Response Analysis of Jack-Up Ship Structures

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Abstract. Jack-Up Ships are specialized vessels in the field of marine engineering, capable of meeting various needs such as offshore oil and gas exploration and extraction, as well as transportation and installation of offshore wind turbine units. With broad prospects for application, they hold significant research value and strategic importance. Based on the finite element analysis method for forced vibration response analysis of jack-up ship, a finite element models of jack-up ship is established to analyse the whole ship modes and the natural and excitation frequencies, and the whole ship vibration response is solved by the finite element method with modal superposition after loading the excitation loads. The reason why the response peak is not near the excitation frequency is analyzed, providing a reference for vibration prediction and control.

Keywords: Jack-up ship; Finite element method; Natural characteristic; Forced vibration; Modal method.

1 INTRODUCTION

With the further development of marine resources in contemporary society, the research and development of marine engineering equipment should also keep pace with the times. As a special ship in the field of ocean engineering, the jack-up ship further developed from the jack-up platform has the characteristics of both ship and jack-up platform, with strong maneuverability, multiple types and complete functions. The more widely used ones are the jack-up marine oil and gas vessels for exploration and exploitation of offshore oil and gas resources exploration and exploitation, and the jack-up wind power installation vessels for offshore wind turbine transportation and installation.

Forced vibration analysis is crucial in the design of jack-up vessels, as it directly impacts the structural integrity, safety, and passenger comfort of the ship. Through this analysis, it is possible to predict and assess the stresses and fatigue that the ship's structure may endure under different working conditions, thereby ensuring safe operation and extending the service life. Additionally, forced vibration analysis helps comply with international and regional vibration standards, reduces maintenance and operational costs, improves fuel efficiency, and enhances passenger comfort. Based on the

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relevant structural drawings of a jack-up ship, this paper applies MSC.PATRAN software to establish three-dimensional finite element models and analyze the forced vibration response of the ship under working conditions to ensure that it meets the specification requirements.

2 FINITE ELEMENT MODEL AND VIBRATION THEORY

2.1 Finite Element Model

This paper uses MSC.Patran software to establish a finite element model of a jack-up ship, including shell elements, beam elements, and mass point elements. The model focuses on the stiffness and mass distribution of the hull, optimizing the grid distribution to reduce the errors in vibration calculation, especially refining the grid at the engine room, the main source of excitation, to better simulate the actual situation[1]. The established finite element model is shown in Figs 1.

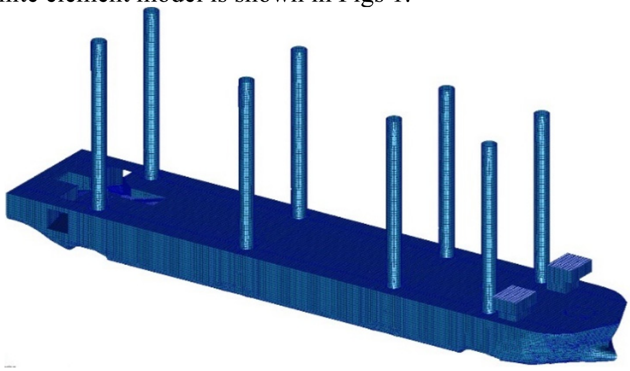

Fig. 1. Jack-up ship

2.2 Vibration Theory and Model Setup

Vibration analysis differs from static analysis in that the primary distinction lies in the time-varying nature of the loads involved in vibration analysis. Consequently, it is essential to consider the stiffness, inertia, and damping of the structure. Vibration forecasting is based on the principles of vibration analysis theory, which mainly includes modal analysis, harmonic response analysis, and random vibration analysis. Modal analysis is commonly used in the field of engineering vibration to study the inherent vibrational characteristics of a structure. This method provides a basis for response and vibration reduction analysis by examining the vibrational modes of the structure (natural frequencies, mode shapes, and modal damping). There are three main methods for vibration calculation: the energy method, the transfer matrix method, and the finite element method. This article primarily employs the finite element method.

The finite element method, integrated with computer technology, has become a primary tool for analyzing structural vibration characteristics, particularly suited for calculating the modal and forced vibration responses of ship structures. In applying the finite element method to ship vibration analysis, the structure must first be discretized into multiple elements connected at nodes to form a computational model. This involves calculating the stiffness matrices for each element and assembling them into a global stiffness matrix while also constructing a mass matrix. In analyzing the natural frequencies of the ship structure, damping effects are usually neglected, forming a system of multi-degree-of-freedom vibration equations. Solving these equations typically involves solving generalized eigenvalue problems to determine the structure's natural frequencies and mode shapes. The establishment of the finite element model considers the requirements for one-dimensional, two-dimensional, three-dimensional, or hybrid models, and based on the model, the stiffness and mass distributions are determined to conduct the finite element analysis.

The lifting of the main hull of a jack-up vessel is primarily achieved through a lifting device connected to the legs. When the platform is in normal operation, the legs are partially inserted into the mud for stabilization, and the model of the leg insertion range is treated with full constraints.

The application of the correct excitation source plays a decisive role in the results of vibration analysis[2]. The main sources of excitation for jack-up ships under transport conditions are the main engine and auxiliary engines, and other equipment excitations are relatively small in comparison and therefore negligible. Manufacturer-provided vibration data for the main and auxiliary engines are shown in Table 1 below.

Table 1. Diesel engine set parameters

equipment name	Rated speed(r/m)	Power(kW)
Main diesel generator sets	1000	4720
Auxiliary diesel generator sets	1500	1760

The main engine and auxiliary engines only have torque on the hull:

$$Cl = 4500 + 2400\cos\left(4\omega t + 62^{\circ}\right) + 360\cos\left(8\omega t + 206^{\circ}\right) \tag{1}$$

In the formula, $\omega = \dfrac{2\pi}{T} = 2\pi f = 2\pi \times \dfrac{1000}{60}$ rad/s

The first-order load frequency of the main engine and auxiliary engine at rated speed are 16.67 Hz and 25 Hz, respectively, and its excitation torque versus time curves are shown in Figs. 2 to 3.

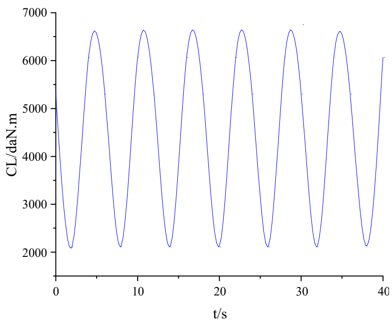

Fig. 2. The curve of excitation torque of main engine changes with time

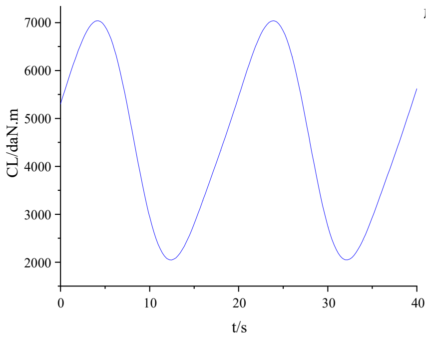

Fig. 3. The curve of auxiliary excitation torque changes with time

2.3 Damping

The damping of offshore platform can be regarded as a combination of viscous damping, structural damping and friction damping. The structural damping mechanism which is the dominant factor is still unclear and difficult to quantify. The viscous damping can be obtained by hydrodynamic analysis. Friction damping is generally not considered in dynamic analysis.

The calculation of this paper refers to the American Bureau of Shipping specification "Guidance Notes on Ship Vibration", the whole ship forced vibration calculation can be taken in the calculation frequency range of the critical damping ratio of 1.5%[3]. According to the relationship between the critical damping ratio and the structural damping coefficient of the *MSC Nastran Dynamic Analysis Guide* (see Equation 11), the structural damping coefficient of the ship is taken to be 3%.

$$\xi = g/2 \tag{2}$$

In the formula: ξ ——Critical damping ratio; g ——Structural damping coefficient.

3 VIBRATION CHARACTERISTICS PREDICTION AND ANALYSIS

Seawater that vibrates together with the vibration of the ship is called additional water. In modal calculations, additional water is equivalent to a portion of the ship's own weight and cannot be ignored. Therefore, in finite element calculations, the modes calculated after applying additional water are called wet modes. The use of finite element calculation method to calculate wet modes not only eliminates the large experimental cost of the experimental method, but also can take more complex factors into account, which is the mainstream direction of the current research[4].

3.1 Vibration Characteristics Prediction

In this paper, we consider the effect of additional water on the modes. From the vibration modes predicted by the numerical results, the first 6 orders of vibration modes are those of the pile legs, and there is no obvious change in the vibration modes of the hull. The hull and pile leg coupling vibration mode appears from the 7th order mode. The higher order hull modes tend to be localised when the intrinsic frequency is large. The environmental parameters for the three operating conditions are listed in Table 2, and the lower order natural frequencies of the pile legs and the hull for the three operating conditions are listed in Tables 3 to 4. Typical modal vibration modes for Loadcase 1 are listed in this paper.

Table 2. Vibration modal analysis of target ship

Loadcsae	Depth of pile leg into mud (m)	Operating depth (m)	Maximum air gap (m)
LC1(Pile legs with additional water)	2.5	6.5	0.5
LC2(Pile legs with additional water)	2.5	6.5	0
LC3(Pile legs and hull with additional water)	2.5	6.5	-0.5

Table 3. First six natural frequencies of the pile legs

Loadcase	Natural frequency (Hz)					
	First order	Second order	Third order	Forth order	Fifth order	Sixth order
LC1	2.2345	2.2682	2.2998	2.3888	2.4786	2.4839
LC2	2.2315	2.2486	2.2730	2.3586	2.4232	2.4289
LC3	2.2082	2.2195	2.2416	2.3240	2.3419	2.3687

Table 4. Hull and pile leg coupling vibration mode and frequency

Loadcase	Natural frequency (Hz)			
	Vertical first-order vibration mode of stern	Horizontal first-order vibration mode	Vertical first-order vibration mode of bow	Horizontal second-order vibration mode
LC1	3.3189	4.3848	4.8045	7.6083
LC2	3.3137	4.4109	4.6780	7.6554
LC3	3.0199	4.4415	3.1423	7.6899

3.2 High Sensitivity Analysis of Additional Water

The working environment of this jack-up ship is wave level 4. Under this sea condition, the meaningful wave height is 1.25~2.5 m. In this paper, two significant wave heights of 1.25 m and 2.5 m are selected to carry out vibration modal analysis of the jack-up ship under the working conditions of 2.5 m of pile leg in mud, 6.5 m of operating water depth and 0 m of maximum air gap. The effect of significant wave height on the vibration mode of the jack-up ship is observed, and the specific working conditions are shown in Table 5.

Table 5. Analysis of the effect of significant wave height on vibration frequency

Loadcase	Depth of pile leg into mud (m)(m)	Operating depth(m)	Maximum air gap(m)	Significant wave height(m)
LC1-1	2.5	6.5	0	0
LC1-2	2.5	6.5	0	1.25
LC1-3	2.5	6.5	0	2.50

In this paper, the first 50 orders of wet modes of the target ship vibration are calculated analytically under the conditions of significant wave heights of 0 m, 1.25 m and 2.50 m. The first six orders of self-oscillation frequencies of the pile leg vibration are selected for comparison. The natural frequencies of the first six orders of the pile leg vibration are selected for comparison, and the curve variation of the first six orders of the natural frequencies of the pile legs from LC1-1 to LC1-3 is shown in Fig. 4. From the figure, it is found that the natural frequencies of the six orders from LC1-1 to LC1-3 have an increasing trend, and the natural frequency of the pile leg decreases as the significant wave height is larger. This is due to the fact that the larger the significant wave height is, the deeper the hull is immersed in seawater, and the greater the effect of the additional water on the pile leg vibration, resulting in a lower natural frequency of the pile leg.

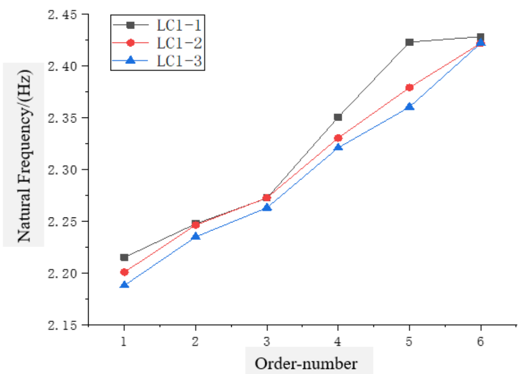

Fig. 4. Frequency curve of pile leg vibration mode of jack-up ship

3.3 Frequency Reserve Assessment

According to China Classification Society (CCS) "Shipboard Vibration Control Guidelines" (2021), in order to prevent resonance from occurring because of the close proximity of the shipboard vibration source excitation force and the ship's natural frequency of a certain order, the requirements for the reserve frequency are as follows: the value of the difference between the intrinsic frequency of the 1st order and the excitation frequency should be at least 8~10%, the value of the difference between the intrinsic frequency of the 2nd order and the excitation frequency The value of the difference should be 10%~12%, and the value of the difference between the 3rd stage intrinsic frequency and the excitation force frequency should be 12%~15%. If the frequency reserve requirement is not met, the frequency of the hull girder needs to be changed or the frequency of the excitation needs to be changed to stagger the frequencies. If it still does not meet the requirements, it is necessary to carry out frequency response analysis or real ship measurements, when it does not meet the requirements of the balance, it must take corresponding vibration damping measures[5].

From the calculation results, it can be seen that: under three kinds of working conditions, the pile legs of the ship vibration natural frequency reserve coefficient are more than 85.10%, and the total hull vibration natural frequency reserve coefficient are more than 53.87%, which is much larger than the normative requirements of the reserve coefficient, and meets the requirements of the balance of power. The low-order natural frequency of the whole hull beam will not meet with the excitation frequency of the main engine, main thrust propeller, telescopic thrust propeller and side thrust propeller, and thus will not produce the overall resonance of the hull. The staggered frequency ensures the safety of the hull structure and provides important guidance for the early construction of the hull structure.

4 SIMULATION RESULTS

The jack-up ship adopts the finite element method of modal superposition to solve the vibration response of the whole ship, the frequency range of calculation is from 1~80Hz, the step size is 0.5Hz. This paper gives the velocity response diagram at a typical compartment, and the results are as follows: Figs. 5 to 10.

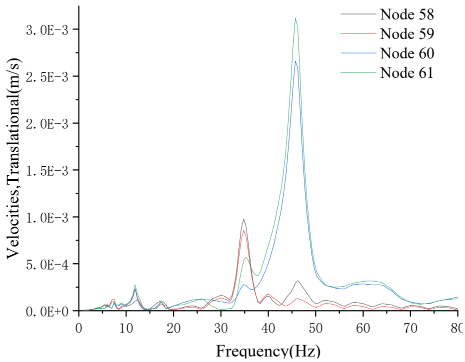

Fig. 5. NO1 Engine room measuring point speed frequency response curve

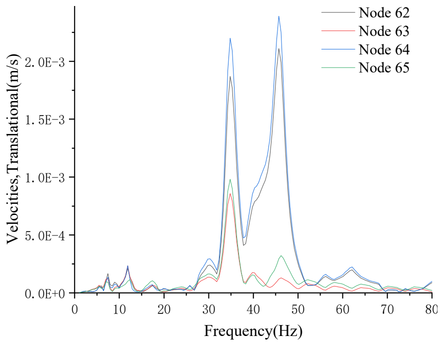

Fig. 6. NO2 Engine room measuring point speed frequency response curve

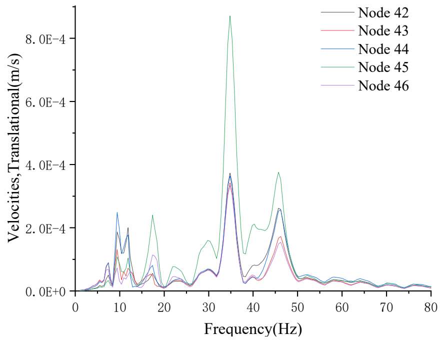

Fig. 7. NO1 Electrical equipment room measuring point speed frequency response curve

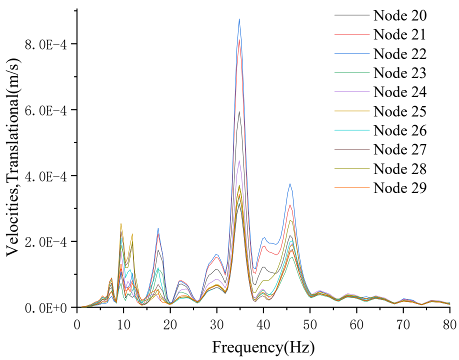

Fig. 8. NO2 Electrical equipment room measuring point speed frequency response curve

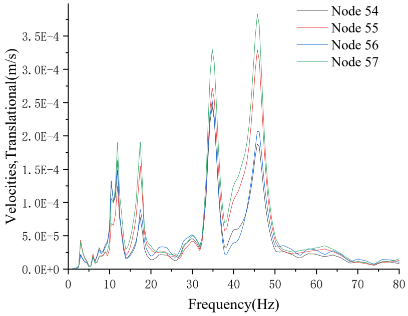

Fig. 9. NO1 Rudder propeller cabin measuring point speed frequency response curve

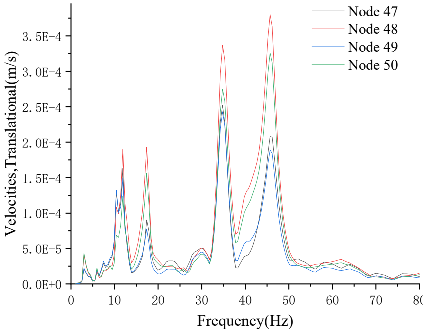

Fig. 10. NO2 Rudder propeller cabin measuring point speed frequency response curve

The vibration response speed and acceleration are evaluated by using the international general vibration standard ISO6954 (2000) for marine structures. The specification ISO6954 assesses vibration in the frequency range of 1 to 80Hz, mainly because the resonance frequencies of various human tissues are concentrated in the range of 1 to 80Hz, and the human body is particularly sensitive to environmental vibration in this frequency range[6][7]. The specification also introduces frequency-based weighting coefficients, on the basis of which the root-mean-square (RMS) value of the weighted values at all frequencies is sought to evaluate the hull vibration, the weighting curve is shown in Fig. 11, and the specific values of the scale are shown in Table 6.

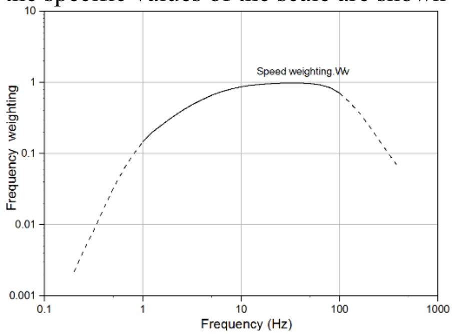

Fig. 11. Velocity correction factor curve

Table 6. ISO 6954 (2000) Vibration Measurement Standards

	Regional classification					
	Passenger Living Area		Crew Living Area		Crew Working Area	
	Acceleration ($mm·s^{-2}$)	Velocity ($mm·s^{-1}$)	Acceleration ($mm·s^{-2}$)	Velocity ($mm·s^{-1}$)	Acceleration ($mm·s^{-2}$)	Velocity ($mm·s^{-1}$)
Limit	143	4	214	6	286	8
Lower limit	71.5	2	107	3	143	4

[a] The value between the two limits indicates that the vibration condition of the ship is acceptable.

According to the requirements of the vibration specification IS06954 (2000)[8], after the excitation force, damping and additional water are applied, 1~80Hz frequency range 1/3 octave forced vibration response calculation is carried out for the whole ship[9], and the velocity at each frequency is corrected by weighting, and the maximum synthetic velocity of typical nodes of the compartments of interest is obtained as shown in Table 7.

Table 7. Synthesis velocity of typical nodes in cabin

Cabin	Node number	Weighted velocity	
		Measurement point value	Standard value
NO1 Engine room	61	3.1	8.0
NO2 Engine room	64	2.4	8.0
NO1 Electrical equipment room	45	0.4	6.0
NO2 Electrical equipment room	21	0.4	6.0
NO1 Rudder propeller cabin	57	0.4	8.0
NO2 Rudder propeller cabin	48	0.4	8.0

Comparison with the ISO 6954 (2000) specification values shows that the platform is in good vibration condition, and since the engine room is where the maximum velocity response peaks on the ship, the other areas of interest are able to fulfil the specification requirements.

5 CONCLUSION

(1)The jack-up ship in the three kinds of operating conditions of the pile legs of the natural frequency and the excitation frequency of the excitation source stagger as low as 85.10%, the total vibration of the hull of the natural frequency reserve coefficient are more than 53.87% is much larger than the balance of the requirements of the 8% ~ 15%, to meet the reserve coefficient of the balance of the requirements. The staggered frequency ensures the safety of the hull structure and provides important guidance for the early construction of the hull structure.

(2)The forced vibration analysis of the whole ship based on the modal method shows that the jack-up ship is under the effect of the combined excitation of the main engine and auxiliary engine. The peak velocity response at the engine room is much higher than that of other compartments, and its peak is 3.1m/s respectively, all of which satisfy the equilibrium requirements and meet the practical expectations.

From figures 5 to 10, it can be seen that the peak velocity response of the cabin appears near 40Hz. The reasons may be as follows:

(a) the main engine and auxiliary engines are in one cabin, which may form a resonance;

(b) the influence of some similar damping materials such as deck dressing, floating floor, and outfitting parts are not considered in the simulation calculation.

However, in general, the calculation results are basically in line with the practical application of engineering, which is of some guiding significance for early vibration prediction and control.

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