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Automation System Vibration Analysis Taking Environmental Factors into Consideration

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Abstract

This paper aims to investigate the vibration behavior of a propulsion system subjected to hull deformations in a two dimension circumstance. As known that large scale ships have great developments in recent years which could cause much severer conditions among the interaction between the propulsion system and ship hull. Excited forces from these waves could make the ship hull deformed which further cause drastic vibrations of the shaft system. As a result, the malfunctions of shaft propulsion system are potential existed as the vibrations of the shaft always exceed its maximum allowable values. This paper establishes a simplified model of the large ship propulsion-hull system to analyze the vibration behavior of the ship propulsion system subjected to the ship hull deformations. The hull deformations were obtained as the excited forces under different sea conditions. Then base on the simplified 2D model, the effects of propeller, supports stiffness, the location of hull excitations, the amplitude of excitations are discussed.

Keywords: *hull deformations, propulsion, vibration behavior, variable parameter*

1. Introduction

The ship plays an important role in the international transportation. Large vessels are built to meet the rapid needs of the worlds' trade in last twenty years. As a key component of the ship power system, the propulsion system transfers the power from the engine to the propellers which produce a thrust to keep the ship moving. The larger size of the ship promotes the parameters of the propulsion such as structures, output power and becoming larger than before which could be easily effected by ship hull deformations and other excitations [1]. Based on the obvious problems, many institutions and scholars have focused on the interactions between the ship hull and the propulsion system. The ABS of U.S.A. developed a new ABS shaft alignment system taking the effect of hull deformations into account [2]. Recent years, the CCS of China also expands the COMPASS shaft calculation system which considered the factor of ship hull deformations [3]. Scholars such as Lech Murawski, Xing, Figureari, Wang, Zhou have been done many works on the interaction between the shaft alignment and ship hull deformations. Longitudinal vibrations of a propulsion system taking the coupling and boundary conditions into considerations have been studied by Lech Murawski [4]. Xing [5] uses finite element models and substructure approaches to analyze the coupled fluid-soild interactions. Figureari [6] studies the dynamic behavior of marine propulsion based on numerical simulation. Wang [7] finds the suitable locations of the shaft bearings to keep the shaft alignment when ship hull deformations occur. Based on a simple numerical two-dimensional model of large ship propulsion-hull coupling system, Tian [8] analyzed the dynamic interactions of ship propulsion system and hull deformations. Zhou [9] uses FEM method to study the vibration response of the shaft under excitations. Based on these researches, this paper uses a numerical method to do some analyses on the vibration of propulsion system subjected to ship hull deformations.

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2. Dynamic Model of Propulsion System

As is known that the ship is a complex construction, lots of components except bearings are assembled in the ship. However, in this paper, a simplified 2-dimensional model including bearings, shaft, propeller and hull is shown in Figure 1. From Figure 1, it can be seen that several bearings mounted on the ship hull connect to the shaft and at the end of the shaft is the propeller. When the sea waves strike the ship hull fiercely, deformations are generated in the ship hull. While the hull deformations have effect on the bearings, excitations could be delivered to the propulsion system through the bearings. As a result, vibrations of the shaft will be more intensive than before. When the vibrations exceed the regular value, misalignment or more damage will occur on the propulsion system [10]. On the contrary, forces from propulsion's violent vibrations and inertia movements could feedback to the hull through the bearings. The hull would receive more forces besides the forces from the sea waves.

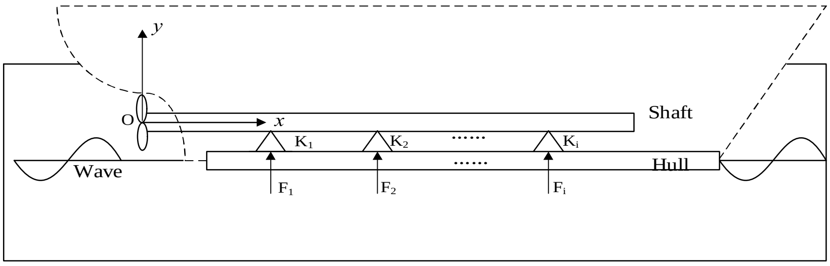


Figure 1. A 2-dimensional integrated ship propulsion system

2.1. Govern Equations of the Propulsion

As the interactions between the hull deformation and the shaft propulsion have great effects on the bending direction where interactions along the axial direction are ignored in this 2-dimensional system. This paper focuses on the bending direction of 2-dimensional circumatance. This coupled system is considered as a linear system in which the motion of the propulsion unit follows the beam / shaft theory.The propulsion system is an uniform flexible beam of length L_S diameter D_s, mass density per unit length $\circ_s$ assembled on the ship hull through several bearings . The shaft beam is divided into $(n+1)$ segments with lengths L_{s1} L_{s2} $\cdots$ L_{sn+1} respectively with the bending stiffness $E_s I_{s1}$, $E_s I_{s2} \ldots E_s I_{sn}$ by n support bearings and the stiffness of these bearings are K_1, $K_2 \ldots K_n$ respectively[11]. The propeller of the propulsion system is mounted at the end of the shaft which the mass is m_p and the rotational inertia is j_p.

The propulsion bending dynamic equation is established as follows:

$$E_s I_{si} \frac{\partial^4 U_i(x,t)}{\partial x^4} + \circ_s \frac{\partial^2 U_i(x,t)}{\partial t^2} = F, i = 1, 2, ..., n+1 \quad (1)$$

Where t is time and F is forces.

In order to solve the equation, the free-free movement is considered where F is regarded as 0. Based on reference [12], the displacement of the propulsion is obtained.

$$U_i(x,t) = \Phi_i(x) Y_i(t) \quad (2)$$

Where $Y_i(t)$ is generalized coordinate and $\Phi_i(x)$ is the mode function.

It can be known that the displacement amplitude of the shaft is $Y_i(t)$ and the shape of the shaft is $\Phi_i(x)$. As F is zero in equation (1) for free vibration, when putting Equation (2) into (1), (1) can be induced as follows:

$$\Phi_i^{(4)}(x)Y(t)+\frac{{}^{\circ}{}_{s}}{E_s I_{si}}\Phi_i(x)\ddot{Y}(t)=0 \tag{3}$$

Equation (3) can be defined as follows:

$$\frac{\Phi_i^{(4)}(x)}{\Phi_i(x)}=-\frac{{}^{\circ}{}_{s}}{E_s I_{si}}\frac{\ddot{Y}(t)}{Y(t)}=a_i^4 \tag{4}$$

Where $a_i^4=\frac{\mu_s^2\,{}^{\circ}{}_{s}}{E_s I_{si}}$ μ_s is the natural frequency. From reference [9], the mode function $\Phi_i(x)$ can be defined as follows:

$$\Phi_i(x)=A_i\cos a_{si}x+B_i\sin a_{si}x+C_i\cosh a_{si}x+D_i\sinh a_{si}x \tag{5}$$

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

2.2. Boundary Conditions and Hull Deformation Excitations

The compatibility conditions enforce continuities of the displacement field, the slope, the bending moment and the shearing force, respectively, across each support and can be expressed as follows:

$$U_i(X_i^L,t)=U_{i+1}(X_i^R,t) \tag{6a}$$

$$\frac{\partial U_i(X_i^L,t)}{\partial x}=\frac{\partial U_{i+1}(X_i^R,t)}{\partial x} \tag{6b}$$

$$E_s I_{si}\frac{\partial^2 U_i(X_i^L,t)}{\partial x^2}=E_s I_{si}\frac{\partial^2 U_{i+1}(X_i^R,t)}{\partial x^2} \tag{6c}$$

$$E_s I_{si}\frac{\partial^3 U_i(X_i^L,t)}{\partial x^3}=E_s I_{si}\frac{\partial^3 U_{i+1}(X_i^R,t)}{\partial x^3}+K_i U_{i+1}(X_i^R,t) \tag{6d}$$

Where X_i^L is the left section location of the shaft on the *ith* bearing and X_i^R is the right section location.

As the propulsion is on the free-free condition, the boundary of the propulsion is as follows:

$$E_s I_{s1}\frac{\partial^2 U_1(0,t)}{\partial x^2}+\mu^2\frac{\partial U_1(0,t)}{\partial x}j_1=0 \tag{7a}$$

$$E_s I_{s1}\frac{\partial^3 U_1(0,t)}{\partial x^3}+\mu^2 U_1(0,t)m_1=0 \tag{7b}$$

$$E_s I_{sn+1} \frac{\partial^2 U_{n+1}(L,t)}{\partial x^2} = 0 \tag{7c}$$

$$E_s I_{sn+1} \frac{\partial^3 U_{n+1}(L,t)}{\partial x^3} = 0 \tag{7d}$$

Where the propeller is considered as a mass at the end of the shaft.

Substituting Equation (5) to compatibility condition Equations (6a), (6b), (6c) and (6d) and boundary condition Equations (7a), (7b), (7c) and (7d), a matrix equation is obtained as follows.

$$\begin{bmatrix} [B_1] & & & & & & \\ [C_1(X_1^L)] & -[C_2(X_1^R)] & & & & & \\ & [C_2(X_2^L)] & -[C_3(X_2^R)] & \ddots & & & \\ & & & [C_{n-2}(X_{n-1}^L)] & -[C_{n-1}(X_{n-1}^R)] & & \\ & & & & [C_{n-1}(X_n^L)] & -[C_n(X_n^R)] & \\ & & & & & [B_n] & \end{bmatrix} \begin{bmatrix} P_b \\ P_1 \\ P_2 \\ \vdots \\ P_n \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \vdots \\ 0 \\ 0 \\ 0 \end{bmatrix}_{n+1} \tag{8}$$

Where $[P_b \quad P_1 \quad P_2 \quad \cdots \quad P_n]^T = [A_1 \quad B_1 \quad C_1 \quad D_1 \quad \cdots \quad A_{n+1} \quad B_{n+1} \quad C_{n+1} \quad D_{n+1}]^T$ is the coefficient matrix. $[C_i]_{4\times 4} = \begin{bmatrix} U_{i,1} & \cdots & U_{i,4} \\ U'_{i,1} & \cdots & U'_{i,4} \\ U''_{i,1} & \cdots & U''_{i,4} \\ U'''_{i,1} & \cdots & U'''_{i,4} \end{bmatrix}$ $(i=1,...,n)$ is the matrix form of the compatibility conditions. $[B_i]_{2\times 4} = \begin{bmatrix} U''_{i,1} & \cdots & U''_{i,4} \\ U'''_{i,1} & \cdots & U'''_{i,4} \end{bmatrix}$ $(i=1,...,n)$ is the matrix form of the boundary conditions.

For the propulsion system, hull deformations are acted on the support bearings. As a result, Dirac delta function was introduced as the hull deformation excitations F_i at the support bearing K_i.

$$F_i = \tilde{F}_i ¢ \left(x - X_i\right) Y\left(t\right) \tag{9}$$

Where $\tilde{F}_i$ is the amplitude of the force, X_i is the location of the bearing in the coordinate system *x-o-y*. The compatibility conditions (6d) is rewritten as follows:

$$E_s I_{si} \frac{\partial^3 U_i(X_i^L,t)}{\partial x^3} - EI_i \frac{\partial^3 U_{i+1}(X_i^R,t)}{\partial x^3} - S_i U_{i+1}(X_i^R,t) = \tilde{F}_i \tag{10}$$

Let $[F_{hi}]_{4\times 1} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \tilde{F}_i \end{bmatrix}$ $[F_{hb}]_{2\times 1} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$ the matrix equation of the propulsion taking the hull deformations into account is derived as follows:

$$\begin{bmatrix} [B_1] & & & & & & \\ [C_1(X_1^L)] & -[C_2(X_1^R)] & & & & & \\ & [C_2(X_2^L)] & -[C_3(X_2^R)] & \ddots & & & \\ & & & [C_{n-2}(X_{n-1}^L)] & -[C_{n-1}(X_{n-1}^R)] & & \\ & & & & [C_{n-1}(X_n^L)] & -[C_n(X_n^R)] & \\ & & & & & [B_n] & \end{bmatrix} \begin{bmatrix} P_b \\ P_1 \\ P_2 \\ \vdots \\ P_n \end{bmatrix} = \begin{bmatrix} [F_{hb}] \\ [F_{h1}] \\ [F_{h2}] \\ \vdots \\ [F_{hn-1}] \\ [F_{hb}] \end{bmatrix} \qquad (11)$$

3. Vibration Analysis of the Propulsion System

Based on the equations developed in section 2, a numerical analysis of the vibrations of the propulsion system is carried on.

3.1. Model Verification

Based on an example of a kind large container vessel the comparison of modal calculation resultsbetween the theory model and the FEA model are shown the correctness of the propulsion model containing the propeller.

Table 1 and Figure 2 show some key parameters of the propulsion and supporting bearings. The propulsion are redistrict into several part for the convenience to calculate.

Table 1. Parameters of the shaft and bearings

Name	Unit	Value
Overall lengthL_s	m	**46.588**
Diameter of shaft D_s	m	**0.795**
Support bearing K_1	N/m	1×10^9
Support bearing K_2	N/m	1×10^9
Support bearing K_3	N/m	1×10^9
Support bearing K_4	N/m	1×10^9

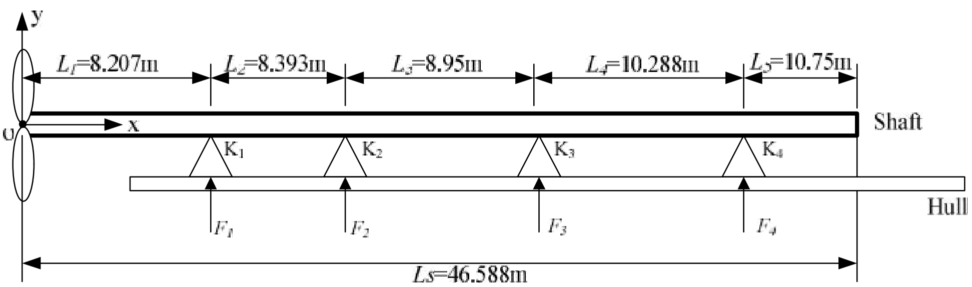


Figure 2. Key parameters of the propulsion system

From Figure 3, it shows that the analytical natural frequencies is quiet coincident with the FEA natural frequencies. Based on the analytical method (Matlab) and the FEA method (Ansys), when excitations apply on the propulsion shaft, the vibration responses which are obtained on the engine port and the propeller port have a good coincidence (Figure 4 and Figure 5).

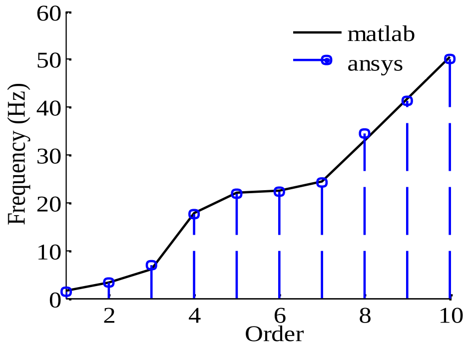


Figure 3.Comparison of the analytical (Matlab) and numerical (Ansys) results for shaft's free frequencies adding supports

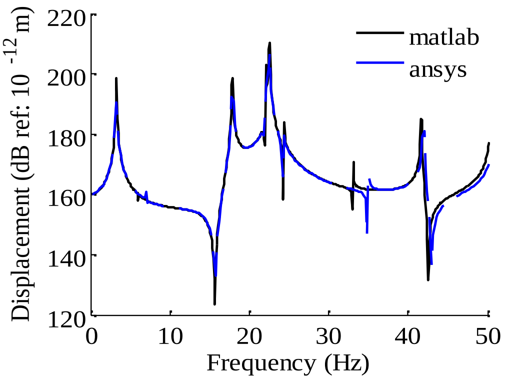


Figure 4. The response of vibration on the main engine end

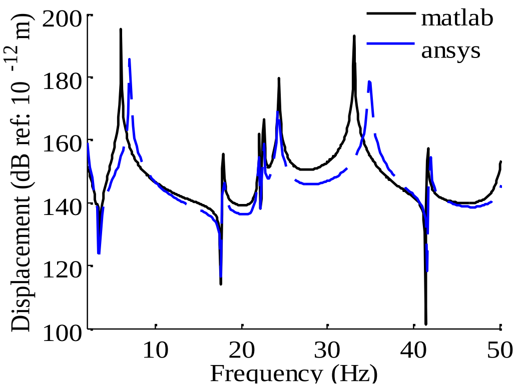


Figure 5. The response of vibration on the propeller end

3.2. Vibration Behavior of the Propulsion Shaft with Different Hull Deformations

Vessels always encounter various sea wave conditions during their operations. Table 2 shows the sea wave conditions which are divided by State Oceanic Administration People's Republic of China [13]. In order to investigate the vibration behavior for the propulsion shaft during its severe operation, this paper chooses the band 'Very Rough' as the sea wave condition.

Table 2. Sea wave conditions division [13]

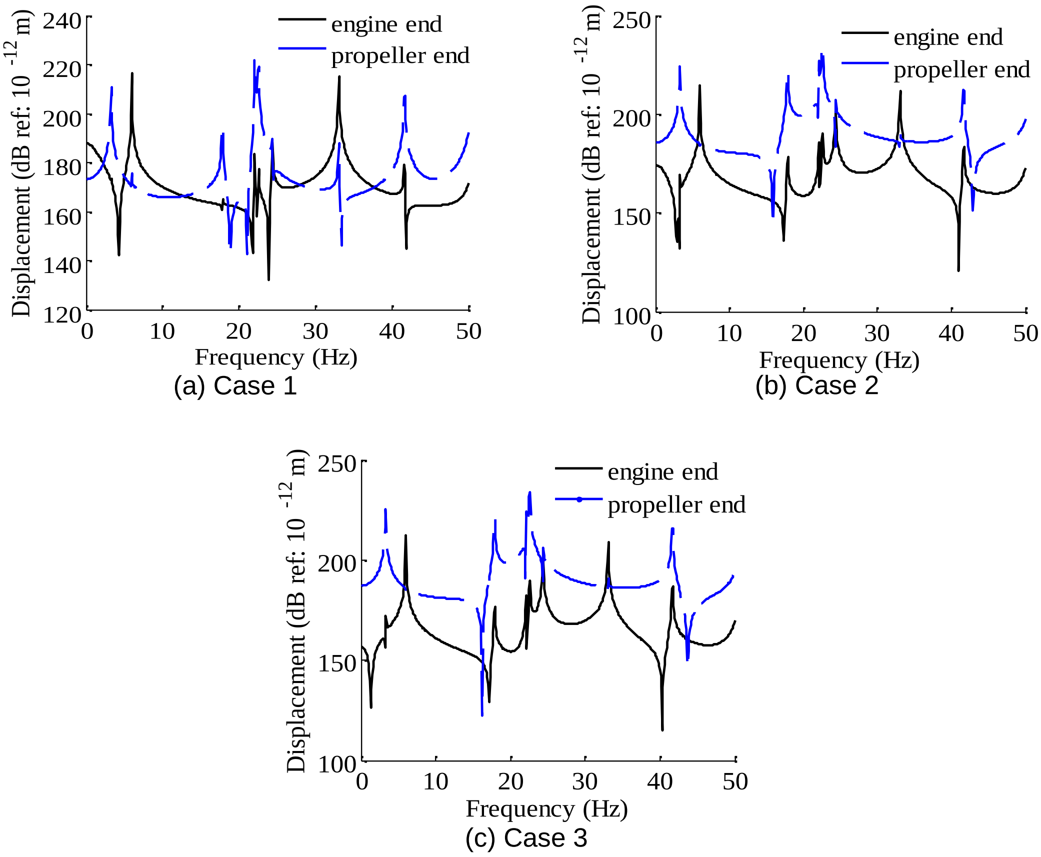
Band of Sea Wave	Significant Wave Height (m)	Band of Sea Wave	Significant Wave Height (m)
Calm-Glassy	Hs<0.1	Very Rough	4.0≤Hs<6.0
Calm-Rippled	0.1≤Hs<0.5	High	6.0≤Hs<9.0
Smooth-Wavelet	0.5≤Hs<1.25	Very High	9.0≤Hs<14.0
Moderate	1.25≤Hs<2.5	Phenomenal	Hs≥14.0
Rough	2.5≤Hs<4.0	Hs: significant wave height	

As a result, some critical cases for the vessel 8530TEU operating in the sea waves are given in Table 3. Based on the Hooke's law, hull deformations on the shaft bearings could be transformed into forces. Therefore, in Table 3, the deformations are described as the forces on each bearing.

Table 3. Parameters of some typical cases

Case	Wave direction (°)	Wave height (m)	Wave frequency (rad/s)	Forces on bearing K_1 (N)	Forces on bearing K_2 (N)	Forces on bearing K_3 (N)	Forces on bearing K_4 (N)
Case 1	90	5	1	1.02×10^6	-1.96×10^5	-0.99×10^4	-1.45×10^5
Case 2	180	5	1	-9.86×10^5	-1.38×10^6	-1.41×10^6	-1.62×10^6
Case 3	0	5	1	-7.79×10^5	-1.33×10^6	-1.09×10^6	-1.55×10^6

According to the parameters supplied in Table 3, comparisons of dynamic responses for the 8530TEU shaft between the engine end and propeller end are illustrated in Figure 6(a)-(c) respectively. In generally, it can be observed that low excitation frequencies have great effect on the propeller prior to engine. To be specific, Figure 6(a) shows that when the excitation frequency is around 23 Hz, the largest displacement of the propeller occurs which means the stern bearing has to endure severe operating conditions from the propeller in case 1. While the excitation frequencies are around 8 Hz and 32 Hz, the maximum displacements of the engine appearwhich means misalignment of the shaft might occur in the engine port.Similarly, Figure 6(b) and Figure 6-(c) shows that when the excitation frequency is around 23 Hz, the largest displacement of the propeller occurs while the most dangerous frequencies for the engine are around 8 Hz and 32 Hz.

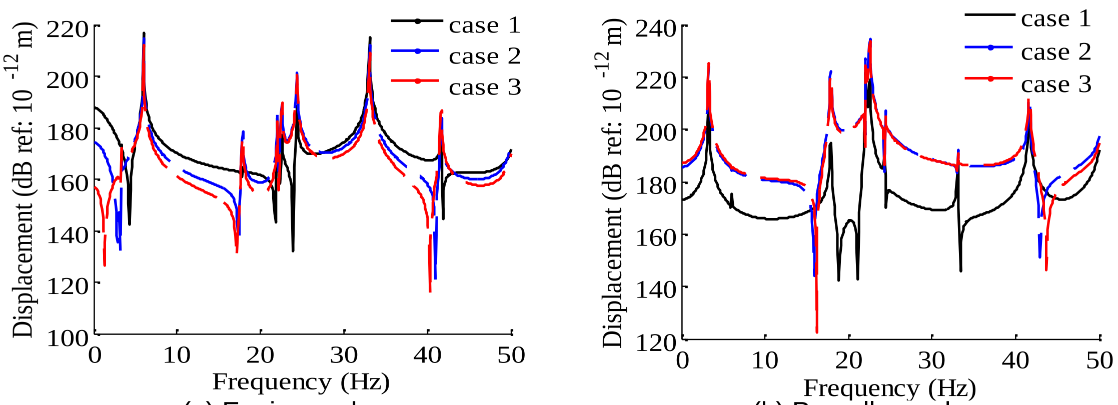


(a) Case 1 (b) Case 2

(c) Case 3

Figure 6. Dynamic responses of shaft engine end and propeller end

Figure 7(a)-(b) illustrate the comparison of the dynamic responses among the three cases on the engine end and propeller end respectively. In generally, it can be seen that the excitation frequencies have not obvious variations when the maximum displacements occur in the both engine end and propeller end under these cases. However, the maximum amplitudes of shaft in engine end and propeller end are different among these three cases. Specifically, in Figure 7(a), the maximum amplitude of the engine end vibration in case 1 is larger than the other two cases. As a result, compared to the other two cases, once the vibration exceed to the allowance when the ship operates in case 1 condition, the shaft could easily receive damages. However, in Figure 7(b), the shaft at propeller end has huge vibrations during its operation in case 3 condition. Monitorsin these two ports could be enhanced to guarantee the reliability of the system when the large vessel 8530TEU sails in the two cases.



(a) Engine end (b) Propeller end

Figure 7. Dynamic responses of shaft for Case 1, 2&3

4. Conclusion

An analytical method to predict a 2D propulsion shaft with propeller under hull deformation excitations at supports is presented in this paper. A model of a typical ship 8530TEU operating in some sea wave conditions is built as an example. Hull deformations are regarded as the external forces applying on the supporting bearings. The vibration solution of a shaft is gained by the bending equation of motion and the boundary/continuity condition equations. The accuracy of the analytical method for the shaft model is confirmed by comparing with the FEA method. Based on this analytical model, the vibration characteristics of the propulsion operating in three typical sea wave conditions are discussed. Conclusions can be drawn as bellows.

(1) Low excitation frequencies have great effect on the propeller prior to engine.

(2) When the vessels operates in the 0°wave direction conditions, the shaft on the propeller end could easily receive more vibrations rather than on the engine end.

(3) When the vessels operates in the 90°wave direction conditions, the shaft on the engine end could easily receive more vibrations rather than on the propeller end.

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