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Time-domain numerical and segmented ship model experimental analyses of hydroelastic responses of a large container ship in oblique regular waves

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ABSTRACT

Real sea conditions are characterized by multidirectional sea waves. However, the prediction of hull load responses in oblique waves is a difficult problem due to numeral divergence. This paper focuses on the investigation of numerical and experimental methods of load responses of ultra-large vessels in oblique regular waves. A three dimensional nonlinear hydroelastic method is proposed. In order to numerically solve the divergence problem of time-domain motion equations in oblique waves, a proportional, integral and derivative (PID) autopilot model is applied. A tank model measurement methodology is used to conduct experiments for hydroelastic responses of a large container ship in oblique regular waves. To implement the tests, a segmented ship model and oblique wave testing system are designed and assembled. Then a series of tests corresponding to various wave headings are carried out to investigate the vibrational characteristics of the model. Finally, time-domain numerical simulations of the ship are carried out. The numerical analysis results by the presented method show good agreement with experimental results.

1. Introduction

The 21 st century is the century of ocean. In the wide ocean there exist very rich resources such as sea chemical elements, ocean biological resources and tidy energy, etc. In the utilization of ocean spaces and exploitation of ocean resources, the demands for high-speed transportation and huge dimensions of ships increase rapidly. In order to ensure the safety of ultra-large vessels at sea, their structural responses in the sea are of particular importance. This is fundamentally a hydroelasticity problem. Since hydroelasticity is important for evaluating the behavior of ultra-large vessels with increasing tonnage in extreme sea states, many experts have made comprehensive and in-depth studies on hydroelastic method in head sea conditions recently [1–4].

Accurate prediction of ship motion and structural load responses is important for both ship design and strength check. The majority of the existing methods focus on the investigation of ship load responses in head seas. Due to reasons such as divergence in horizontal motion and bending-torsional coupling effects, the simulation of ship loads in oblique regular waves is much more complex than that in head waves. In fact, the actual ocean, where ships sail, is characterized by multidirectional sea waves. Therefore, it has very practical significance to research and develop a kind of prediction method for ship load responses under arbitrary heading angles. Unlike hydroelastic studies for head waves, the interactions of elastic structures with oblique waves that involve time-domain analysis have scantly been studied in the recent years [5–8]. Fukasawa [9] utilized the digital filter technique to eliminate the divergence of ship horizontal motion. Although the divergence for sway and yaw could be avoided by that mean, it is not neither convenience nor time-saving. Chen et al. [10] analyzed second-order hydroelastic responses of a floating plate in multidirectional waves. In their article, by use of phase angles, the effects of the direction of multidirectional incident wave are taken into account.

However, for prediction of hydroelastic responses of ships in oblique waves, the divergence problem of ship horizontal motion (sway and yaw) has rarely been solved. When the ship horizontal motion is simulated numerically in time domain, there is no horizontal restoring force. So once there is any drift phenomenon, no mechanism could make the ship back to the original track and

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course, and the steady state solutions cannot be obtained even in the case of regular waves. The main cause is due to lack of rubber, so ship motion becomes unstable under Munk moment action. The solutions of horizontal motions are drifting in process of time steps, and this drifting affects other modes of motions badly [11]. These aspects bring great difficulties for the accurate prediction of ship load responses in oblique waves.

In addition, the growth in size of ships increases the dynamic response of the structure on impulsive wave loads (whipping). Whipping, which is a transient vibration induced by ship girder slamming, increases both the extreme load and fatigue damage on the structure. For extreme sea states, hull girder vibration due to impact force must be taken into account for the study of load responses of ultra-large vessels. Segmented model experiments constitute an invaluable tool in the field of ship dynamic response [12–14]. Though the test method is becoming more and more mature, due to the limitations of current test technique and test site, most segmented ship model experiments were performed in head waves, which restricts the development of estimation of actual sea states to some extent.

Therefore, this paper is aimed at accurately predicting ship load responses in oblique regular waves both numerically and experimentally. The three dimensional (3-D) nonlinear time-domain hydroelasticity theory in head waves has been completed in the authors' previous work [15]. On this basis, a 3-D nonlinear time-domain hydroelasticity theory in oblique regular waves is adopted in Section 2, with an artificial auto-pilot introduced to address the divergence problem of the ship horizontal motion in oblique waves. Unlike traditional methods that phase angles or encounter frequencies are used to take the effect of wave headings into consideration, the method presented in this paper solves divergence problem of hull time-domain equations of motion in oblique waves fundamentally. An alternative approach to the conventional head wave test measurement is to perform tests by segmented model measurement in oblique waves. The model structural design of a 13000TEU container ship and the oblique wave testing system are introduced in Section 3. Then experimental procedures and results are described and analyzed. From the comparison of the calculated data with test data, the effectiveness of PID autopilot model and accuracy of the time-domain hydroelastic method in the paper are analyzed and demonstrated in Section 4.

2. Numerical simulations

2.1. Nonlinear hydroelasticity theory

Transient phenomena observed in fluid-structure interactions of ships and other floating structures, such as whipping induced by slamming, can only be studied in time domain. Especially the problems involving nonlinear hydrodynamic actions have to be solved in time domain or approximately by adding certain nonlinear terms to the linear equations. In this paper, the hydroelastic motion equation in the time domain is expressed as follows:

$$\begin{aligned}&([a]+[\mu])\ddot{p}_r(t)+[b]\,\dot{p}_r(t)+\int_0^t[K(\tau)]\dot{p}_r(t-\tau)d\tau+\\&([c]+[C])p_r(t)=F_I(t)+F_D(t)+F_{slam}(t)\end{aligned}\tag{1}$$

where, $[a]$, $[b]$ and $[c]$ denote generalized structural mass matrix, generalized structural damping coefficient matrix and generalized structural stiffness matrix respectively; *P(t)* is the mode principal coordinates;

$[\mu]$ denotes the added mass, which is a $r\times r$ order constant matrix; $[K(\tau)]$ denotes retardation function, which is also a $r\times r$ order matrix dependent on time τ. $[K(\tau)]$ depends on the shape of the underwater portion of hull and time interval, and reflects characteristic of wave damping and hydrodynamic. $r=1\sim6$ denotes motion modes of rigid hull; $r\geq7$denotes motion modes of flexible hull;

$F_I(t)$ is incident wave force; $F_D(t)$ is diffraction wave force; $F_{slam}(t)$ is slamming force.

If the ship motion in waves can be regarded as a kind of forced vibration under the action of wave excitation, Eq. (1) can be expressed in the following form:

$$[a]\,\ddot{p}_r(t)+[b]\,\dot{p}_r(t)+[c]\,p_r(t)=\left\{F(t)\right\}\tag{2}$$

where $F(t)$ is the nonlinear fluid forces. The fluid forces $\left\{F(t)\right\}$ acting on the ship at a given time can be expressed as:

$$\begin{aligned}\left\{F(t)\right\}&=\left\{F_S(t)\right\}+\left\{F_I(t)\right\}+\left\{F_D(t)\right\}\\&+\left\{F_R(t)\right\}+\left\{F_{slam}(t)\right\}\end{aligned}\tag{3}$$

where $F_S(t)$ is the hydrostatic restoring force;$F_R(t)$ is the radiation force.

Combined with 3-D wet surface mesh of elastic hull, the velocity potential of flow field is solved by the Source-Sink Distribution Method. Then all the fluid forces in Eq. (3) are obtained. In this paper, in order to take more factors into the calculation to meet the requirement of practical engineering, some nonlinear terms, such as nonlinear restoring force, nonlinear incident wave force and slamming force, are added to the motion Eq. (1) artificially.

2.1.1. Calculation of nonlinear fluid forces

The hydrostatic restoring forces have been directly computed through the composition method, i.e., integrating the hydrostatic pressure from instantaneous wet surface *S* (t) and hull gravity.

$$F_S(t)=-\rho g\sum_{k=1}^{m}p_{ka}\iint_{S(t)}\vec{n}\cdot\vec{u}_r w_k ds-F_G(r=1,2,\cdots m)\tag{4}$$

$$F_G=\int_L F_g(x)w_r(x)dx_b\tag{5}$$

where w_k is the k-th mode of vertical displacement; $F_g(x)$is weight collection degree of each station.

By use of the interception of instantaneous grid, the contributions of the incident wave force and diffraction wave force can be calculated directly from the incident potential and diffraction potential.

$$\begin{cases}F_I(t)=-\rho\zeta_a\displaystyle\iint_{S(t)}\vec{n}\cdot\vec{u}_r(i\omega-U\frac{\partial}{\partial x})\phi_0 ds\\F_D(t)=-\rho\zeta_a\displaystyle\iint_{S(t)}\vec{n}\cdot\vec{u}_r(i\omega-U\frac{\partial}{\partial x})\phi_d ds\end{cases}(r=1,2,\cdots m)\tag{6}$$

where, ϕ_0 and φ_d are instantaneous incident wave potential and diffraction wave potential under unit wave amplitude, respectively; ζ_a is wave amplitude; $\vec{n}$ is the normal vector, which is defined positive when pointing into body from the boundary surface; $\vec{u}_r$ is the r-th principal modes of the structure.

The momentum slamming theory is used to predict slamming loads in this paper:

$$F(x,t)=-\{\frac{d}{dt}[m(x,t)\frac{d}{dt}Z_R(x,t)]-\rho gs(x,t)\}\tag{7}$$

where, *m(x,t)* is the instantaneous added mass; $Z_R(x,t)$ is vertical relative displacement of ship motion to wave; *s(x,t)* is instantaneous sinking area.