Study on the effect of alignment style on shafting-shell coupled system radiated noise caused by propeller force

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ABSTRACT

For the underwater structure, the propeller excitation force play an important role which can induce structure vibration and radiated noise in low frequency. Because of the arrangement of shafting, it is difficult to control the vibration of shafting-shell system. The effect of shafting alignment style on shafting-shell coupled system radiated noise is studied in this paper. The 2D Reynolds equation is calculated by finite difference method. The fluid film pressure distribution of each bearing is calculated. Load increment method is used to calculate the fluid film stiffness and finite element method is adopted to calculate the bearing support stiffness. On the basis of stiffness value, the propeller, shafting and shell coupled system is built by finite element method. The characteristic of shell underwater radiated noise caused by propeller force is studied. The alignment style of shafting is changed. The effect of alignment style on shafting-shell coupled system radiated pressure is calculated.

Keywords: Fluid film stiffness, Underwater structure, Finite element method, Alignment style
I-INCE Classification of Subjects Number(s): 54.3

1. INTRODUCTION

For the propeller running in the non-uniform wake fluid field, the periods lift and resistance are formed on each blade surface. The lift and resistance can be decomposed into three forces and three torques which are decided by the number of blades and the shafting rotation speed. The forces and torques will induce the lateral, longitudinal vibration of shafting. In the following, the shafting longitudinal and lateral vibration energy transfers from shafting to bearing supporting, thereby to the structure which will cause underwater vehicle radiate strong noise.

The shafting and simply support plate coupled system is adopted as analysis model in paper [1]. As the main path of longitudinal propeller force, the thrust bearing stiffness is obtained by experimental method. The vibration characteristic of plate are calculated. With the development of simulation method, the computational fluid dynamics is used to calculate the unsteady hydrodynamics of blade forces[2]. The five blades propeller and the SUBOFF submarine are adopted to be the analysis model. The single and total blade forces are calculated. On the basis of propeller force, the coupled vibration characteristic of propeller, shafting and structure is studied by the transfer matrix method and finite element method[3]-[4]. In addition, Some methods which can reduce structure vibration are proposed. The dynamic absorber arrange at thrust bearing side is used to reduce the exciting force. The vibration amplitude at longitudinal natural frequency is decreased[5]-[7].

However, the shafting-shell system vibration caused by lateral propeller force is less studied. Comparing with the single longitudinal propeller force transmission path, the lateral propeller force mainly transfer through every bearings to structure. It is difficult to control such structure vibration. In this paper, the shafting alignment is used to try to decrease the shell vibration and radiated noise. The shaft alignment is widely used in the modern ship because it can release the heavy load of some bearings[8]. Furthermore, it can also change the axis line shape of shafting and the stiffness of bearing which will change the vibration performance of shafting. In such condition, the supporting bearings play an important role because the bearings are the only path for propeller excitation energy transfer from shafting to structure. The transmission power of each bearing is decided by bearing stiffness

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which can be divided into the bearing support stiffness and the liquid film stiffness. In this paper, the relation between shaft alignment style and structure radiated noise characteristic is studied.

2. ANALYSIS OF BEARING SIFFNESS

2.1 Mathematical formulation

The 2D Reynolds equation is used in this paper to describe the finite width journal bearing sunder steady-state conditions which is as follows:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6U\frac{\partial h}{\partial x} \tag{1}$$

In equation (1), η means the dynamic viscosity of lubricating oil, P is the film pressure, x is the coordinate of the circumferential direction, z is the coordinate of the axial direction, U means the tangential velocity of the journal surface and h is the film thickness.

A half step finite difference method is used to solve the equation above. The grid distribution in the axial and radial direction is shown in Fig 1.

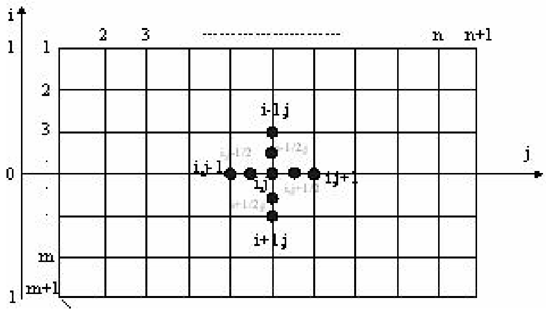


Figure 1 – Grid distribution

The liquid film pressure at each point can be calculated by the following equation. In addition, the Reynolds boundary condition is applied in this paper.

$$P_{i,j}=\frac{A_{i,j}P_{i,j-1}+B_{i,j}P_{i,j+1}+C_{i,j}P_{i-1,j}+D_{i,j}P_{i+1,j}-F_{i,j}}{E_{i,j}} \tag{2}$$

The correlation coefficients A_{ij}-F_{ij} are shown as follows. H means the film thickness.

$$\begin{cases} A_{i,j}=H^3{}_{i,j-1/2} \\ B_{i,j}=H^3{}_{i,j+1/2} \\ C_{i,j}=H^3{}_{i-1/2,j}\left(\dfrac{D}{L}\dfrac{\Delta\varphi}{\Delta\lambda}\right)^2 \\ D_{i,j}=H^3{}_{i+1/2,j}\left(\dfrac{D}{L}\dfrac{\Delta\varphi}{\Delta\lambda}\right)^2 \\ E_{i,j}=A_{i,j}+B_{i,j}+C_{i,j}+D_{i,j} \\ F_{i,j}=\Delta\varphi(H_{i,j+1/2}-H_{i,j-1/2}) \end{cases} \tag{3}$$

The bearing capacity F_x and Fy which will be used to calculate the film stiffness can be calculated by multiplying film pressure and grid area.

$$\begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix}=\begin{bmatrix} \dfrac{\partial f_x}{\partial x} & \dfrac{\partial f_x}{\partial y} \\ \dfrac{\partial f_y}{\partial x} & \dfrac{\partial f_y}{\partial y} \end{bmatrix} \tag{4}$$

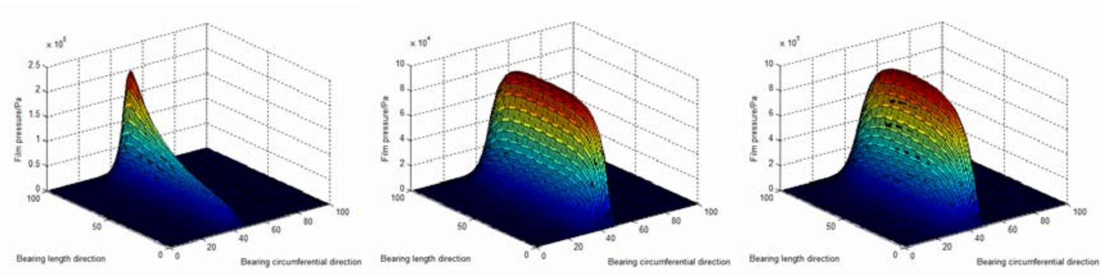
The liquid film stiffness can be achieved by small perturbation method, and The stiffness matrix is as follows, then the calculation process is written to program in the MATLAB software.

2.2 Bearing stiffness discussion

Some preliminary results are shown in this section. A typical marine propulsion shafting bearing is used for calculation. The bearing parameters are shown in Table 1. As discussed previously, the pressure distribution results are shown in Figure 2.

Table 1 – Design parameters for the reference bearing

	bearing load	bearing length	diameter	rotational speed	radial clearance	lubricant viscosity
stern bearing	814 N	300 mm	105 mm	360 r/min	0.4 mm	0.001 Pa.s
middle bearing	519 N	200 mm	105 mm	360 r/min	0.4 mm	0.001 Pa.s
thrust bearing	3978 N	150 mm	95 mm	360 r/min	0.4 mm	0.02 Pa.s



(a) stern bearing (b) middle bearing (c) thrust bearing

Figure 2 –The pressure distribution of bearings

Figure 2 shows that the film pressure distribution of middle bearing and thrust bearing is nearly symmetry. However, the shafting line is curved by gravity and the tilt is formed in each bearing especially in stern bearing. The maximum value can be found near the stern side. Furthermore, the viscosity of lubricating oil play an important role in the pressure distribution.

In the following step, the shafting alignment style is changed. Three cases are listed in this section. The first case is original model. The second one is reducing the height of stern bearing 1mm and the third one is 2mm. The bearing film stiffness in each case is obtained by load increment method. The bearing load is tested with jack-lifting, which is shown in Table 2. The bearing stiffness are tested in jack-lifting process, which is 1e8N/m. The Equivalent stiffness of each bearing can be seen in Table 3.

Table 2– The bearing load N

	Thrust bearing	Middle bearing	Stern bearing
Case 1	814	519	3978
Case 2	680.6	781.9	3849
Case 3	547.2	1044.5	3719.9

Table 3 – The stiffness of bearing N/m

	Thrust bearing	Middle bearing	Stern bearing
Case 1	1.4137e07	4.6992e07	1.8829e09
Case 2	1.0142e07	1.0069e08	1.8101e09
Case 3	6.8263e06	1.7300e08	1.6669e09

Table 2 shows that the load of stern bearing is decreased with the height of stern bearing reduced. At the same time, the load of middle bearing is increased and thrust bearing load is increase. The load of bearings are averaged.

Table 3 shows that the stiffness of bearing is changed too. The trend is similar with bearing load.

The middle bearing stiffness decreased. The stern and thrust bearing stiffnesses are increased, which will change the natural frequency of shafting.

3. NUMBER RESULTS AND DISCUSSIONS

3.1 Model description

The shafting-shell coupled model in this paper is composed by four parts, which are a conical shell, a cylindrical shell, a half sphere shell and the shafting. The shafting is connected with the shell by stern bearing, middle bearing and thrust bearing.

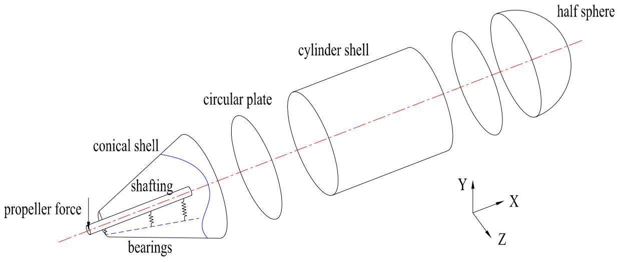


Figure 3 –The calculation model

The structure is assumed to have a Youngs's modulus of E=2.06E11N/m2, Poisson's ratio $\nu = 0.3$. The sound speed in the surrounding fluid is assumed to be 1500m/s. The density of fluid is assumed to be 1000 kg/m^3.

The shell is meshed into 9763 elements and 8364 nodes. Including the propeller and bearing, the shafting is meshed into 81 elements and 81 nodes. The propeller is predigested as a mass point. Each bearing supporting is assumed as stiffness element connected with the shell and the shafting. The unit lateral force is applied at the propeller side.

3.2 Modal Analysis of Shafting-shell system

Three cases are calculated in this section. The stiffness of bearing is changed respectively and the shafting natural frequencies are shown in table 4.

Table 4– The shafting natural frequency Hz

	Case1	Case2	Case3
1	45.3915	44.7598	43.4141
2	61.9817	66.0513	69.4169
3	98.2031	97.777	97.2012
4	160.959	162.193	163.994
5	249.603	248.462	248.324

(a) First order（45. 392Hz） (b) Second order（61. 982Hz）

Figure 4 –The modal shape of shafting

(a) First order（19.538Hz） (b) Second order（40.589Hz）

(a) First order（57. 482Hz）　　(b) Second order（85. 123Hz）

Figure 5 –The modal shape of shell

The effect of shafting alignment style on shafting natural frequency is complex. Each bearing stiffness is changed with the change of alignment style. Some bearing stiffness is increasing and some decreasing, which cause natural frequency of shafting irregular changed. The value mainly depends on the modal shape.

Comparing with the shafting bearing stiffness, the stiffness of shell structure is higher. The natural of shell has little changed in three cases. For the size and the complexity of shell, more local modal frequencies appear in low frequency. The front forth bending modal shapes of the shell are shown in figure 5.

3.3 underwater noise response of shell

Using the shell surface vibration velocity data at normal direction, the underwater radiated sound power and pressure can be calculated by direct boundary element method shown in figure 6.

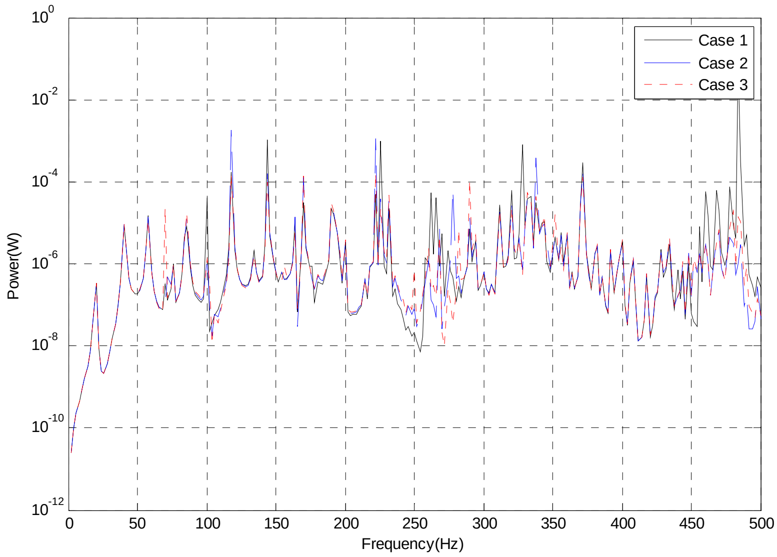


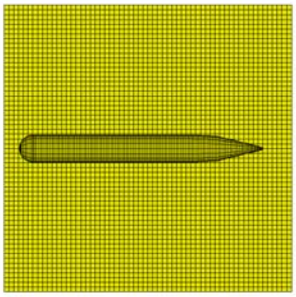
Figure 6 – The underwater radiated sound power of ship structure

As one system, the propeller excitation transfer through shafting, bearing to the shell, so the shell-shafting system response curve contains both the shafting and shell vibration natural frequencies. For the series connection of shell stiffness, the equivalent stiffness of bearing is lower than that of single bearing which decreases the natural frequencies of shafting. However, the additional stiffness of shafting has little effect on the shell natural frequencies.

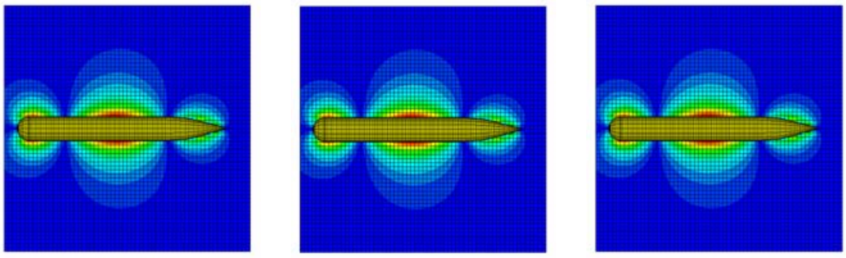
The effect of alignment style on shell radiated noise is relatively small. According to figure 6, there are three obvious formants in low frequency, which are 20 Hz, 40 Hz and 58Hz. These are the natural frequencies of shell and the modal shapes are shown in figure 5. The amplitude of each formant has little change.

The effect of alignment style on shell radiated noise is great near the shafting natural frequencies and the trend is complex. The first trend is the offset of formants such as 222Hz, 275 Hz、330Hz. Furthermore, the amplitude of formants are changing at some frequencies such as 99Hz, 120Hz, 146Hz and so on.

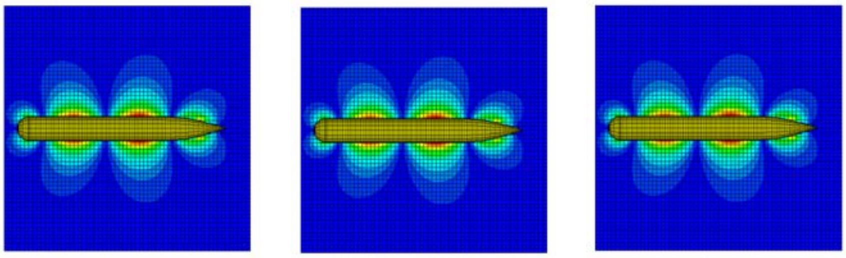
The underwater pressure distribution caused by alignment style is shown in figure 7. The pressure distribution in case1, case2 and case3 can be seen in left column, the middle column and the right column separately.



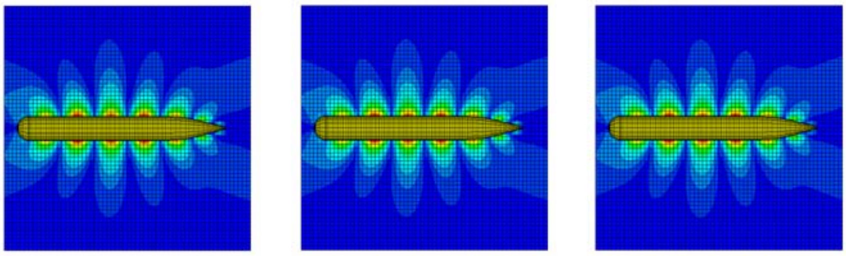
(a) Field point of underwater pressure distribution



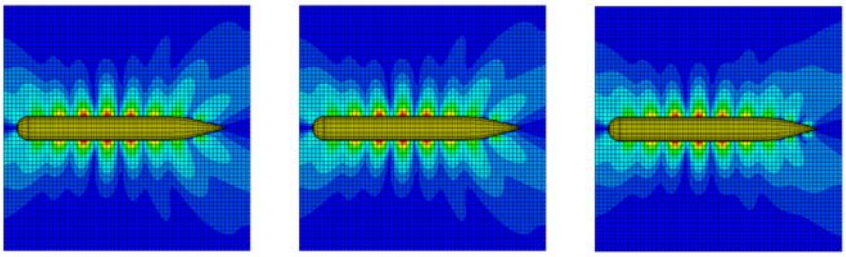
(b) Underwater radiated pressure distribution at 20Hz



(c) Underwater radiated pressure distribution at 58Hz



(d) Underwater radiated pressure distribution at 164Hz



(e) Underwater radiated pressure distribution at 258Hz

Figure 7 – The underwater radiated pressure distribution of ship structure

The pressure distribution of shell-shafting system is mainly decided by the modal shape of shell. The propeller lateral force transfers through the shafting, supporting bearing to the structure, and causes the bending vibration of shell. The pressure distribution is similar with the modal shape of shell. The figure 7 shows the trend of underwater pressure distribution clearly with the frequency increasing.

The effect of alignment style on the shell underwater pressure distribution is little. Similar with the trend of sound power, the pressure distributions are nearly same in three cases. The difference alignment style causes the bearing transfer forces amplitude and frequency changed. The degree of variation is decided by the modal shape of shafting. According to the modal shape of shafting, the effect of alignment style on the first two shafting modal frequency is little. Furthermore, the transfer force at stern bearing is not changed because the little change of bearing stiffness.

The effect of alignment style on the shell underwater pressure amplitude near the shafting natural frequency is great especially at some orders. Because of the modal shape of shafting, the transfer forces at middle bearing and thrust bearing are changed, increasing or decreasing. The amplitude of sound pressure is decided by the shafting modal shape and the whole sound pressure contribution of each bearing force.

4. CONCLUSIONS

The effect of alignment style on shell-shafting system underwater radiated noise caused by propeller exciting force is studied in this paper. The stiffness of supporting bearing is calculated by finite element method and 2D Reynolds equation. The shafting and shell natural vibration characteristic are analyzed. The structure underwater radiated pressure distribution and sound power are calculated.

1. The alignment style of shafting mainly influences the shafting line shape, the stiffness of oil film. The result shows that oil film stiffness is influenced by alignment style greatly, which is mainly influencing each bearing load. As a series system, the oil film stiffness makes the equivalent stiffness of bearing changed.

2. The change of shafting natural frequency is complex with the alignment style changing. At some orders, the natural frequency is increasing and some is decreasing. The trend of natural frequency is decided by modal shape and the location of bearing. However, the shell natural frequency is not changed.

3. The effect of alignment style on shell radiated sound power is great at the shafting natural frequencies and the trend is complex. The trend is decided by the shafting modal shape and each bearing stiffness variation. The effect of alignment style on the shell underwater pressure distribution is little, but the amplitude of pressure is changed greatly.

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REFERENCES

1. Pan J., Farag N.. Propeller induced structural vibration through the thrust bearing. Proc. of Acoustics 2002, Adelaide, Australia 2002. p.390-399 .
2. Yingsan Wei, Yongsheng Wang. Unsteady hydrodynamics of blade forces and acoustic response of a model scaled submarine excited by propeller's thrust and side-forces. Journal of Sound and Vibration. 2013;332:2038-2056
3. Sascha Merz, Nicole J. Kessissoglou.Structure and acoustic responses of a submarine hull due to propeller force. Journal of Sound and Vibration. 2009;325: 266-286.
4. Xu M B, Zhang W H. Vibration power flow input and transmission in a circular cylindrical shell filled with fluid. Journal of Sound and Vibration. 2000;234(3):387-403.
5. Paul G. Dylejko, Nicole J. Kessissoglou. Optimization of a resonance changer to minimize the vibration transmission in marine vessels. Journal of Sound and Vibration. 2007;300: 101-116.
6. Sascha Merz, Nicole Kessissoglou. Minisation of the sound power radiated by a submarine through optimisation of its resonance changer. Journal of Sound and Vibration. 2010;239: 980-993.
7. Cao Yipeng, Zhang Wenping. A Study on Reducing the Underwater Structure Acoustic Radiation Caused by Longitudinal Vibration of Shafting. Shipbuilding of China. 2008;49(2):36-43.
8. Lech Murawski.Shaft line alignment analysis taking ship construction flexibility and deformations into consideration. Marine Structure. 2005;18:62-84.