modal natural frequency of the propeller shaft and let ε be the influence coefficient of the added oil on the propeller shaft, which is the ratio of $f\beta$ to $f\alpha$. The result is shown in Figure 8.

Table 3. First 6 Orders of Dry and Wet Modal Natural Vibration Frequencies of the Propeller Shaft

Order	Dry modal natural frequency $f\alpha$/Hz	Wet modal natural frequency $f\beta$/Hz	Influence coefficient ε
1	79.296	63.464	1.2946
2	85.689	70.532	1.2149
3	769.82	766.61	1.0042
4	1319.5	1312.1	1.0056
5	1342.8	1333.3	1.0071
6	4182.1	4155.4	1.0064

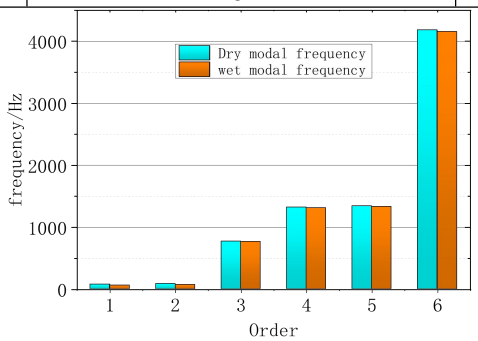


Figure 7. Dry and Wet Modal Natural Frequency Curves of the Propeller Shaft

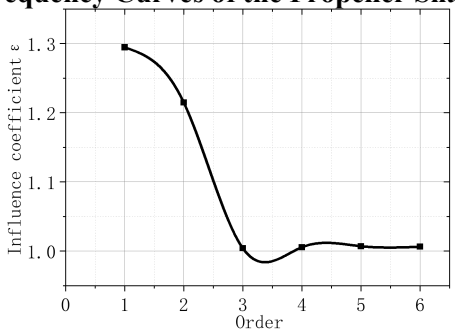


Figure 8. Influence Coefficient Curve

By comparing the results displayed in Table 3 and Figure 8, it can be seen that the influence of the vibration characteristics of the propeller shaft is different under the dry and wet modal conditions at different orders. The influence coefficient of the oil on the natural frequency of the propeller shaft is between 1.2149 and 1.2946. It can be analyzed that, for the first 3 orders, the suppressive effect of the oil on the vibration of the propeller shaft is very obvious, and it is the highest at the 1st order. The higher the order, the less the suppression effect. The influence coefficient is approximately the same from the 3rd order to the 6th order and the suppression effect is also the same. For the study of the amplitude of the propeller shaft, let the wet modal amplitude of the propeller shaft be Lβ, and the suppression coefficient of the oil on the propeller shaft, that is, the ratio of the wet modal amplitude to the dry modal amplitude be δ. The first 6 orders of dry and wet modal amplitudes of the propeller shaft are

shown in Table 4 and Figure 9, and the relationship between the suppression coefficient δ and the order is shown in Figure 10.

Table 4. First 6 Orders of Dry and Wet Modal Amplitudes of the Propeller Shaft

Order	Dry modal amplitude Lα/mm	Wet modal amplitude Lβ/mm	Suppression coefficient δ
1	4.4091	4.4073	1.00041
2	4.4012	4.3988	1.00055
3	5.6441	5.644	1.00002
4	6.677	6.6735	1.00052
5	6.6562	6.6518	1.00066
6	3.1712	3.1707	1.00016

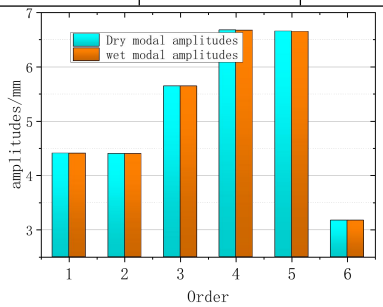


Figure 9. Dry and Wet Modal Amplitudes of the Propeller Shaft

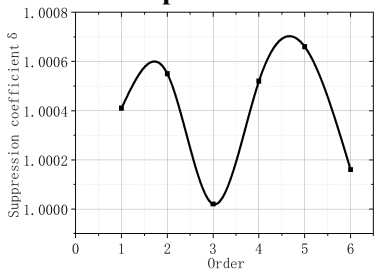


Figure 10. Suppression Coefficient Curve

It can be seen from the comparison of the results in Table 4 and Figure 10 that there is a difference between the degrees of influence of dry and wet modal conditions on the vibration characteristics of the propeller shaft at different orders. Based on Table 4, the suppression coefficient of the wet modal amplitude relative to the dry modal amplitude of the propeller shaft is in the range 1.00002 to 1.00066. Based on the suppression coefficient δ and order relation curve in Figure 10, it can be seen that the suppression of the oil on the amplitude of the propeller shaft is the most prominent at the 3rd and 6th orders.

4.3 Analysis of the Influence of Viscosity on Wet Modes

This deep-sea thruster has oil as the medium in which the propeller shaft rotates. Since the viscosity of the oil will have a suppression effect on the vibration of the propeller shaft, it is very important to select oil with an appropriate viscosity to suppress the vibration of the propeller shaft. In this work, 6 kinds of oil are chosen to conduct the wet modal analysis of the first six orders of the propeller shaft based on the fluid-structure interaction with the surface of the propeller shaft.

By conducting experiments on such thrusters, it is found that the oil temperature in the cabin is approximately 70 ℃ for normal operation of the motor. The viscosities of different types of oil that can be injected and used in the cabin at a temperature of 70 °C are shown in Table 5.

Table 5. Comparison of Viscosity Parameters of the Oil

Type	22	32	46	68	100	150
Viscosity mm2/s(70℃)	8.5	11	15	20	27	41

According to the fluid-structure interaction (FSI) analysis of the oil and the propeller shaft, when the viscosity of the oil is changed, the first 6 orders of natural frequencies of the bending vibration of the propeller shaft are obtained from the analysis results, where $f1$ is the natural frequency of the propeller shaft when type 22 oil is coupled with the propeller shaft, $f2$ is the natural frequency of the propeller shaft when type 32 oil is coupled with the propeller shaft, $f3$ is the natural frequency of the propeller shaft when type 46 oil is coupled with the propeller shaft, $f4$ is the natural frequency of the propeller shaft when type 68 oil is coupled with the propeller shaft, $f5$ is the natural frequency of the propeller shaft when type 100 oil is coupled with the propeller shaft and $f6$ is the natural frequency of the propeller shaft when type 150 oil is coupled with the propeller shaft. The results are displayed in Table 6 and the natural frequencies of the propeller shaft are plotted and displayed in the form of a bar chart, as shown in Figure 11.

Table 6. First 6 Orders of Natural Frequencies of the Propeller Shaft under Different Viscosities

Order Frequency /Hz	$f1$	$f2$	$f3$	$f4$	$f5$	$f6$
1	63.764	63.714	63.659	63.599	63.534	63.464

2	71.332	71.182	71.027	70.867	70.702	70.532
3	767.76	767.51	767.36	767.16	766.91	766.61
4	1314.2	1313.9	1313.6	1313.2	1312.7	1312.1
5	1337.4	1336.8	1336.2	1335.3	1334.4	1333.3
6	4172.9	4171.4	4168.9	4165.4	4160.9	4155.4

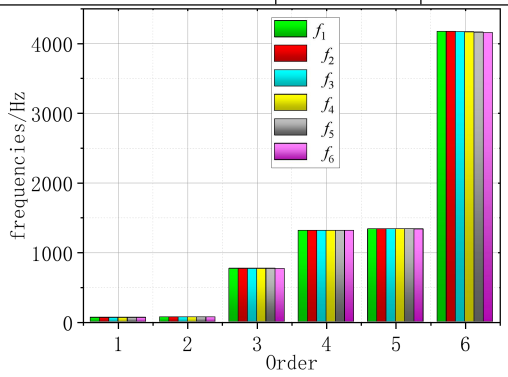


Figure 11. Line Chart of Wet Modal Natural Frequencies of the Propeller Shaft

According to Table 6 and Figure 11, it can be analyzed that changing the viscosity of the oil will change the natural frequency of the propeller shaft. A rough analysis shows that increasing the viscosity of the oil will reduce the natural frequency of the propeller shaft. In order to better analyze the coupled vibration characteristics of the oil and the propeller shaft, the results are further processed. The natural frequency change rate of the propeller shaft after coupling with type 22 and type 32 oil is defined as $\eta12=(f1-f2)/f1$; the natural frequency change rate after coupling with type 32 and type 46 oil is defined as $\eta23=(f2-f3)/f2$; the natural frequency change rate after coupling with type 46 and type 68 oil is defined as $\eta34=(f3-f4)/f3$; the natural frequency change rate after coupling with type 68 and type 100 oil is defined as $\eta45=(f4-f5)/f4$; and the natural frequency change rate after coupling with type 100 and type 150 oil is defined as $\eta56=(f5-f6)/f5$. The results are shown in Table 7, and a line chart of the coupled natural frequency change of the propeller shaft is plotted, as shown in Figure 12.

From Table 7, it can be analyzed that by comparing the change rates among oils of different viscosities, when coupled with type 22 and type 32 oil, the relative change rate of the natural frequency of the propeller shaft is between 0.023% and 0.21%; when coupled with type 32 and type 46 oil, the relative change rate of the natural frequency of the propeller shaft is between 0.02% and 0.218%; when coupled with type 46 and type 68 oil, the relative change rate of the natural frequency of the propeller shaft is between 0.026% and 0.225%; when coupled with type 68 and type 100 oil, the relative change rate of the natural frequency of the propeller shaft is between 0.033% and 0.232%; and when coupled with type 100 and type 150 oil, the relative change rate of the natural frequency of the propeller shaft is between 0.039% and 0.24%. The natural frequency change rate of the propeller shaft is shown on the line chart of Figure 12, from which it is intuitively understood that the natural frequency of the propeller shaft can be decreased by increasing the viscosity of the oil to a certain extent. In the changes of the first 3 orders, it can be seen that the change rate of the vibration frequency suppressed by the 6 types of oil compared with oil of the adjacent viscosity is similar, and the suppression effect increases linearly. The suppression effect of the 6 oil types on the vibration of the propeller shaft rises in an exponential manner following the 3rd order, and the higher the oil viscosity, the greater will be the suppression effect.

Table 7. Natural frequency change rates of the propeller shaft under different viscosities

Order Change rate /%	η12	η23	η34	η45
1	0.078	0.086	0.094	0.102
2	0.210	0.218	0.225	0.232
3	0.032	0.020	0.026	0.033
4	0.023	0.023	0.030	0.038
5	0.045	0.045	0.067	0.067

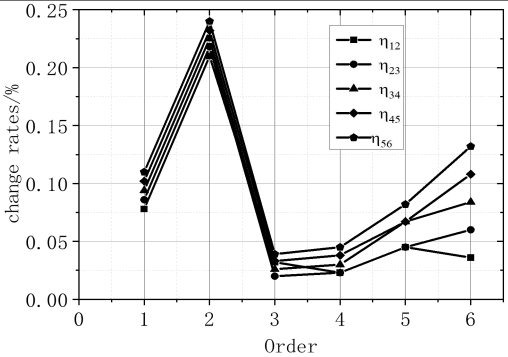


Figure 12. Line Chart of Frequency Change Rates

5. Conclusions

Based on the theoretical analysis, combined with the research on the actual operating

environment, this study provides certain guidance for the selection of oil filled in the deep-sea thruster, and provides ideas and directions for reducing the vibration of the deep-sea thruster and extending its service life.

1) The natural frequency of the propeller shaft can be effectively reduced by filling the housing of the deep-sea thruster with oil. The suppression effect on the natural frequency is most clearly seen at low-order vibrations, and with the increase of the order, the suppression effect decreases and remains gentle.

2) When the vibration orders are the same, the mode shape of the propeller shaft in oil is similar to that in air, and the ratio of the wet modal amplitude to the dry modal amplitude is 1.00002-1.00066. The analysis shows that the oil obviously suppresses the amplitude, and the suppression coefficient changes gently from low to high order and the suppression effect is the same from the 1st to 6th order.

3) The higher the oil viscosity, the greater the suppression effect on the natural frequency of the propeller shaft. By comparing the first 6 orders of modal analysis of the propeller shaft, the change rate of the natural frequency of the propeller shaft under oils of different viscosities varies between 0.02% and 0.24%. The suppression effect on the natural frequency is linear in the first 3 orders of modal analysis, and in the latter 3 orders of modal analysis, the suppression effect on the natural frequency is exponential, and the higher the viscosity, the greater the suppression effect on the natural frequency.

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