Modelling and hull vibration calculation of very large container ship

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Abstract

Hull vibration is inevitable in large vessel's operation. Too large hull vibration will not only cause damage to the hull structure, but also will affect the normal use of marine equipment and crew at work and life. Therefore, to predict the performance of the ship's structure vibration in ship design stage is essential, which can guide reasonable solutions and necessary damping measures. The vibration calculation report includes: Ship's free vibration calculation and forced vibration response calculations caused by vibration source on board the ship. The former is mainly to obtain the free vibration characteristics of the whole ship structure, to find hull's 1~3 order of vibration modes which are most likely to occur and the corresponding frequency values. The latter is mainly get the vibration response of hull ship's forced vibration caused by excitation source, to avoid low-frequency vibration's affect to the work and life of the crew. Its response should meet the relevant standards and requirements.

Keywords: modelling, free vibration, forced vibration, container ship, FEM

1 Introduction

The rapid increase in speed and size of the ships constructed during the 1960's led to the realization that wave-induced ship hull vibrations can give rise to significant stresses in the hull. The vibrations are usually classified as either whipping or springing, depending on whether the vibration mode is transient or steady state [1]. Ship hull vibration can be generally classified into two categories, global and local vibrations. For global vibration, the whole hull girder of a ship is vibrating in response to the excitation at particular revolutions of the main engines, propellers and auxiliary machinery, or from water waves [2]. Local vibration occurs when only an isolated part of the ship structure is in resonance [2,3]. Local resonance can be treated locally by modifying the resonating structural component or by adding vibration absorption and damping devices. Nevertheless, excessive deformation of the ship hull is more likely to come from global vibration, particularly at the first few fundamental modes of the ship structure.

Generally speaking, the global vibration of a ship, including natural frequencies and mode shapes of a complete ship hull structure, is analysed by strip theory where natural frequencies of the entire ship are calculated from beam theory and the estimation of weight and moment distributions at each strip of the ship hull. A method of such an approach is given by Todd [2]. Van Gunsteren also investigated wave-induced ship hull vibration [4], the two-node vertical vibration mode of two ship models using a modified strip theory is calculated. Because of the fast advance of computer technology and the increasing speed and capacity of modern computers, it's possible to analyse the low frequency dynamic response of a complete ship structure in a three-dimensional model using finite element analysis (FEA). FEA is used increasingly in the analysis and design of complex ship structures [5].

Lin and Pan [6, 7] used a closed form solution to define the characteristics of input mobility [8] of finite ribbed plate structures to force and moment excitations. Their research showed that the point force input mobility of a finite rib-stiffened plate is bounded by those of the corresponding unribbed plate and beam(s) of the ribbed plate. The input mobility is dominated mainly by the beam bending stiffness when the force excitation is applied to the beam, and its plate stiffness controlled when the beam is more than a quarter of plate bending wave length away from the force location. The result has been verified experimentally by Nightingale and Bosmans [9]. The torsional moment input mobility of a ribbed plate (for ribs having relatively small torsional stiffness) is dominated by the plate bending stiffness and its frequency-averaged value can be represented by that of the corresponding infinite plate. Lin [7, 10] extended the study to further investigate the characteristics of wave propagation and attenuation of finite periodic and irregular ribbed plates by employing a modal approach. He found that vibration of a ribbed plate structure can be confined by imposing irregularity to rib locations on the plate. The study of vibration characteristics of rib-stiffened plates provides a general understanding of wave propagation and its control mechanism in ribbed plate structures. However, when the structure becomes complex as in the case of a complete

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ship hull structure, the complexity of analysis increases due to the coupling of different wave types and multiple wave propagation paths in the structure[11].

In this paper, the calculation includes two parts: Calculation and analysis of the hull free vibration and forced vibration caused by vibration source on board. The former is mainly to ascertain the free vibration characteristics of the hull girder to determine whether the hull girder may be related to the main exciting force resonance under normal operating conditions. The latter is to ascertain the forced vibration characteristics (velocity and acceleration) of hull girder and superstructure caused by hull excitation source under normal operating conditions. To compare the vibration response values of some important local structure in the superstructure and engine room with the corresponding requirements of ISO standards to check whether necessary measures to be taken to improve the vibration performance of the ship [12].

2 Large container ship's finite element model

The calculation is based on the 9200TEU container ship designed by CIMC Naval Architecture and Ocean Engineering Design Institute. The finite element model used in 9200TEU container ship's vibration calculation is built entirely in accordance with the relevant design drawings, and the processing and analysis operations is completed by commercial finite element analysis software MSC/PATRAN and NASTRAN.

Global coordinate system of the model is right hand Cartesian coordinate system:

- X direction goes along ship's length pointing to bow;
- Y direction goes along ship's breadth pointing to Portside;
- Z direction goes along ship's depth pointing to deck.

Structural model and applied loads are in International System of Units (N, mm, s)

All plate structures, such as shell, transverse bulkhead, inner bottom, web frame and longitudinal bulkheads etc. are modelled by CQUAD4 and CTRIA3 shell elements. All girders and stiffeners are modelled by eccentric beam with appropriate combination. Small structures with hole are adjusted when finally balancing the quality of the whole ship structure and large structures with holes are modelled according to their actual shape as possible.

Mesh size in longitudinal direction is strong frame spacing and in transverse direction is longitudinal spacing. There are 2 elements in vertical direction between platforms. The floors and girders in double bottom are divided into 3 elements along the height direction.

Finally, there are and 450000 finite elements and 1320000 degree of freedom. Finite element models used in different loading conditions are shown in Figures 1 and 2.

During the process of establishing the finite element model the hull weight and the position of gravity centre are controlled by adjusting density of material of some elements and adding structural mass point. The deadweight in different loading conditions is adjusted by adding structural mass points at corresponding loading position and the mass points are linked to surrounding nodes by MPC. Final total weight and gravity centre of the model and those in loading manual are shown in Table 1.

Hull added mass is computed by empirical equation. Vertical and horizontal added mass is respectively computed in each loading condition. The computed added mass is added to shell underwater in the form of concentrated mass point.

In the computation of vertical vibration, the added mass is computed by the formula proposed by F M Lewis and F H Todd:

$$m_{av} = \frac{1}{2}\pi a_v C_V K_i \rho b^2 , \qquad (1)$$

added mass formula in horizontal vibration is:

$$m_{aH} = \frac{1}{2}\pi a_H C_H K_i \rho d^2 . \qquad (2)$$

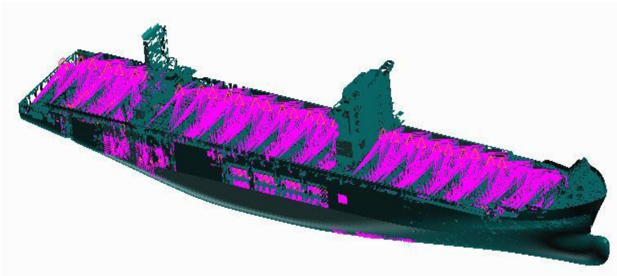

FIGURE 1 14TD loading condition element modal

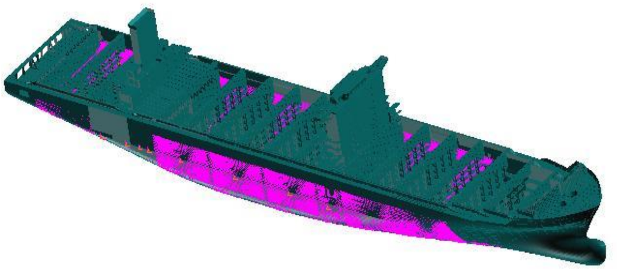

FIGURE 2 Ballast loading condition element modal

TABLE 1 Finite element modal's weight adjustment

Loading Condition	Ballast Arrival		14TD	
	Loading manual	*Finite element model*	*Loading manual*	*Finite element model*
Mass	66958.4	66100	147442.0	146900
Centre of gravity-X	-16.09	-17	-6.13	-6.4
Centre of gravity-Y	0	0.073	0	9.09e-2
Centre of gravity-Z	11.48	11.48	18.676	20.5

3 Free vibration analysis and calculation of global structure

Calculation and analysis of free vibration of global ship structure is to test whether the main excitation source frequency is close to the natural frequency of the hull, so as to modify the design to avoid resonance when necessary. In global ship structure analysis of free vibration, a very important part is to simulate the ship free floating in the water without boundary conditions.

Table 2 shows the typical free vibration frequencies from FEM calculation, some of the global ship typical free vibration typical modes under the 14TD loading conditions are shown in Figure 3-5, and Figure 6-8 represent the global ship typical free vibration modes under the ballast loading conditions.

Main engine excitation frequency is:

$$f = \frac{78}{60} = 1.3Hz\,, \tag{3}$$

which is close to the 2-nd order vertical natural vibration, frequency1.356Hz under the BLSA condition, the difference is 4.3%; compared with the 3 order vertical natural vibration frequency under the 14TD condition is 1.476Hz, the difference is 13.53%, satisfies the requirements.

According to the “Ship vibration control guide 2000”, in order to avoid the resonance between the low order vibration frequencies and the excitation frequencies, the 1-3 order natural frequencies must be different from the excitation frequency at the level ±8%~±10%, ±10%~±12%, ±12%~±15%.

If the requirements are not satisfied, the vibration response calculation and the vibration response measurement are necessary, which will be done according to the ISO standard (6954 resolution, 1979.9): vertical and horizontal vibration evaluation criteria of merchant ships, whose length are greater than 100 m. If the response values are greater than the standard values, vibration reducing measures should be taken.

TABLE 2 Typical free vibration frequencies from FEM calculation (Hz)

Loading condition	**Vertical vibration**			**Torsional vibration**	**Horizontal& Torsional vibration**				
	1 order	*2 order*	*3 order*	*1 order*	*1&2 order*	*2&2 order*	*2&3 order*	*3&3 order*	*3&4 order*
14TD	0.470	0.897	1.476	0.354	0.603	0.933	1.056	1.279	--
BLSA	0.716	1.356	1.914	0.516	0.674	1.43	1.68	--	--

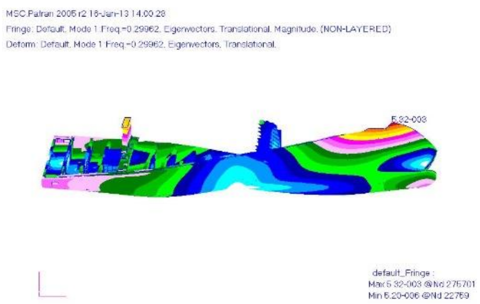

FIGURE 3 1 order torsional vibration mode under the14TD loading condition (0.299Hz)

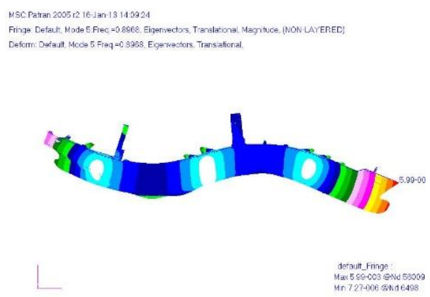

FIGURE 4 2 order vertical vibration mode under 14TD loading condition (0.897Hz)

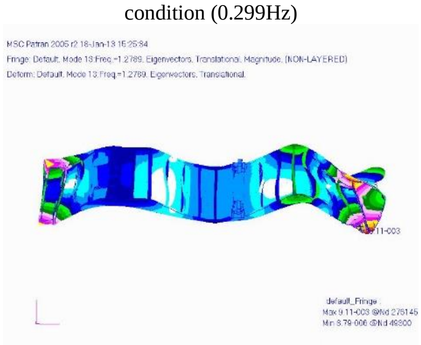

FIGURE 5 3 order horizontal and torsional vibration modes under 14TD loading condition (1.279Hz)

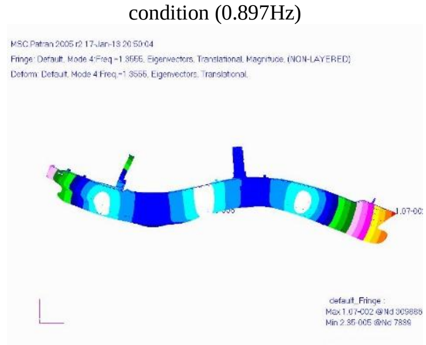

FIGURE 6 2 order vertical vibration mode under BLSA arrival loading condition (1.356Hz)

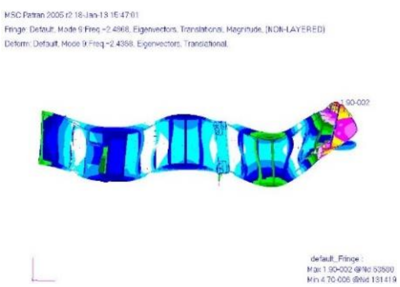

FIGURE 7 3 order horizontal and torsional vibration modes under BLSA arrival loading condition (2.437Hz)

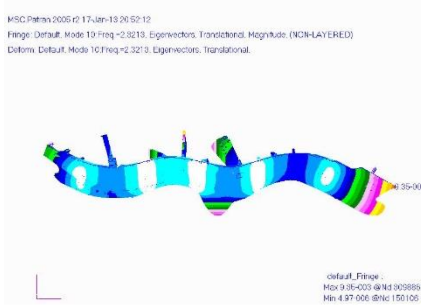

FIGURE 8 4 order vertical vibration mode under BLSA arrival loading condition (2.32Hz)

4 Global ship forced vibration response calculation

A Global ship forced vibration frequency response analysis considering the excitation force, which is the main excitation force, is a method to define the whole hull vibration response characteristics, so as to make an investigation of the working and living comfort on the supper structure.

Theoretically, propeller excitation force is the surface force, which is calculated from the integral along the hull surface, the force applying area is D×D (D is the diameter of the propeller). When the bottom is V type, the excitation force distribution is very irregular, accursedly usually, the value calculated by the propeller surface pressure formulation is used in the dynamic response calculation. The pressure could be applied on the active area directly. In this calculation the measured values of the typical points provided by the manufacturer is used, the applied area is D×D, about 8.9m*8.9m. Figures 9 and 10 are excitation force diagrams.

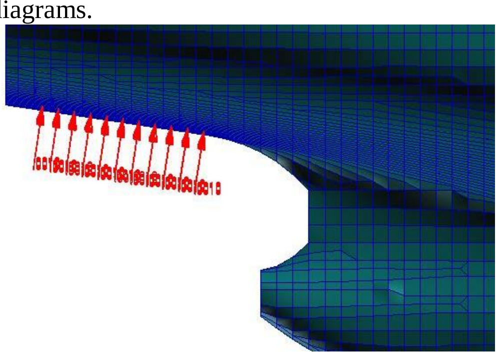

FIGURE 9 Propeller excitation force acting diagram (14TD loading condition, draft 14.8m)

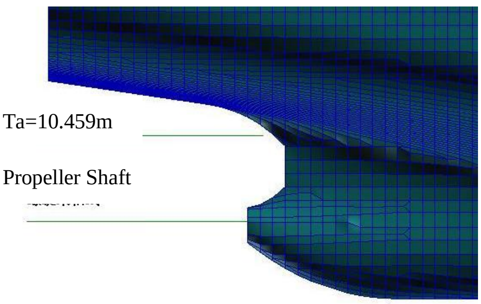

FIGURE 10 Excitation force range diagram (BLSA arrival loading condition, there is almost no pressure on the bottom plate)

Basically the structural damping could be obtained from ship experiment. However, in calculation usually the advantage of the damping coefficients obtained from classification societies through a large number of experiments, such as DNV and GL defined the damping coefficients in vibration response calculation, could be used. In this report we use the damping coefficient defined by GL in the vibration response calculation of 2750TEU container ship, the damping coefficient is 1% when the frequency is 0Hz, 8% when the frequency is 20Hz; under the BLSA loading conditions that is 0.5% when the frequency is 0Hz, 6% when frequency is 20Hz, shown as in Figure 11.

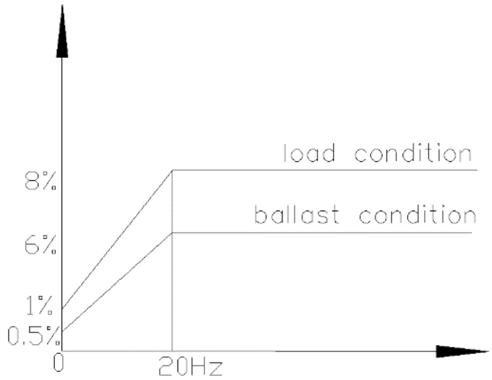

FIGURE 11 Damping factor to frequencies

Main engine excitation source has been shown in Table 3, the vibration excitation source would be loaded on the centre of gravity of the main engine through the junction points of the main engine and the internal bottom plate, the local structure diagram is shown in Figure 12.

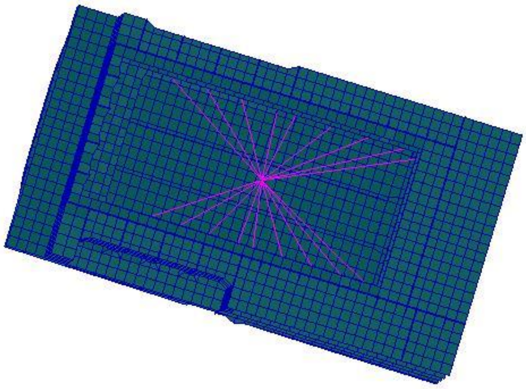

FIGURE 12 Local structure diagram with excitation source of main engine

TABLE 3 Engine imbalance force and imbalance moment

External forces(KN)		**Guide force X-moments in (KNm)**	
1.order:Horizontal	16	1. order:	503
1.order:Vertical	16	2. order:	21
2.order:Vertical	19	3. order:	1905
4.order:Vertical	80	4. order:	1947
6.order:Vertical	3	5. order:	1987
External moments(KNm)		6. order:	4103
1.order:Horizontal a)	835	7. order:	758
1.order:Vertical a)	835	8. order:	545
2.order:Vertical	764	9. order:	100
4.order:Vertical	468	10. order:	176
6.order:Vertical	107	11. order:	126
Guide force H-moments in (KNm)		12. order:	174
1x No.of cyl.	1078	13. order:	61
2x No.of cyl.	97	14. order:	76
3x No.of cyl.	-	15. order:	140
		16. order:	76

The velocity and acceleration to frequencies curves of the typical points are shown in the Figures 13-20; the absolute values of velocities and accelerations are used.

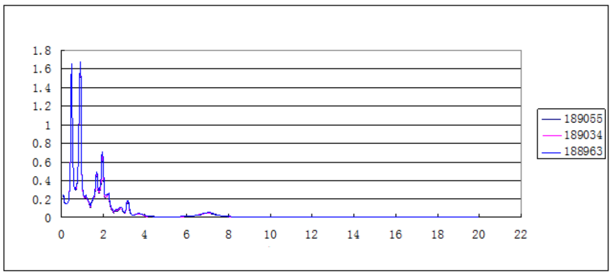

FIGURE 13 Velocity curves of typical points on A deck

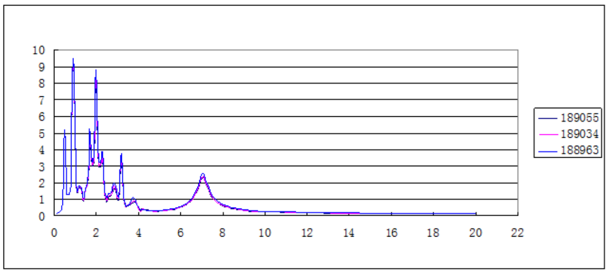

FIGURE 14 Acceleration curves of typical points on A deck

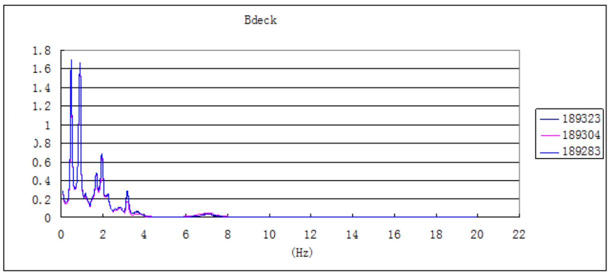

FIGURE 15 Velocity curves of typical points on B deck

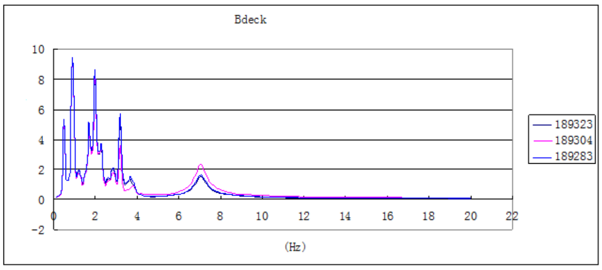

FIGURE 16 Acceleration curves of typical points on B deck

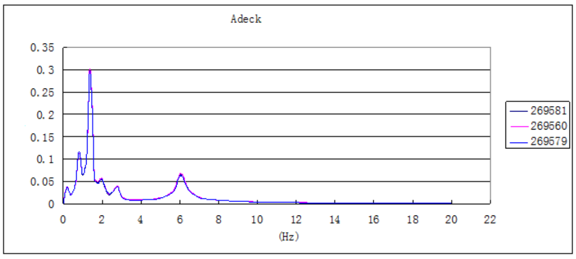

FIGURE 17 Velocity curves of typical points on A deck

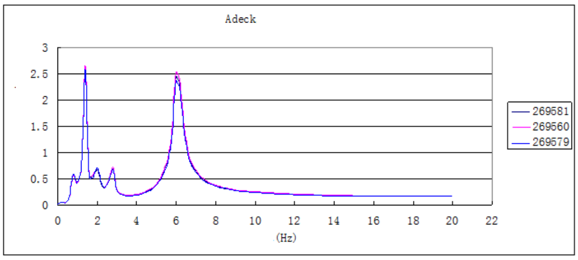

FIGURE 18 Acceleration curves of typical points on A deck

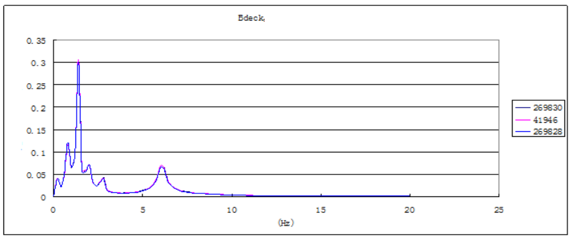

FIGURE 19 Velocity curves of typical points on B deck

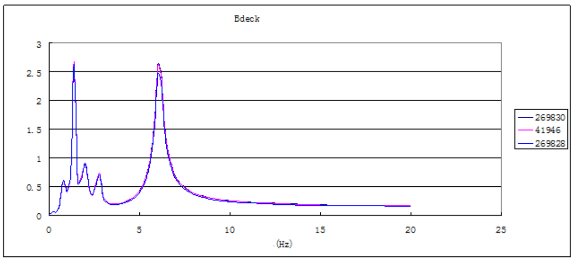

FIGURE 20 Acceleration curves of typical points on B deck

By the velocity and frequency response curves and acceleration frequency response curves it could be shown that, when the frequency is equal to 0.498Hz and 0.896Hz, velocity and acceleration response become larger values. Compared to the free vibration mode of 14TD loading condition, 0.47Hz is the frequency of the 1 order vertical vibration mode. So when the excitation force frequency is 0.498Hz, the vibration response of the local structure of the supper structure will be larger; 0.897Hz is the natural frequency of 2 order vertical vibration, so when the excitation force frequency is 0.897Hz the second vibration peak value appears. However all the calculated velocities and accelerations are under the corresponding criteria values in ISO6954 (2000E).

From the velocity and acceleration frequency response curves it could be seen that, with the increase of vibration frequency, vibration response amplitude increases as well, when the excitation frequency approach to the 1st order natural frequency of the hull girder. The amplitude will reach the maximum value firstly, at this moment the mode of the forced vibration is close to the 1 order mode, and the resonance will occur.

When the excitation frequency is further increasing, the hull response decreases quickly, then increases gradually, until the excitation frequency approach to 2 order natural frequency, the amplitude will reach the second peak point, and the second resonance is going to happen, followed by the third order, fourth order resonance and so on.

Through calculation and analysis, under this condition, the velocity response and acceleration response of typical points satisfy the corresponding requirements.

5 Conclusions

The calculation of free vibration and forced vibration of 9200TEU container vessel is carried on in a 3D FEM environment, with the help of the large commercial FEM software MSC/PATRAN and NASTRAN to achieve, by adjusting the material density and applying concentrated mass to equal the loading mass distribution in the manual. Added mass is calculated by empirical formula calculate, and added to the mass points below the water line.

From the calculation, we can see that, vibration response amplitude increases with the increase of vibration frequency, the amplitude will reach the maximum value when the excitation frequency approach to the 1st order natural frequency of the hull girder, the resonance will occur at the moment the mode of the forced vibration is close to the 1 order mode. And hull response decreases quickly when the excitation frequency is further increasing, then increases gradually, until the excitation frequency approach to 2 order natural frequency, the amplitude will reach the second peak point, and the second resonance is going to happen, followed by the third order, fourth order resonance etc.

Through the analysis of free vibration and forced vibration of 9200TEU container ship, most of the forced vibration response values of checking points are less than the standard criteria and satisfy the vibration requirements.

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