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DESIGN WAVE CALCULATION OF A PASSENGER CATAMARAN UNDER MULTIPLE LOAD CONTROL PARAMETERS

Shenwei Ge [1,2]
Ji Zeng *[3]
Bowen Jin [2]
Wen Zhou
Xiaolong Qin [4]
[1] Merchant Marine College of Shanghai Marine University, China
[2] Shanghai; Nantong Shipping College, Jiangsu Nantong, China
[3] Ocean Science and Engineering College of Shanghai Marine University, China
[4] China Merchants Cruise Shipbuilding Co. LTD., China

* Corresponding author: *zengji@shmtu.edu.cn (J. Zeng)*

ABSTRACT

To determine the loading conditions considering the action of both bending and torque moment for a passenger catamaran moving among waves, a method for calculating equivalent design waves under multiple load control parameters was derived based on wave load prediction results using three-dimensional potential flow theory. The method was developed by defining the wave amplitude discrepancy factors between the primary and second load of the combined bending and torquing equivalent design wave. The primary goal was to find a reasonable design wave. Finally, the design waves of a target passenger catamaran ship were calculated using the proposed method, and each load component of every design wave for the target hull was recalculated. The average error compared with the object load component was less than 1%, which verifies the effectiveness of the method and offers an effective engineering evaluation method for a catamaran.

Keywords: catamaran, load prediction, equivalent design wave, multiple load control parameters

INTRODUCTION

In the 1990s, the American Bureau of Shipping (ABS) proposed a shipping engineering design method called the dynamic load approach, or DLA, which uses the results of the direct wave load analysis to evaluate a ship's structural strength. At the same time, ABS also submitted the concept of an equivalent design wave. Combining this concept with DLA provides a very effective method for designing the hull structure [1]. The principle of an equivalent design wave is to use extreme values to simulate regular waves with equivalent wave frequencies, incidence angles, and amplitudes in random waves in a ship design. Thus, it leads to a simple cosine wave and a suitable time instant for extracting a set of concurrent load components to be applied to the structural models of ships [2]. Because this approach converts actual irregular waves into an equivalent regular wave, it is called the equivalent design wave approach (hereinafter referred to as the design wave approach) [3]. A set of dynamic load control parameters (hereinafter referred to as load control parameters, or DLP) is used in this approach to define any response processes (e.g., vertical bending moment, shear force, and vertical acceleration at the centre of gravity). When one DLP is at its maximum, a critical load is established for the hull structural analysis. The equivalent design wave is defined based on the extreme value of the DLP in long-term prediction.

The design wave approach can reflect the maximum wave loads on ships in their sailing lines more accurately and determine the maximum loads of longitudinal bending moment, torque, and other forces acting on the hull girder.

Therefore, it can be applied to ship structures of various types and scales, and is not restricted by the calculation methods for the load employed by each classification society.

In the calculation of the equivalent design wave, the extreme value of the DLP in long-term prediction is the key factor in ship structure analysis. Ochi [4] and Stiansen and Chen [5] pointed out that long-term extreme loads can be predicted using short-term statistical analysis of dynamic loading parameters in worst-case sea conditions. Prior to this, Ochi [6] calculated the extreme values for ships with various dynamic load parameters by applying a wave spectrum. Later, Buckley [7] analysed ship structure strength by applying the extreme load value under various sea conditions. Because of the popularisation of commercial software for hydrodynamic analysis, this technology has been widely used in ship wave load analysis.

In a study on the application of a design wave of a catamaran, Pu et al. [8] utilised a method to study the structural system reliability of a SWATH. The load was derived from the direct calculation using three-dimensional potential flow theory. Heggelund et al. [9] analysed the transverse strength of a catamaran with a cargo hold model. In this study, two kinds of local loads and four kinds of global loads were considered in the strength analysis. However, in the analysis of global ship loads, the distributions of the vertical bending, torque, and transverse bending moment were calculated and loaded on the model independently. Xu et al. [10] used experimental and finite element methods to analyze the ultimate strength of an inland catamaran. The objective ultimate load was the vertical bending moment. These studies all performed structural analysis under a single load or a single DLP. Sun and Zou [11] studied the hydrodynamic response of a slender catamaran in regular waves only in a head sea. However, compared with conventional cargo and passenger ships with a single hull, the loading form of catamaran ships is unique and more special and complex. In waves, catamaran ships are affected not only by the combined load of global longitudinal bending and the transverse bending moment, but also by the transverse bending moment and torsion moment [12-13]. When a single load acts on the hull structure alone, it does not necessarily place the hull structure in the most dangerous state. Moreover, these global loads cannot act on the hull structure at their maximum values at the same time. The intermediate state, where the bending moment and torque both make a certain contribution, is a dangerous condition for the hull structure. From this point of view, Lin et al. [14] confirmed that catamaran structures are in dangerous conditions with oblique wave directions of 30° or 120°. Therefore, this loading condition must be evaluated during the structural strength analysis of the catamaran. In all the aforementioned references, the torque, transverse bending moment, and vertical bending moment were applied to the structure in the form of a single force separately, and then the stresses were superimposed for the strength evaluation, rather than loading the design wave obtained by direct analysis on the hull structure for evaluation.

In this study, a passenger catamaran is used as an example, and an approach that is suitable for design wave calculation in multiple load control parameters is derived. This approach can be applied to the objective bending–torsional combination conditions effectively, which can find the reasonable wave parameters in one design wave reaching both the objective bending and torsional moment. Compared with the object loads, the error is less than 1%.

PROBLEM FORMULATION

WAVE LOAD PREDICTION

Short- and long-term prediction

The response of a ship hull under various load control parameters in regular waves with unit amplitude is generally called the wave load transfer function (RAO), i.e., the amplitude–frequency response operator.

The effect time of the wave load short-term prediction is half an hour to several hours, at which time the ship's speed, heading angle, and sea condition can be considered to be stable. Therefore, short-term prediction can be regarded as a stationary stochastic process with zero mean value; that is, the effect of the wave and the response of the ship is a smooth linear system. Through spectrum analysis, the response of the hull can be obtained as follows:

$$S_y(\omega) = \left|H(\omega)\right|^2 S_\omega(\omega) \tag{1}$$

where $S_y(\omega)$ is the response spectrum of the hull under a wave load, $|H(\omega)|^2$ is RAO, and $S_\omega(\omega)$ is the wave spectrum.

The effect time of wave load long-term prediction is on the scale of several years or the entire life of the hull, and the ships speed, heading angle, and sea condition all change; hence, it can no longer be regarded as a stable random process. Thus, a weighted combination of a series of short-term probability distributions is employed:

$$Q_y(x) = \frac{\sum_i \sum_j \sum_k \sum_l nP_iP_jP_kP_l n\left(1-F^*(x)\right)}{\sum_i \sum_j \sum_k \sum_l nP_iP_jP_kP_l} = \frac{1}{N} \tag{2}$$

where Q is the exceeding probability level; n is the number of responses in a given sea state and ship condition per unit time; P_i is the probability of occurrence of a sea state described by significant wave heights and periods (wave parameters that are taken from a scatter diagram); and P_j, P_k and P_l are the probabilities of the wave direction (heading to wave), ship speed, and loading condition respectively; $F^*(x) = 1-\exp[-x^2/(2\sigma^2)]$, which is the cumulative distribution function of the Rayleigh probability distribution; and N is the number of wave load cycles during the ship life, $N = 10^8$.

Using the principle of long-term prediction, the extreme wave load can be obtained at the expected exceeding probability level.

DESIGN WAVE CALCULATION METHOD

Single load control parameter

The design wave approach is based on the calculation results of long-term prediction; however, a design wave is used instead of a random wave. Thus, the maximum load that the ship may encounter in the entire life cycle is obtained, after which the wave is loaded onto the hull structure for strength analysis.

Based on the regular wave linear theory, the amplitude of the designed wave can be calculated using the following equation under single load control parameters:

$$A_\omega = \frac{L_j}{RAO_j} \tag{3}$$

where A_ω is the amplitude of the design wave, RAO_j is the extreme value of the target load control parameters in regular waves, and L_j is the extreme value of the target load control parameters under long-term prediction.

Multiple load control parameters

By determining a single load control parameter, one or several of the most dangerous design wave loads (such as maximum longitudinal bending moment or maximum transverse torque) can be determined. However, in some sea conditions, especially in oblique sea conditions, a catamaran is always affected by both longitudinal bending moment and transverse torsional moment or transversal bending moment at the same time, and the worst load condition for the hull structure does not necessarily reach the maximum bending and torsional moments simultaneously. Hence, at this point, if only a single load control parameter is used to determine the design wave, it is not accurate. At present, major classification societies have already considered the combined influence of bending and torsional moment for a catamaran in oblique sea conditions [15-16]. In the structural analysis of a catamaran using the China Classification Society (CCS) method, the combined loads should be considered, including the longitudinal bending moment with torque and the transverse bending moment with torque, which are calculated as follows:

$$O_i = \eta_{i1} M_{by} + \eta_{i2} M_{ty} \tag{4}$$

$$O_j = \eta_{j1} M_{bx} + \eta_{i2} M_{ty} \tag{5}$$

where O_i and O_j are the objective loads on the hull girder; η_{i1}, η_{i2}, η_{j1}, and η_{j2} are the coefficients of each load control parameter component, respectively; M_{by} is the longitudinal bending moment for the hull girder; M_{bx} is the transverse bending moment for the hull girder; M_{ty} and is the transverse torque for the hull girder (Fig. 1).

Calculating the aforementioned wave bending moment and torque using an empirical formula is convenient for calculating the objective loads. However, it is difficult to find the wave load required to reach multiple objects in complex sea conditions using the design wave approach. Therefore, a design wave calculation approach is proposed in this paper that takes the objective design wave of Eq. (4) as an example. Before calculating the design wave in this study, three principles should be noted:

(1) For each objective design wave load, the two load components must reach their maximum values (i.e., the product of the objective value and component coefficient η under a single load control parameter); moreover, the calculated profile of the objective load is determined by the single load control parameter.

(2) The calculation of the design wave should be as large as possible to make the evaluation of the structural strength relatively safe. However, the load component should not exceed the objective value to any great extent; otherwise, the load applied to the finite element model will be too conservative, and the structural strength will also be too conservative.

(3) Based on the principle of the design wave approach, the amplitude of the designed wave is obtained at the maximum *RAO*; hence, design waves that are seriously inconsistent with the actual wave amplitude under a single load control parameter can be eliminated.

In the calculation of *RAO*, the information concerning the various *RAOs* includes the amplitude response at each frequency and corresponds to a phase value ε_b. This phase ε_b is the phase of the maximum value of each load at a certain frequency, which varies with the cosine rule; hence, it can be

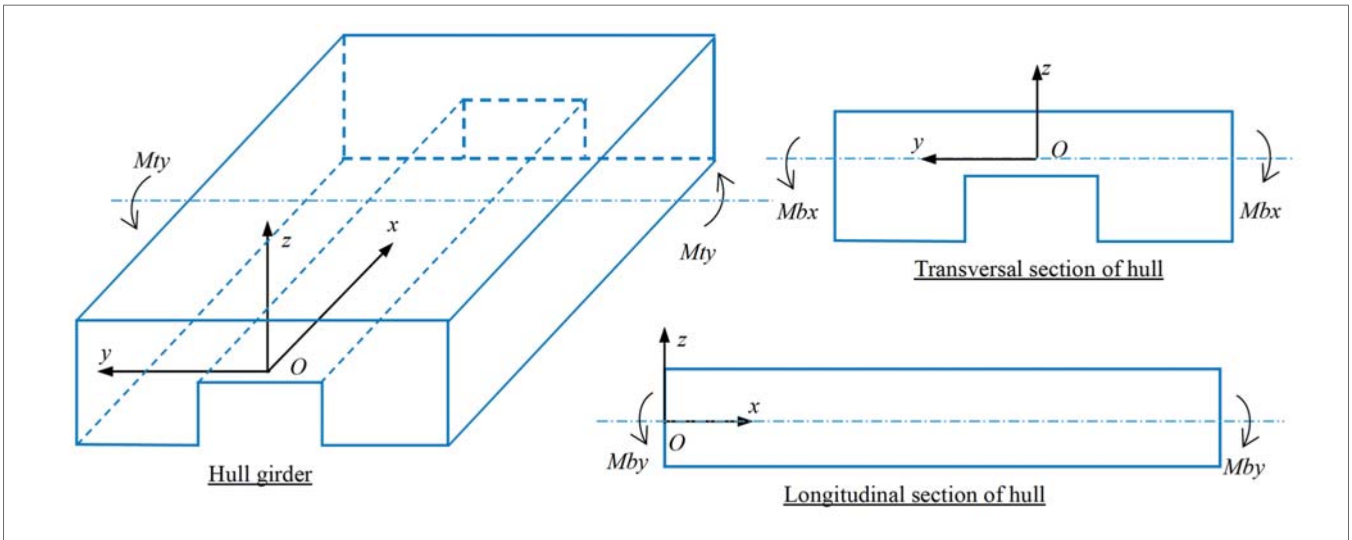


Fig. 1. Definitions of M_{by}, M_{bx} and M_{ty} in drawings

regarded as a reference moment for the load, denoted as the "reference phase".

In the combined condition of Eq. (4), each component load is calculated by multiplying a set of coefficients η_{i1} and η_{i2}, based on long-term prediction. Assuming $\eta_{i1} \geq \eta_{i2}$, we call the load with the larger coefficient η_{i1} the "major control load", whereas the load with the smaller coefficient η_{i2} is called the "minor control load".

During the calculation of the design wave, it is first considered that the major control load needs to reach the objective values $\eta_{i1} M_{by}$. Assuming that the RAO of the ith wave direction and jth circle frequency is defined as $RAO_{by,ij}$, the reference phase is defined as $\varepsilon_{b,by,ij}$. Hence, to achieve its objective value, the wave amplitude should reach the following value:

$$A_{by,ij} = \frac{\eta_{i1} M_{by}}{RAO_{by,ij}} \tag{6}$$

Then, the transient phase of the design wave is satisfied with $\varepsilon_{by,ij} = \varepsilon_{b,by,ij}$.

Because of the phase difference between the major and minor control loads, when the major control load reaches its maximum value, the minor load cannot reach its target value. Instead, a certain reduction is made on the objective value, and the reduction coefficient is expressed as

$$\cos(\varepsilon_{by,ij} - \varepsilon_{b,ty,ij}) = \cos(\varepsilon_{b,by,ij} - \varepsilon_{b,ty,ij}) \tag{7}$$

According to the cosine equation of wave motion, the transfer function $RAO_{ty,ij}$ of the minor control load $\eta_{i2} M_{ty}$ can also be used for a similar reduction; hence, the effective transfer function can be expressed as

$$RAO_{e,ty,\,ij} = RAO_{ty,\,ij} \cdot \cos(\varepsilon_{b,by,ij} - \varepsilon_{b,ty,ij}) \tag{8}$$

Then, to achieve the objective value for the minor control load, the wave amplitude reaches the following value:

$$A_{ty,ij} = \frac{\eta_{i2} M_{ty}}{RAO_{e,ty,ij}} = \frac{\eta_{i2} M_{ty}}{RAO_{ty,ij} \cdot \cos(\varepsilon_{b,by,ij} - \varepsilon_{b,ty,ij})} \tag{9}$$

For each design wave, there is only one amplitude; hence, only when the amplitudes of the major and minor control loads are equal can the design wave be considered to be successful. The design wave can be determined by filtering the wave direction and frequency, which makes $A_{by,ij} \approx A_{ty,ij}$; the closer these values are to each other, the more successfully the design wave can be calculated. Therefore, a difference factor f_{ij} for the wave amplitude level between the major and minor control loads is defined in this paper as follows:

$$f_{ij} = \left| \frac{A_{by,ij}}{A_{ty,ij}} - 1 \right| \tag{10}$$

By calculating all the wave directions and frequencies, all f_{ij} can be obtained. In engineering applications, the design wave can be considered reasonable with values of $f_{ij} \leq 0.05$, and if the value of f_{ij} is very small and closer to 0, the design wave will be more consistent with the requirements of the objective design wave load. In addition, if all the difference factors of the wave amplitude level are large, some additional wave directions and frequencies should be calculated near the minimum value of f_{ij}; then, the new f_{ij} should be compared to obtain the most reasonable design wave.

In the process of searching for design waves, it is necessary to eliminate the design waves that do not conform to reality, according to principle (3). By calculating the design wave under the major and minor load control parameters, two design wave amplitudes can be obtained. Usually, ships travelling in the China Sea area encounter wave heights that are generally less than 11.5 m [3]. Therefore, waves with design wave amplitudes larger than 11.5 m should be excluded; hence, the search process for the design wave in this study is limited as follows:

$$\sqrt{A_{by,ij}{}^2 + A_{ty,ij}{}^2} < 23\sqrt{2}/2 \tag{11}$$

If a combination case of the ith wave direction and jth frequency exceeds the aforementioned limitation, it will be skipped and the wave search will be continued until the minimum f_{ij} is found. The wave angle and circular frequency corresponding to this minimum f_{ij} are the required ones.

Finally, according to principle (2), the amplitude of the designed wave is determined as follows:

$$A = \max(A_{by,ij}, A_{ty,ij}) \tag{12}$$

The flow of this approach is shown in Fig. 2. Input is defined in terms of catamaran geometry, mass distribution, mass moment of inertia, scantlings, speed, sea state, and so on. Output is defined as the parameters of the objective design wave. The various analysis steps are detailed in the following sections.

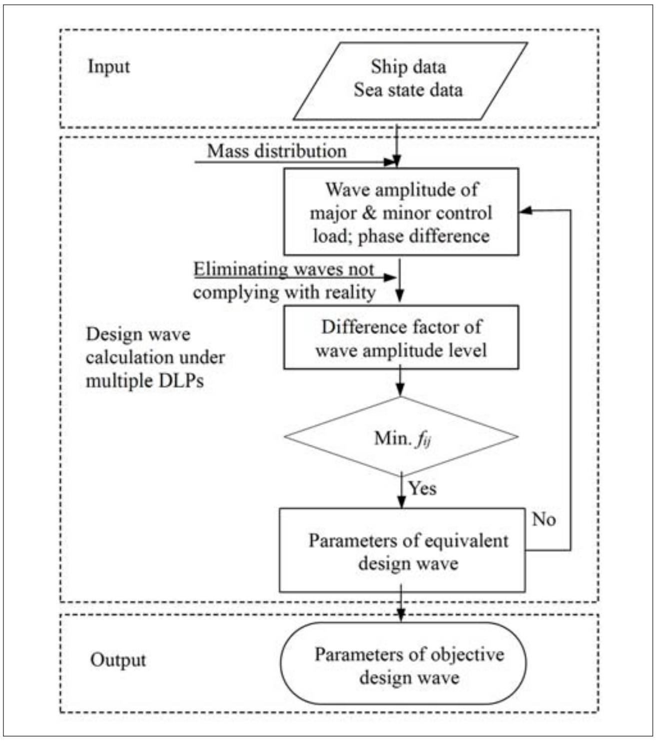


Fig. 2. Load calculation and strength analysis process

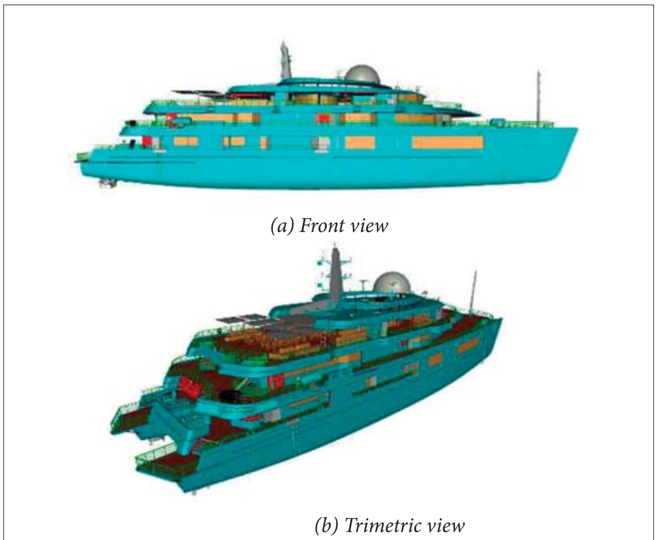
REAL SHIP CALCULATIONS

The main parameters of the passenger catamaran under study are listed in Table 1. The target catamaran cruise ship primarily sails back and forth on fixed routes in the coastal area of the Greater Bay Area (GBA) in China, as shown in the area in the red box in Fig. 3. The appearance of the entire ship is shown in Fig. 4. This ship has four structural decks and one artistic modelling deck (it does not contribute ship strength); various cabins, such as restaurants, theatres, bars, and conference halls, are distributed on each deck.

Tab. 1. Parameters for target catamaran

Principles	Value
Design water line L_{wl}	66.67 m
Moulded breadth B	18.70 m
Moulded depth D	5.20 m
Design draft T	2.55 m
Displacement Δ	1325.9 t
Sailing area –	Coastal/inland of China

Fig. 3. Main navigation area of target catamaran



(a) Front view

(b) Trimetric view

Fig. 4. Appearance of target catamaran

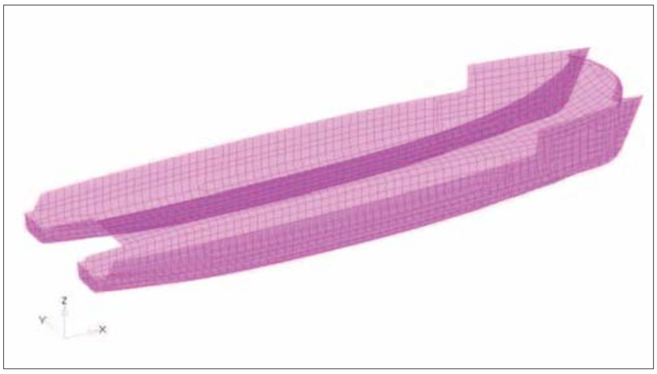


Fig. 5. Hydrodynamic wet surface model

Wave load prediction

Based on the three-dimensional potential flow theory, the hydrodynamic calculation software WALCS, developed by the China Classification Society and Harbin Engineering University, was used in this study to predict the wave load of the target catamaran passenger ship.

Transfer function (RAO) calculation

WALCS was used to build the hydrodynamic wet surface model of the catamaran passenger ship, as shown in Fig. 5, which has a total of approximately 6500 elements. In the load calculation, the ship speed was 0 kn, and the heading angles of the wave were from 0°–180° at intervals of 10°. Thus, there were 19 wave directions in total, including the following wave, beam wave, oblique wave, and heading wave. The incident wave frequency range was 0.2–2.5 rad/s, with a step of 0.1 rad/s. Thus, there were 24 circular frequencies in total. In the setting of the load integral section, 19 transverse sections were set in the longitudinal direction from stern to bow, and 9 longitudinal sections were set in the transverse direction from port to starboard. Moreover, the transverse sections in the longitudinal direction were distributed evenly, whereas the longitudinal sections in the transverse direction were denser at the cross-structure because the strength of the cross-structure connecting the catamaran body is the main problem in structural evaluation.

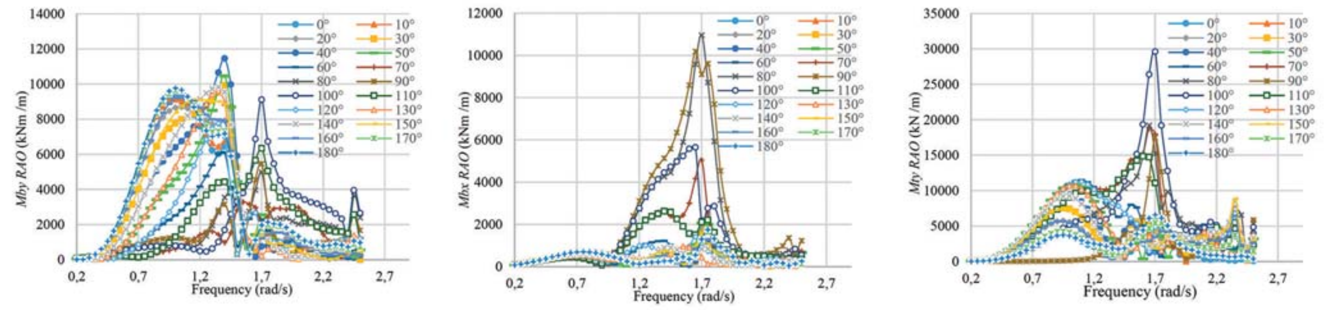
The mass distribution of the catamaran was adopted in the form of mass blocks; the entire ship was divided into 38 mass blocks by transversal and longitudinal cutting. The centre of gravity of each section was adjusted according to the actual data.

Through calculation, the RAOs of the loads under the full-load condition were obtained as shown in Fig. 6.

Short- and long-term prediction

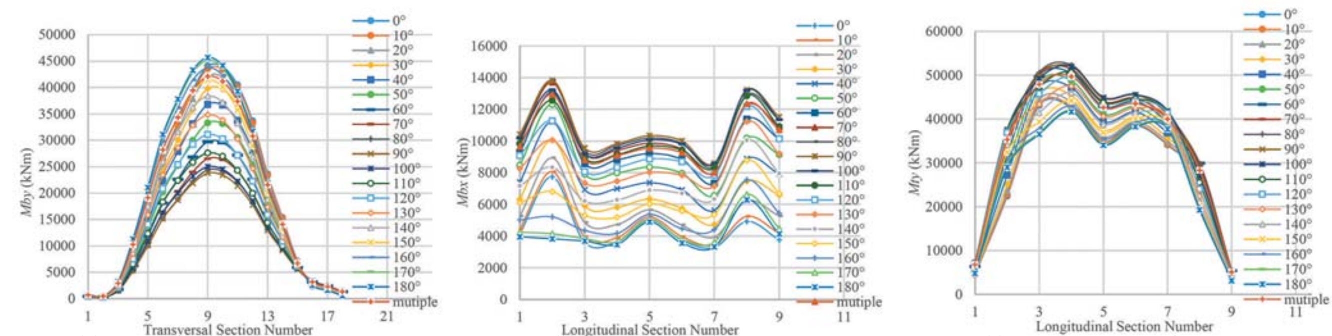
China's coastal water and the northwest Pacific are divided into several regions in the *Northwest Pacific Ocean Wave Statistics* set, and the navigable area for the target catamaran is S1 [17]. Therefore, the wave statistics data of this region were selected as the input data for the long-term calculation. In the calculation, the P-M double parameter spectrum was selected as the wave spectral density function.

According to the principle of long-term prediction, it is calculated at the level of 10^{-8} exceeding probability for the target



(a) Longitudinal bending moment *(b) Transverse bending moment* *(c) Transverse torsional moment*

Fig. 6. Pure response in regular waves (RAO)



(a) Longitudinal bending moment *(b) Transverse bending moment* *(c) Transverse torsional moment*

Fig. 7. Extreme wave loads on hull girder for long-term prediction

catamaran, and extreme wave loads acting on the hull girder are obtained as shown in Fig. 7. The maximum longitudinal bending moment is 42050 kNm, the maximum transverse bending moment is 12950 kNm, and the maximum transverse torque is 49720 kNm.

DESIGN WAVE DETERMINATION

Based on the method above, the design waves under combined bending–torsion conditions were calculated, and the design wave calculation under the following condition was taken as the research object for detailed analysis:

$$O_i = 0.8M_{by} + 0.6M_{ty}$$

The load control parameters of the design wave under this formula are the longitudinal bending moment and transverse torque. Based on the long-term prediction results, the maximum value of M_{by} is 42050 kNm, and the maximum value of M_{ty} is 49720 kNm. Therefore, the target values of the major and minor load control parameters were 33640 kNm and 29832 kNm, respectively.

Then the design wave amplitude under the major load control parameters and the minor load control parameters are calculated in every wave direction and frequency. After eliminating design waves that did not conform to the actual wave, the difference factor of the wave amplitude level was calculated as shown in Table 2, where the minimum value of f_{ij} was 0.00. The wave amplitude, circle frequency and wave direction angle corresponding to this value are the wave amplitude, circle frequency and wave direction angle of the required design wave. The phase was 329.2°, corresponding to the major load control parameters. According to the results of the design wave, the hull structure reaches the objective load in the 140° oblique wave state. It can be seen that the objective combination wave condition of bending and torsion was generated.

The rule requires eight combined bending–torsion design waves, as follows:

$$\begin{aligned} O_3 &= 0.8M_{bx-in} + 0.6M_{ty} & O_9 &= 0.8M_{by-hog} + 0.6M_{ty} \\ O_4 &= 0.8M_{bx-out} + 0.6M_{ty} & O_{10} &= 0.8M_{by-sag} + 0.6M_{ty} \\ O_5 &= 0.6M_{bx-in} + 0.8M_{ty} & O_{11} &= 0.6M_{by-hog} + 0.8M_{ty} \\ O_6 &= 0.6M_{bx-out} + 0.8M_{ty} & O_{12} &= 0.6M_{by-sag} + 0.8M_{ty} \end{aligned} \tag{13}$$

These design waves were calculated based on the aforementioned methods. All the parameters of the eight design waves are shown in Table 3.

DISCUSSION AND CONCLUSION

To verify the accuracy of the design wave calculation method under multi-load control parameters in this study, a load analysis of the aforementioned eight bending–torsional

Tab. 2. Difference factor for wave amplitude level

f_{ij}											
ω/β	0°	… …	90°	100°	110°	120°	130°	140°	150°	… …	180°
0.20	0.28	… …	--	--	--	--	--	--	--	… …	--
… …	… …	… …	… …	… …	… …	… …	… …	… …	… …	… …	… …
0.65	0.28	… …	--	--	--	--	--	--	0.13	… …	--
0.70	0.30	… …	--	--	--	--	--	0.70	0.09	… …	--
0.75	0.32	… …	--	--	--	--	1.61	0.64	0.05	… …	--
0.80	0.35	… …	--	--	--	--	1.48	0.57	0.00	… …	--
0.85	0.39	… …	--	--	--	--	1.33	0.48	0.06	… …	--
0.90	0.45	… …	--	--	--	2.49	1.13	0.36	0.15	… …	--
0.95	0.52	… …	--	--	--	1.95	0.86	0.19	0.26	… …	--
1.00	0.62	… …	--	--	--	1.39	0.56	0.00	0.39	… …	--
1.05	0.73	… …	--	--	--	0.89	0.25	0.21	0.53	… …	--
1.10	--	… …	--	--	--	0.51	0.03	0.41	0.67	… …	--
1.15	--	… …	--	--	0.91	0.24	0.26	0.59	--	… …	--
1.20	--	… …	--	--	1.02	0.10	0.42	0.73	--	… …	--
1.25	--	… …	--	--	1.21	0.06	0.52	--	--	… …	--
1.30	--	… …	--	--	1.46	0.08	0.58	--	--	… …	--
1.35	--	… …	--	--	1.62	0.15	0.62	--	--	… …	--
1.40	--	… …	--	--	1.08	0.24	0.69	--	--	… …	--
1.45	--	… …	--	3.01	0.04	0.30	--	--	--	… …	--
1.50	0.40	… …	--	3.44	3.09	--	--	0.46	--	… …	--
1.55	--	… …	--	3.47	2.45	--	--	--	--	… …	--
1.60	--	… …	1.65	3.73	1.68	--	--	--	--	… …	--
1.65	--	… …	1.82	3.26	1.26	--	--	--	--	… …	--
1.70	--	… …	2.50	2.54	0.54	--	--	--	--	… …	--
… …	… …	… …	… …	… …	… …	… …	… …	… …	… …	… …	… …
2.00	--	--	--	--	--	--	--	--	--	--	--

Note: “--” indicates that the difference factor is not calculated for wave amplitudes that do not conform to the actual situation

Tab. 3. Parameters of design waves

Case	Amplitude A(m)	Heading angle θ (°)	Phase ε (°)	Frequency ω (rad/s)
3	8.43	110	329.2	1.80
4	8.43	110	149.2	1.80
5	5.03	110	42.7	1.15
6	5.03	110	–138.3	1.15
9	4.75	140	330.5	1.00
10	4.75	140	150.5	1.00
11	6.79	40	6.6	0.75
12	6.79	40	–173.40	0.75

Tab. 4. Comparison between the actual load value and the objective value of each designed wave components

		Object1 kNm	Object2 kNm	Actual1 kNm	Actual2 kNm	ER.1	ER.2
O_3	$0.8M_{bx\text{-}in} + 0.6M_{ty}$	10360	29832	10390	29720	0.29%	−0.38%
O_4	$0.8M_{bx\text{-}out} + 0.6M_{ty}$	−10360	29832	−10390	29720	0.29%	−0.38%
O_5	$0.6M_{bx\text{-}in} + 0.8M_{ty}$	7770	39776	7777	41200	0.09%	3.58%
O_6	$0.6M_{bx\text{-}out} + 0.8M_{ty}$	−7770	39776	−7777	41200	0.09%	3.58%
O_9	$0.8M_{by\text{-}hog} + 0.6M_{ty}$	33640	29832	33640	29810	0.00%	−0.07%
O_{10}	$0.8M_{by\text{-}sag} + 0.6M_{ty}$	−33640	29832	−33640	29810	0.00%	−0.07%
O_{11}	$0.6M_{by\text{-}hog} + 0.8M_{ty}$	25320	39776	25430	39760	0.43%	−0.04%
O_{12}	$0.6M_{by\text{-}sag} + 0.8M_{ty}$	−25320	39776	−25430	39760	0.43%	−0.04%
					Average:	0.20%	0.77%

Fig. 8. Aerial photos of actual ship sailing

combination design waves was carried out on the target hull, and the component loads obtained are shown in Table 4. By comparing each component load under every condition, it was found that the errors were very small, and the average error of each loading condition was within 1%. Therefore, the actual design wave load is almost the same as the object, which is sufficient to show that the method described in this paper can effectively calculate the design wave parameters under multi-load control parameters.

Based on the calculation of the aforementioned design waves and the strength analysis of the entire ship, the structural strength assessment of the target catamaran has been recognised by the China Classification Society. In addition, this target ship was delivered in June 2021, and it has received good feedback and acquired a good reputation. Fig. 8. shows aerial photographs of the actual ship sailing.

Above all, this paper proposes a design wave calculation approach for catamarans under multiple load control parameters. By defining the difference factor of the wave amplitude level in this study, the design wave for the catamaran can be calculated effectively under multiple load control parameters; the average error is less than 1% compared with the objective load components. Hence, a reasonable design wave can be obtained using this approach.

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CONTACT WITH THE AUTHORS

Shenwei Ge
e-mail: geshenwei@jssc.edu.cn
Merchant Marine College
of Shanghai Marine University, Shanghai;
Nantong Shipping College, Jiangsu Nantong
China

Ji Zeng
e-mail: zengji@shmtu.edu.cn
Ocean Science and Engineering College
of Shanghai Marine University, Shanghai
China

Bowen Jin
e-mail: jinbowen@shmtu.edu.cn
Merchant Marine College
of Shanghai Marine University, Shanghai
China

Wen Zhou
e-mail: Zhouwen@cmhk.com
China

Xiaolong Qin
e-mail: Qinxiaolong@cmhk.com
China Merchants Cruise Shipbuilding Co. LTD.
China