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Experiment and Analysis of Correlation Between Large New Ships' Shafting Vibration and Stern Bearing Wear

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Abstract: Ship stern bearing wear is a significant problem often appeared during sea trials for large new ships which directly affects the subsequent link of ship construction thus the whole construction progress. And even worse it may cause the owners abandon the ships. Ship shafting vibration is the integrated embodiment of its working status and shafting performance which can reflect the dynamic response characteristics such as the main engine incentives, propeller incentives and the coupling incentives. It can also reflect the stern bearing wear probability. This study starts with the large new ships' shafting vibration experiment, by the shafting vibration experiment and analysis and researches on shafting vibration and its correlation with stern bearing wear, thereby provides checking and analysis method for stern bearing wear.

Key words: Shafting vibration, bearing wear, correlation experiment, tern bearing

INTRODUCTION

Ship shafting is one of an important part of ship power plant, transferring the power turned out by main engine to the propeller and the axial thrust of propeller to the hull and thus causing the ship sailing. Ship shafting vibration is an objective common phenomenon. Due to the changeable operation condition of the main engine, varying vibration can be produced at different stages. When the shafting vibration natural frequency is identical with its excitation frequency, resonance can be produced. Large ships, equipped with high-power main engine and high-thrust propeller, the incentive function of the shafting are increased and accordingly the shafting vibration characteristics is varying which eventually will lead to abnormal wear of stern bearing and the heating problem. Through the investigation and research of several large Chinese shipyards, we found that in the process of large ship building, the stern bearing abnormal wear in varying degrees happened in the major shipyards in China, the highest wear probability is reached to 35%. Large new ships' stern bearing wear is closely related to the axis of the ship shafting. The general research method is starting from shafting alignment calculation and the alignment techniques. It is a new method to study the stern bearing wear problem by starting with the analysis of the propeller incentive, main engine incentive and coupling incentive, thus to create the correlation between the ship shafting vibration characteristic and stern bearing wear.

EXPERIMENT PLAN

The shafting system of a 64000 DWT bulk carrier is the experimental object. Three sensors are provided in the shafting. Sampling shafting torsional vibration, horizontal and vertical vibration is respectively by using ANZT6 and ZD-IV method, as shown in Fig. 1 (Chen, 2002; Du and Zhou, 1996; Yan *et al*., 2009; Zhang and Tang, 1997). In order to guarantee the test data consistency, the three shafting vibration signals are taken at the same time during the testing process. Vibration data acquisition process is from the lowest stable speed to the rated speed, i.e. the 26-89 r min^{-1}. The speed interval is no more than 5 r min^{-1}. The data should be taken 2 times when each working condition is stable. The ship parameters related to the shafting vibration is as follow: the main engine is a two-stroke low-speed diesel engine its output power is 8050 kW, the rated speed is 89 r min^{-1}, the number of cylinders is 5; Propeller blade number is 5, 6.7 m in diameter.

TORSIONAL VIBRATION ANALYSIS

The obtained vibration test data is singularity judged and deleted by using ANZT6 shafting vibration test system and spectrum analysis technology, the maximum amplitude and the corresponding shaft speed of various orders of shafting torsional vibration are got, as shown in Table 1 (Du and Zhou, 1996). From the table it is can be seen that the torsional vibration amplitude of 1order and

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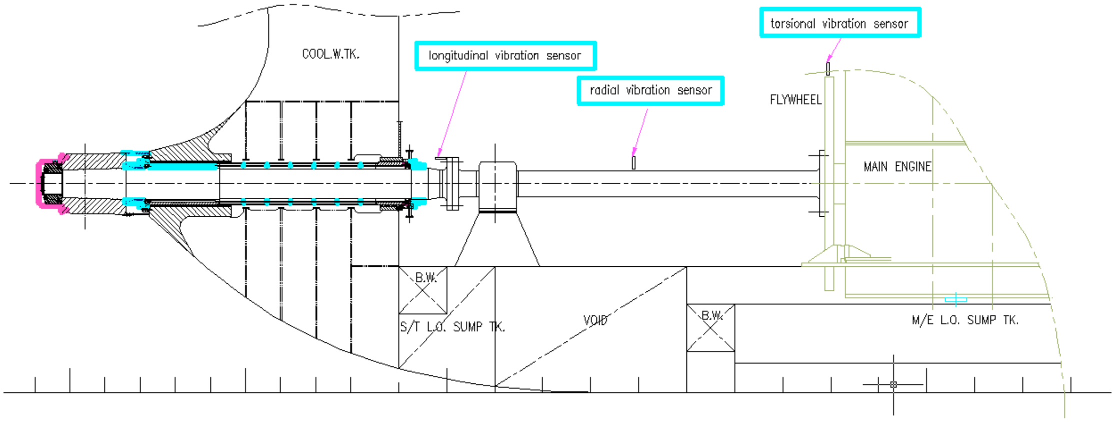


Fig. 1: Shafting and sensor position

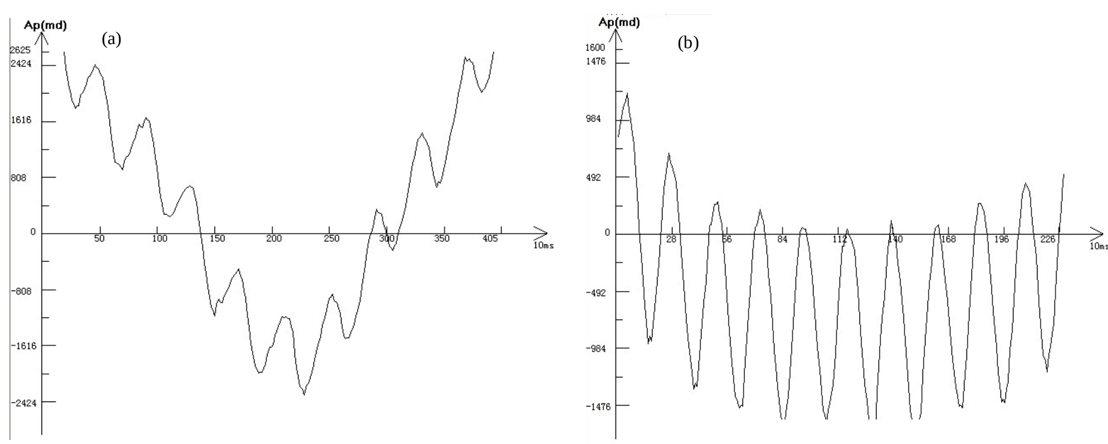


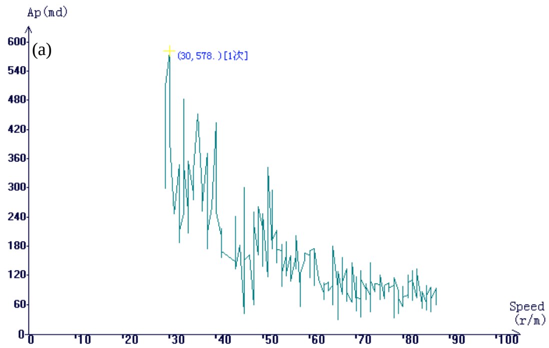
Fig. 2(a-b): Harmonic wave of resonance speed, (a) 1st order and frequency response of 0.5Hz, (b) 5th order and frequency response of 0.9Hz

Table 1: Maximum amplitude and speed of 1-10 order (torsional vibration)

Order	1	2	3	4	5	6	7	8	9	10
Amplitude (md)	578	140	77	99	855	75	32	24	19	75
Speed (r min^{-1})	30	29	52	51	53	51	51	49	51	29

5 order is significantly greater than other orders and accordingly the stern bearing wear or failure probability is far greater than other order Shafting torsional vibration spectrum characteristics of 1st order and 5th order are shown in Fig. 2 and 3. Fig. 2 is the diagram of harmonic wave of resonance speed. Fig. 3 is the diagram of harmonic-speed curve. ANZT6 is associated with the fundamental frequency of the torsional vibration signal and the shaft rotating speed (Fig. 2a and Fig. 3a, the fundamental frequency is 0.5 Hz and Fig. 2b and Fig. 3b, the fundamental frequency is 0.9 Hz), the rotating speed is higher, the fundamental frequency is greater and the acquisition time is shorter, within 2 cycles. Fig. 2 shows the periodic variation characteristics of main engine's cylinder incentives, propeller hydrodynamic incentives and their coupling effect in different cylinder ignition working condition. It can be seen that there is a peak value of the cylinder incentive. The peak number in a cycle is consistent with the main engine cylinder number. When main engine is running at low speed, the peak difference between each cylinder is greater. When running at medium speed, the peak difference is significantly narrowed. It can also reflect that along with main engine increasing speed, each cylinder combustion situation is stabilized and power difference is narrowed. This characteristic is consistent with the law of marine diesel engine combustion.

In shafting torsional vibration analysis, the resonance speed of maximum torsional vibration amplitude is taken. The value of maximum resonance speed is 53 r min^{-1}



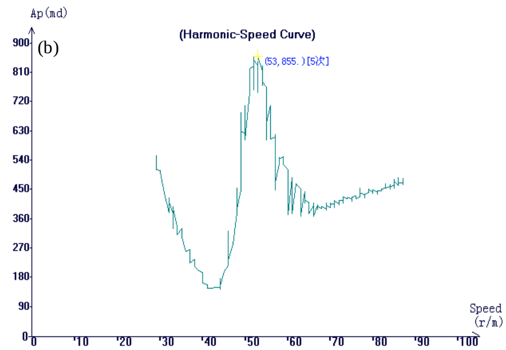


Fig. 3(a-b): Harmonic-speed curve, (a) 1st order, (b) 5th order

Table 2: Resonance speed and frequency

Mode	Resonance order	Resonance speed ($r\ min^{-1}$)	Frequency measured ($l\ min^{-1}$)	Frequency calculated ($l\ min^{-1}$)	Relative error (%)	Angular deflection (deg)
1	5	53	265.0	260.4	1.77	0.855

Table 3: Torsion stress

Item	Measurements value	Permitted value (τ_c/τ_t)
Crankshaft stress (N/mm^2)	19.68	22.81
Intermediate shaft stress (N/mm^2)	109.69	60.60/103.02
Propeller shaft stress(N/mm^2)	61.81	36.5/75.15

Table 4: Maximum amplitude and shaft speed at various orders (radial vibration)

Order	1	2	3	4	5	6	7	8	9	10
Amplitude (mm)	0.130	0.052	0.038	0.099	0.085	0.120	0.015	0.020	0.019	0.015
Speed ($r\ min^{-1}$)	73.630	51.410	87.930	51.560	84.030	50.050	51.410	48.040	26.020	42.040

according to Fig.3 (b) and the shafting vibration belong to 1 node and 5th order and the measured frequency is 265 min^{-1}. In theory the calculated frequency is 260.4 min^{-1} and the relative error between measured frequency and calculated frequency is 1.77% which is less than 5%. So amplitude value of torsional vibration at maximum resonance speed can be used to calculate the amplitude value, stress and torque of the other shafting points by the uncontrolled vibration of table Holzer, as shown Table 2.

The torsion stress of each endured section can be calculated by uncontrolled vibration of table Holzer. Their results are shown in Table 3. The measured stress value of the crankshafts is less than permitted value. And the measured stress value of the propeller shaft is higher than the permitted value for continuous running and less than maximum value (Permitted value for transient running). If the measured stress value of the intermediate shaft is higher than maximum value, the shaft system damping treatment is required and torsional vibration should be recalculated and check again.

RADIAL VIBRATION ANALYSIS

Table 4 (Chen, 2002; Liu *et al*., 2013) shows the maximum shafting transversal amplitude at various orders and its corresponding shaft rotation speed which is obtained by using ZD-IV shafting vibration testing and analysis system, the singularity judgment and deletion method. It can be concluded from Table 4 that the transverse vibration amplitude of 1st order and 6th order is greater than the other order, their corresponding working condition has become the special order for analyzing the stern bearing wear conditions.

Shafting transverse vibration spectrum characteristics of 1order and 6th order is shown in Fig. 4 and 5. Fig. 4 is the radial vibration analysis curves at 1 order. Fig. 5 is the diagram of radial vibration analysis curves at 6th order. According to design drawings, the ship stern bearing clearance is 0.8-1.0 mm. Referring shafting transverse vibration calculation, the vibration amplitude of testing position is accordance with stern bearing position. Thus, by comparison it can be drawn that the ship transverse vibration impact on the stern bearing wear can be ignored. By comparison Fig. 4a and 5a, the amplitude of shafting transverse vibration at 1st order is increasing as the shaft rotating speeding and at 6th order there is a maximum value within the main engine revolution range. From Fig. 4b and Fig. 5b it can be seen that the main engine incentives and propeller incentives can be ignored when the main engine is running at low speed, but as the speed increases, the host and propeller coupling excitation

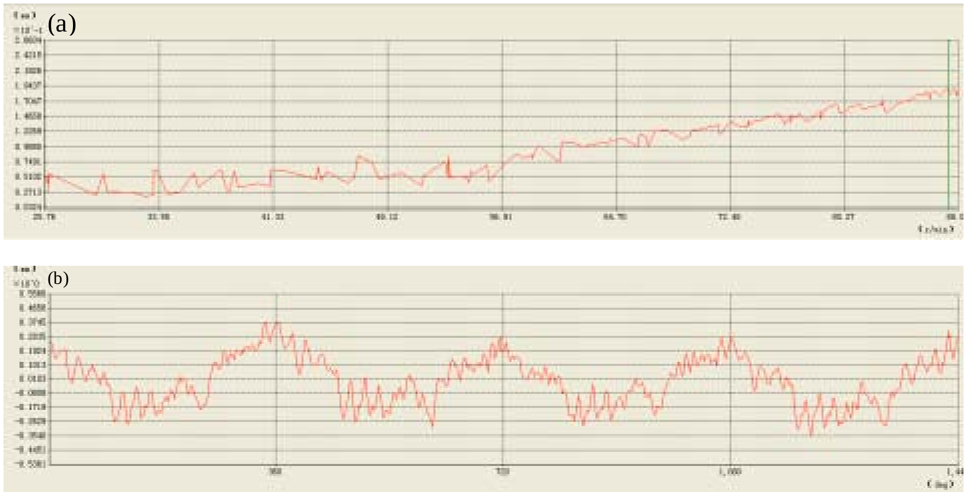


Fig. 4(a-b): Diagram of radial vibration analysis curves (1st order), (a) amplitude-speed curve, (b) diagram of harmonic wave

Fig. 5(a-b): Diagram of radial vibration analysis curves (6th order), (a) amplitude-speed curve, (b) diagram of harmonic wave

characteristic reflects obvious peak, namely in vibration waveform Fig. 6 represents a cycle 5 host and one propeller incentives within a cycle.

LONGITUDINAL VIBRATION ANALYSIS

By using the same method as used in the transverse vibration analysis, the maximum shafting longitudinal amplitude at various orders and its corresponding shaft rotation speed is obtained, as shown in Table 5 (Chen, 2002; Liu *et al*., 2013). From Table 5 it can be seen that the longitudinal vibration amplitude at 5th order is bigger than that of the other orders which makes their corresponding working condition become the typical order for analyzing the stern bearing wear conditions.

Spectrum characteristics of shafting longitudinal vibration at 5 orders are shown in Fig. 6. From Fig. 6a it can be seen that there is a peak value of the shafting longitudinal vibration amplitude within the main engine speed range, the corresponding speed of the shafting

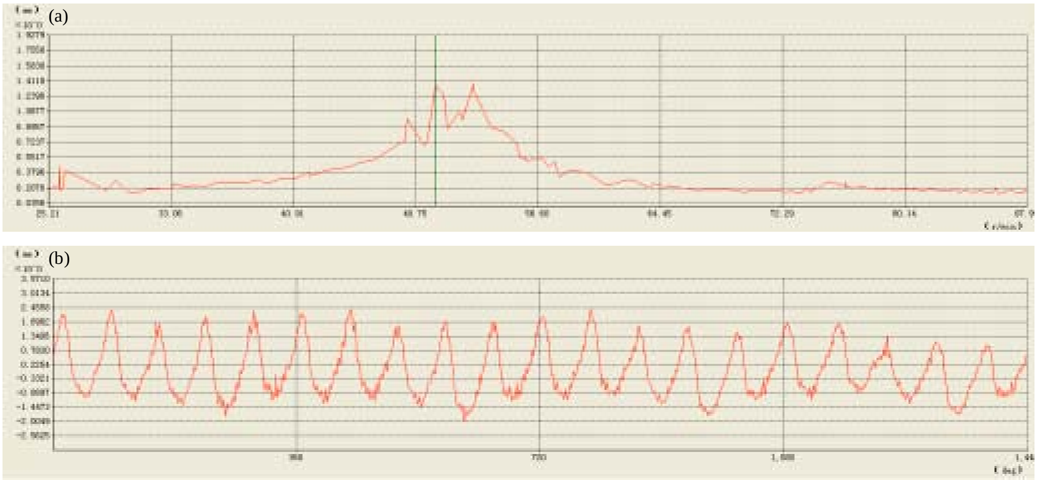


Fig. 6(a-b): Diagram of longitudinal vibration analysis curves (5th order), (a) amplitude-speed curve, (b) diagram of harmonic wave

Table 5: Maximum amplitude and shaft speed at various orders (longitudinal vibration)

Order	1	2	3	4	5	6	7	8	9	10
Amplitude (mm)	0.410	0.101	0.152	0.108	1.376	0.106	0.116	0.085	0.059	0.475
Speed (r min^{-1})	62.640	61.150	48.190	50.560	50.020	87.990	73.880	52.480	52.480	26.260

system is the resonance speed. At the same time within the 50~53 r min^{-1} speed range, there are large vertical shafting vibration amplitudes. These amplitudes are in the permissible range, so the impact on the stern bearing wear can be ignored. As shown in Fig. 6b, there are five peaks, showing the coupling effect of main engine and propeller incentives.

CONCLUSION

There is a correlation existing between the large new ship shafting vibration and stern bearing wear. This correlation is reflected under the coupled action of the propeller incentives and the main engine incentives. The maximum amplitude of shafting torsional vibration and longitudinal vibration occurs in the main engine incentive order while the maximum amplitude of shafting transversal vibration occurs in the main engine and the propeller excitation coupling order. The risk of stern bearing wears increases as the shafting vibration amplitude rises, especially embodied in the transverse vibration characteristics. Further study is to be carried on in future time. Currently only the correlation between ship shafting vibration and stern bearing wear is created. There is still no mathematical model to describe the correlation. In further research work, the correlation model will be focused on and more efforts will be put on probing into the numerical simulation and experimental research.

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