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Analysis on the Effect of Support Structure Deformation on the Vibration Characteristics of Drive Shaft

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ABSTRACT

The drive shaft is important part of power transmission system for most mechanical equipment, which has directly influence on not only keeping normal operation crucially in the set course, but also the vibration of equipment. There are several influence essential factors of the vibration characteristic. Support structure deformation is one important factor with incapable avoidance. In worst case, it would lead the heavy damage of equipment. The simplified model of typical drive shaft was the narrow sense concept for this study. This paper used research means including theoretical analysis 、 numerical calculation and experimental verification. Based on the dynamic equations, this paper aims to establish the dynamic model of the drive shaft. The analytical results of the model are compared with the numerical and experimental results to confirm that the model is reliable. The shaft vibration characteristics are studied under structure deformation with different parameters respectively. The analysis of model experiment can guide the design and installation technology of drive shaft .Groundwork is made for further research in shaft vibration problem considering support structure deformation.

Keywords: Support structure deformation, Shaft, Bearing
I-INCE Classification of Subject Number: 76

1. INTRODUCTION

The vibration characteristics of ships has always been the focus of research. Drive shaft is one of most important components of ships, which will directly affect the normal use of ships. The vibration characteristics of drive shaft has also attracted much attention. Generally speaking, drive shaft is affected by two parts of excitation force, one is propeller force which transmission through shaft causing the vibration of ship, the other is the instability of the rotation of propulsion shaft which causing the vibration of ship. With the advent of large ships, the difficulty of shaft design is increasing. At the same time, the requirements of vibration performance are also raised.

The vibration performance of propulsion shaft is affected in many ways, for example the propeller, the design of propulsion shaft ,the support structure. The structure of ship is used as support structure to provide support for stable running of shaft. The ship is sailing on the water. The hull will be deformed under different draft, which will directly

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affect the vibration performance and operation of propulsion shaft. This paper focuses on the analysis and study of the influence of ship structure deformation on the vibration performance of propulsion shaft.

2. MODEL OF PROPULSION SHAFT

A simplified model consisting of intermediate shaft, stern shaft and supporting bearings is established, which is used for analyzing of shafting vibration performance caused by ship hull deformation. As shown in figure 1. The hull deformation is transmitted to the shaft at the supporting bearings, which caused vibration.

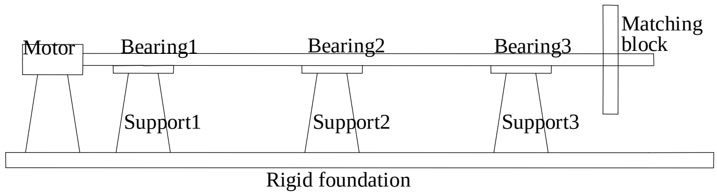


Figure1 Simplified model of propulsion shaft

Supposing that a ship has three elastic supporting bearings, whose position are B_1、B_2、B_3.The stiffness of bearings are K_1、K_2、K_3.The bearings divide the shaft into four parts with length L_1、L_2、L_3、L_4. The shaft is homogeneous beam in simplified model. The area of cross section is S, and the density is ρ.The oil film force between bearing and shaft and the nonlinear effect of bearing damping are not considered. With Euler-Bernoulli beam theory, the vibration equation of each shaft can be expressed as Equation 1[1]:

$$E_i I_i \frac{\partial^4 U_i(x,t)}{\partial x} + \rho_i \frac{\partial^2 U_i(x,t)}{\partial t^2} = 0, i = 1,2,3,4$$

Equation 1

$E_i I_i$ is the bending rigidity of shaft i, t is the time.

By using the method of separating variables, the solution of the equation can be expressed as Equation 2:

$$U_i(x,t) = \Phi_i(x) Z_i(t) \qquad \text{Equation 2}$$

$Z_i(t)$ is the generalized coordinates of each shaft segment. $\Phi_i(x)$ is the modal function of each shaft segment.

It can be seen that the amplitude of vibration performance of shaft segment varies with time according to $Z_i(t)$,and the vibration form with $\Phi_i(x)$.The derivative to x expressed by apostrophe. The derivative of t is represented by a dot. By using Equation 2 to bring into Equation 1 can be obtained Equation 3:

$$\frac{\Phi_i''(x)}{\Phi_i(x)} + \frac{\rho_i}{E_i I_i} \frac{\ddot{Z}(t)}{Z(t)} = 0 \qquad \text{Equation 3}$$

$\frac{\Phi_i''(x)}{\Phi_i(x)}$ is a function of x. $\frac{\ddot{Z}(t)}{Z(t)}$ is a function of t. Therefore, only when two of the following conditions are met as Equation 4.

$$\frac{\Phi_i''(x)}{\Phi_i(x)} = -\frac{\rho_i}{E_i I_i}\frac{\ddot{Z}(t)}{Z(t)} = a_i^4 \qquad \text{Equation 4}$$

where $a_i^4 = \dfrac{\omega_i^2 \rho_i}{E_i I_i}$, ω_i is the structure circular frequency of each shaft segment. $\Phi_i(x)$ can be given as Equation 5[2]:

$$\Phi_i(x) = A_i \cos a_i x + B_i \sin a_i x + C_i \cosh a_i x + D_i \sinh a_i x \qquad \text{Equation 5}$$

where A_i 、 B_i 、 C_i 、 D_i are the real constants of each shaft segment.

The continuous conditions of the displacement、rake ratio、bending moment、shearing force of each shaft at the bearing are shown in the following formula under free vibration conditions because of the presence of multiple bearings.

$$U_i(X^L{}_i, t) - U_{i+1}(X^R{}_i, t) = 0$$

$$\frac{\partial U_i(X^L{}_i, t)}{\partial x} - \frac{\partial U_{i+1}(X^R{}_i, t)}{\partial x} = 0$$

$$E_i I_i \frac{\partial^2 U_i(X^L{}_i, t)}{\partial^2 x} - E_i I_i \frac{\partial^2 U_{i+1}(X^R{}_i, t)}{\partial^2 x} = 0$$

$$E_i I_i \frac{\partial^3 U_i(X^L{}_i, t)}{\partial^3 x} - E_i I_i \frac{\partial^3 U_{i+1}(X^R{}_i, t)}{\partial^3 x} - K_i U_{i+1}(X^R{}_i, t) = 0$$

$X^L{}_i$, $X^R{}_i$ are the coordinates of the left section and the right section of the bearing, respectively. K_i is the rigidity of supporting bearing.

The influence of the propeller, the flange and the main machine shall not be taken into account for the time being. So the boundary conditions at both ends of the shaft are shown in the following.

$$U''(0,t) = 0 \text{，} \quad U'''(0,t) = 0$$

$$U_{n+1}''(0,t) = 0 \text{，} \quad U_{n+1}'''(L,t) = 0$$

3.NUMBERICAL ANALYSIS OF PROPULSION SHAFT WITH DEFORMATION

Hull deformation is mainly affected by loading state change and marine environment. This paper only considers the influence of hull deformation caused by the change of loading state.

In general, the deformation of the hull of the support part can be obtained in advance by using the finite element software. The deformation of the hull can be calculated under the condition of sitting pier, no load and full load with loading of hydrostatic pressure on the outer plate of the hull. The relative deformation is obtained by comparing and analyzing of deformation in no-load, full-load and pier state. The vertical deformation is considered due to small transverse deformation during navigation.

The shaft models of ship are established in the condition of sitting pier, no-load and full-load by using the BEAM188 unit in the ANSYS software with the relative displacement of the centerline of the shaft obtained from calculation. The shaft is basically free of relative displacement in the bearing-pier state. The finite model of shaft is shown in figure 2 ,and the parameter is shown in Table 1.

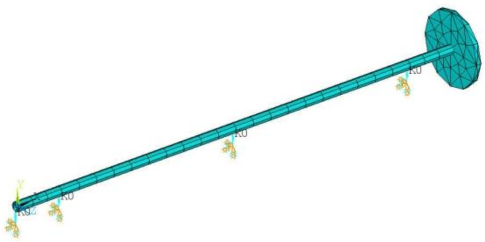


Figure2 Finite model of shaft

Table1 Parameter of propulsion shaft segments and bearings

Item	Value
Diameter of shaft	0.098m
L_1	0.5m
L_2	2m
L_3	2m
L_4	0.51m
B_1 （B_2、B_3 same as B_1）	1×10^9N/m

The crankshaft is established based on relative deformation curve of centerline of shaft to simulate the no-load and full-load state of ship. Assume that the stiffness of each bearing is the same, the shaft has the same vibration modes in vertical and transverse directions. In modal analysis, only the degree of freedom of beam element in the direction of vertical displacement and longitudinal rotation angle is reserved. The ends of the bearings simulated by spring element are fully constrained. The natural frequency of the shaft can be obtained as shown in Table 2.

Table2 Natural frequency of the shaft(Hz)

	Simulate	Numerical
First	24.5	23.64
Second	61.3	59.57
Third	74.1	71.26

4. ANALYSIS OF MODEL TEST

Based on the above analysis, a propulsion shaft bench is designed as shown in figure 3 for depth analysis.

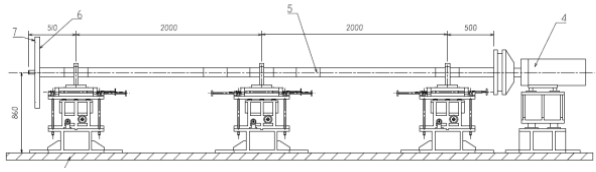


Figure3 Drawing of test bench

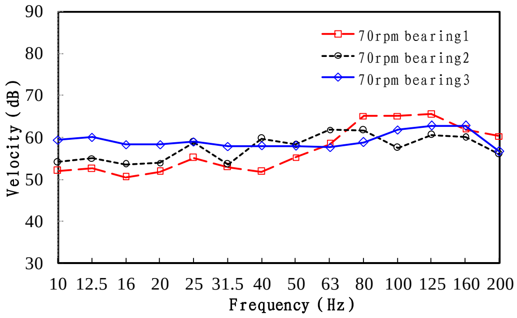
Figure4 Photograph of test bench

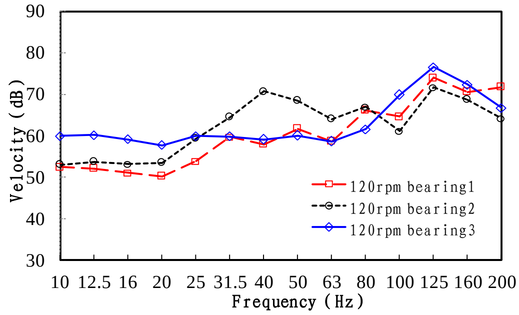
In order to compare and analyze the test results, the test conditions can be divided into two categories, as shown in Table 3. The vertical height of the supporting parts of the intermediate bearing and the stern bearing were changed respectively. The initial state is well aligned with the shaft. The test was carried out under multiple rotational speed conditions.

Table3 Operation conditions of the test

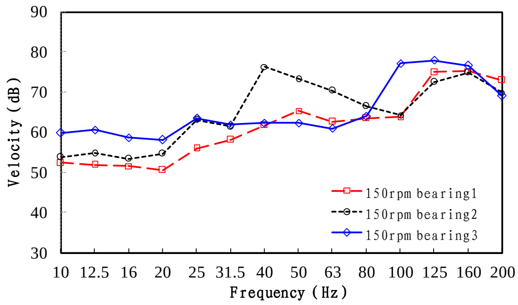
	Condition number	Relative displacement（mm）		
		Front bearing	Intermediate bearing	Stern bearing
Initial	0	0	0	0
1	1-1	0	0.5	0
	1-2	0	1	0
	1-3	0	1.5	0
	1-4	0	2	0
	1-5	0	-0.5	0
	1-6	0	-1	0
	1-7	0	-1.5	0
	1-8	0	-2	0
2	2-1	0	0	0.5
	2-2	0	0	1
	2-3	0	0	1.5
	2-4	0	0	2
	2-5	0	0	-0.5
	2-6	0	0	-1
	2-7	0	0	-1.5
	2-8	0	0	-2

First of all, the vibration characteristics of the supporting parts in the initial state are analyzed as shown in figure 5. The magnitude of vibration in the intermediate supporting bearing is higher than that in the other two, because of deflection of shaft caused by weight of distribution block. It has effect both on the vibration of the stern bearing and the intermediate bearing. The vibration magnitude is the highest at the stern bearing below 25Hz, second is the intermediate bearing. The bearing near the motor end has the lowest vibration. With the increase of the rotational speed, the vibration of the intermediate bearing increases in the 25Hz~80Hz band. The vibration magnitude of the other two are relatively flat. The magnitude of vibration at the stern bearing has returned to the highest level in the 100Hz~200Hz band.





(a) (b)

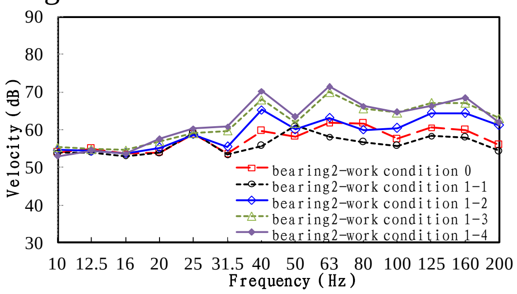


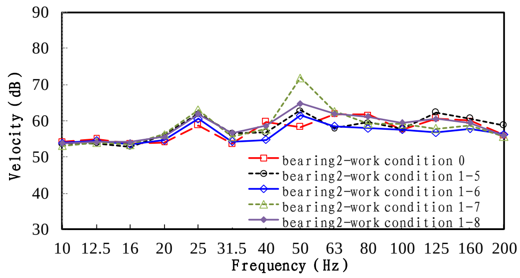
(c)

Figure5 Vibration comparison of various supporting bearing in initial state

Working condition 1 change the vertical relative height of the intermediate bearing as shown in figure 6. The vibration magnitude of the intermediate bearing after the upward adjustment of height is basically equivalent to the initial state below 25Hz. The magnitude of vibration increases with the vertical height at the peak frequency. It tends to flatten when increased to a certain level.

The vibration regularity is not obvious when the vertical height of the intermediate bearing changes downward. The vibration change at the stern bearing is not obvious in working condition 1.





(a) (b)

Figure6 Vibration comparison of supporting bearing2 in working condition 1

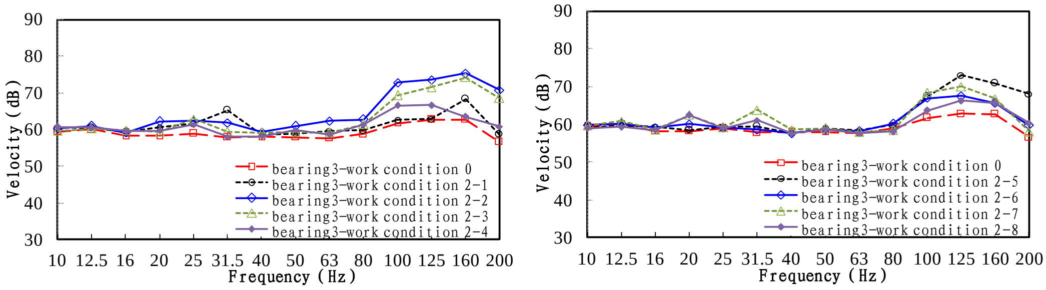
The total vibration acceleration at bearings in working condition 1 is shown in Table 4 in the 10Hz~200Hz band. The vibration of the intermediate bearing in working condition1 is lower than the initial state in some conditions. The vibration of the stern bearing in working condition1 is higher than the initial state.

Table4 Total vibration acceleration in working condition 1（dB）

Condition number	Bearing2	Bearing3
0	69.9	71.1
1-1	68.3	72.5
1-2	72.5	72.1
1-3	76.0	73.3
1-4	77.0	73.7
1-5	70.5	74.2
1-6	68.8	73.0
1-7	73.9	73.7
1-8	71.1	73.5

Working condition 2 change the vertical relative height of the stern bearing as shown in figure 7. The vibration magnitude of the stern bearing after the upward adjustment of height is basically equivalent to the initial state below 80Hz. But the vibration characteristics of the stern bearing are different from the working condition1. The magnitude of vibration at the stern bearing increases significantly in the 100Hz~200Hz band. However, there are no significant correlation between the growth and the height change of the stern bearing.

The vibration characteristics of vertical height downward state of the stern bearing are different from that of upward change. There are no significant correlation between the growth and the height change of the stern bearing.

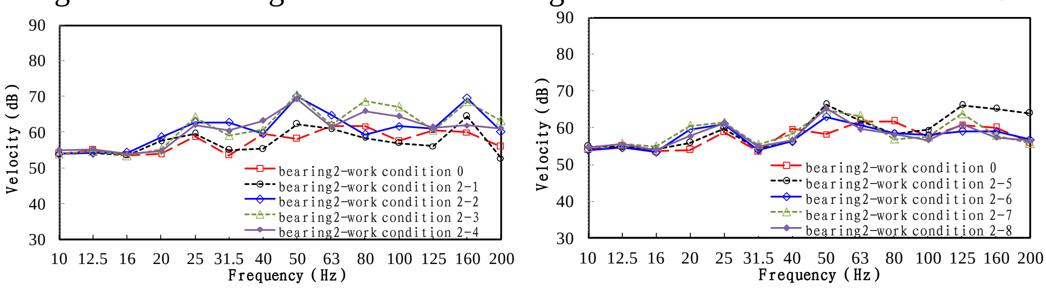


(a) (b)

Figure7 Vibration comparison of supporting bearing3 in working condition 2

The magnitude of vibration in the intermediate bearing is also obvious in working condition 2 as shown in figure 8. When the vertical height of the stern bearing position changes upward, the intermediate bearing is actually lower than the initial state because of the deflection of the shaft. There are still no significant correlation between the growth of the magnitude of vibration at the intermediate bearing and the height change of the stern bearing.

The vibration of the stern bearing and the intermediate bearing changes greatly when adjusting the relative height of the stern bearing which is more difficult to control.



(a) (b)

Figure8 Vibration comparison of supporting bearing2 in working condition 2

The total vibration acceleration at bearings in working condition 2 is shown in Table 5 in the 10Hz~200Hz band. The vibration of the stern bearing in working condition 2 is much higher than the initial state.

Table5 Total vibration acceleration in working condition 2（dB）

Condition number	Bearing2	Bearing3
0	69.9	71.1
2-1	70.2	73.8
2-2	75.3	80.1
2-3	76.3	78.3
2-4	74.3	73.5
2-5	73.1	77.2
2-6	70.1	73.8
2-7	71.5	75.2
2-8	70.5	73.0

5. CONCLISIONS

Combined with the above calculation, analysis and research, the following conclusion can be drawn.

1)The effect of hull deformation on the vibration characteristics of the intermediate bearing is less than that on the stern bearing. The vibration characteristics of the intermediate bearing is controllable when the deformation fluctuates in a small range.

2)The influence of hull deformation on the stern bearing is relatively large , which is consistent with the actual situation. From the analysis of the tests, it can be seen that the regularity of vibration change is not obvious. But in a word, the vibration of the stern bearing is relatively controllable when the hull deformation changes vertically downward. It has certain reference significance for the shaft to adopt the corresponding installation technology improvement measures in the shaft alignment.

6. REFERENCES

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