CFD Computation of Wave Forces and Motions of DTC Ship in Oblique Waves

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Numerical simulations of the Duisburg Test Case (DTC) ship free to heave, roll, and pitch motions in oblique waves are presented. The computations are carried out by an in-house computational fluid dynamics (CFD) solver, naoe-FOAM-SJTU, based on volume of fluid (VOF) and overset grid methods. An open source library, waves2Foam, is used to generate desired wave conditions and prevent the wave reflecting internally from the computational domain. This study focuses on the short wave; therefore, only one wave length is chosen. The diffraction and radiation effects have significant influence on the nonlinearity of wave forces. The results of longitudinal and lateral mean drift forces reach their maximum values at headings of 60° and 90°, respectively, which is caused by the increment of ship motions and enhancement of wave diffraction. The time history and Fast Fourier transform results of longitudinal and lateral forces show a strong nonlinear property. Then the analysis on ship motions and wave patterns demonstrates that the nonlinearity has a connection with the ship motion and complex wave diffraction and wave slamming. The results of ship motions show that the heave and pitch motions are mainly dominated by wave frequency, whereas the roll motion is correlated with its natural frequency.

INTRODUCTION

In response to global warming, the International Maritime Organization (IMO) implemented the Energy Efficiency Design Index (EEDI) effective January 2013. The EEDI requires each newly-built vessel to meet the regulation for vessel emissions. To cut down emissions, some ship designers and builders choose to lower the installed power and ship’s speed instead of putting effort to optimize ship’s speed-powering performance. This leads to rising concerns regarding the sufficiency of propulsion power and steering devices to maintain maneuverability of ships in adverse conditions. It is evident that when a ship is operating in adverse conditions, the mean drift forces and moments will act on the ship and change its course. Therefore, it is necessary to develop suitable tools to effectively evaluate mean drift forces and moments and assess ship maneuverability in waves.

There are many previous studies that focus on mean drift forces, most of which used the potential theory. Grue and Palm (1993) discussed the effect of the steady second-order velocities on the mean drift forces and moments acting on the marine structure in waves and a (small) current. Later, Hermans (1999) presented numerical results for two classes of tankers—namely, a very large crude carrier (VLCC) and a liquefied natural gas carrier (LNG)—and a semisubmersible and compared them with experimental data obtained at the Maritime Research Institute in the Netherlands (MARIN). Tanizawa et al. (2000) applied linear and fully nonlinear numerical wave tanks (NWTs) to study wave drift force acting on a two-dimensional Lewis-form body in a finite-depth wave flume. More recently, Liu and Papanikolaou (2016) worked on the fine-tuning of the far-field method using the Kochin function for predicting the added resistance of ships in oblique waves.

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KEY WORDS: Wave forces, naoe-FOAM-SJTU solver, overset grid, oblique waves, DTC ship model.
the flow separation phenomena behind the hull when the ship is operating in oblique waves. Wang et al. (2017) conducted numerical simulations of a free running ship in different waves (i.e., head wave, bow quartering wave, and beam wave) and found that the CFD method is reliable in predicting the ship’s hydrodynamic performance and maneuverability.

The focus of this work lies on numerical predictions on drift forces of the DTC in high and steep regular head, following, and oblique waves. These climates were selected to check the capability of numerical models to capture the nonlinear effect. In the present work, CFD solver naoe-FOAM-SJTU (Shen and Wan, 2011; Shen and Wan, 2012; Cao et al., 2013) is used to conduct the numerical investigation. The experimental data (Sprenger et al., 2016) are provided by the Norwegian Marine Technology Research Institute (MARINTEK). After the validation is done, it is possible for us to implement this numerical model to conduct the maneuverability computation of ships in adverse wave conditions. For example, computation can be performed for the zigzag maneuvers and turning circle maneuvers in high and steep waves. On this basis, it is possible to develop a procedure to perform a holistic assessment of ship performance and to formulate minimum powering requirements to ensure safe ship operation in adverse weather conditions.

The rest of this paper is organized as follows: In the next section, a brief introduction of the numerical methods is given. The geometry and conditions of the computational cases follow. Then, the computational results and the comparison with experimental data are discussed. Finally, a summary of the paper is presented.

NUMERICAL METHODS

Governing Equations

The incompressible Navier–Stokes equations are the governing equations, which can be written as follows:

\[ \nabla \cdot \mathbf{U} = 0 \quad (1) \]

\[ \frac{\partial (\rho \mathbf{U})}{\partial t} + \nabla \cdot (\rho \mathbf{U} \mathbf{U}) = -\nabla p_d - \rho \mathbf{g} \cdot \mathbf{x} + \nabla \cdot (\mu_d \nabla \mathbf{U}) \quad (2) \]

where \( \mathbf{U} \) is fluid velocity field and \( \mathbf{U}_g \) is the grid velocity; \( p_d = \rho - \rho g \cdot \mathbf{x} \) is the dynamic pressure, obtained by subtracting the hydrostatic component from the total pressure \( \rho \); \( \rho \) is the mixture density; \( \mathbf{g} \) is the gravity acceleration; and \( \mu_d \) is dynamic viscosity.

Volume of Fluid Method

The volume of fluid (VOF) method with artificial compression is used to capture the air-water interface. Details of the VOF simulation procedure in OpenFOAM are described in Rusche (2002). A VOF method with a bounded compression technique is applied to capture the free-surface interface. The transport equation is expressed as

\[ \frac{\partial \alpha}{\partial t} + \nabla \cdot (\rho \mathbf{U} (U - \alpha) + \nabla \cdot (\mathbf{U}_g (1 - \alpha) \alpha) = 0 \quad (3) \]

where \( \alpha \) is the volume of the fraction, indicating the relative proportion of fluid in each cell, and its value is always between zero and one:

\[
\begin{align*}
\alpha &= 0 \quad \text{air} \\
\alpha &= 1 \quad \text{water} \\
0 &< \alpha < 1 \quad \text{interface}
\end{align*}
\]

Fig. 1 Top view of the computational domain with wave generation zone

In Eq. 3, \( U_g \) is the velocity field used to compress the interface, and it takes effect only on the surface interface as a result of the term \((1 - \alpha)\alpha\). The expression of this term can be found in Berberović et al. (2009).

Overset Grid Technique

Overset grid is a grid system composed of multiple blocks of overlapping structured or unstructured grids. In a full overset grid system, a complex geometry is decomposed into a system of geometrically simple overlapping grids. Boundary information is exchanged between these grids via interpolation of the fluid variables. In this way, the overset grid method removes the restrictions of the mesh topology among different objects and allows grids to move independently within the computational domain, and it can be used to handle large-amplitude motions in the field of ship and ocean engineering. The most critical objective in the overset grid is the accomplishment of information exchange between grid blocks. Based on the numerical methods from OpenFOAM, including the cell-centered scheme and unstructured grids, Suggar++ (Noack et al. 2009) is utilized to generate the domain connectivity information (DCI) for the overset grid interpolation in the solver naoe-FOAM-SJTU. In this way, the solver can handle arbitrary motions in the simulation. For example, Wang et al. (2016a) used the same solver to simulate ship self-propulsion with moving rudders and rotating propellers. Furthermore, Wang et al. (2016b) carried out turning circle simulation using the same solver. In those studies, the moving rudders and rotating propellers were handled by the dynamic overset grid method. More details about the overset grid technique can be found in Shen et al. (2015).

Wave Generation

In general, the incoming wave is generated by imposing the boundary conditions on the tank inlet. To absorb the wave reflection from the outlet, a damping zone is set at the end of the tank to avoid the wave reflection. This conventional wave generation method suffers reflection between the inlet and structure, especially in the case free of current. One solution is to extend the distance between the inlet and object. However, the disadvantage of this solution is the accompanying smaller time step and increasing computational time. To overcome this problem, the open source library waves2Foam (Jacobsen et al., 2012) is imposed in our solver. In the work of Jacobsen et al. (2012), a method called the relaxation technique is developed to achieve the functionality of both wave generation and absorption in a uniform way and to avoid wave reflection from the boundaries. In this study, an annular relaxation zone is set up to generate waves with different
heading angles (Fig. 1). The variables $\phi$, such as $U$ or $p$, inside the relaxation zone can be expressed as follows:

$$\phi = \alpha_R \phi_{\text{computed}} + (1 - \alpha_R) \phi_{\text{target}}$$  \hspace{1cm} (5)

where the variable $\phi_{\text{target}}$ is a function of space and time known from the Stokes wave theory, and $\phi_{\text{computed}}$ is the variable computed from the FVM. $\alpha_R$ is always 1 at the interface between the non-relaxed part of the computational domain and the relaxation zone and is 0 on the inlet boundary. And $\alpha_R$ smoothly varies from 1 to 0 in the relaxation zone. Such distribution of $\alpha_R$ makes the $\phi$ relaxed by the value of $\phi_{\text{computed}}$ and $\phi_{\text{target}}$ in the relaxation zone. In this way, the relaxation zone achieves the functionality of both wave generation and absorption in a uniform way and avoids the wave reflection between the boundaries.

**GEOMETRY AND CONDITIONS**

The model chosen in this study is the DTC container, which has comprehensive experimental data from MARINTEK. The arrangement of model tests can be found in Sprenger et al. (2016). DTC is a generic post-Panamax 14000 TEU (20-foot equivalent units) container ship developed at the Institute of Ship Technology, Ocean Engineering and Transport Systems (ISMT) of the University of Duisburg-Essen. Its geometry is shown in Fig. 2, appended with a twisted rudder with a Costa bulb and a NACA 0018 base profile. The DTC principal particulars are listed in Table 1 in model scale.

On each side of the vessel, a segmented bilge keel is attached symmetrically to the hull along the parallel middle body, but the keels are not considered in this study. The reason for this simplification is that the existence of keels increases the complexity of mesh generation.

This study focuses on the mean drift forces acting on the DTC hull in different heading angles $\mu$; therefore, only one wave height and wave period is chosen, while $\mu$ changes every 30° from 0° to 180° ($\mu = 0°$ refers to head waves). Wave height $H$ is 0.157 m and wave period is 1.128 s in model scale. In addition, it should be mentioned that there is a difference in the setup of ship motion between the experiment and calculation. The setup in the simulation makes the hull free to heave, roll, and pitch, and it fixes the hull’s surge, sway, and yaw motions; the experiment applied a soft spring mechanism to confine the nonrestoring motions. This compromise will cause a difference between experimental and computational results, which will be discussed later.

There are two blocks of grids in the computational domain. One is called the background grid and the other is the hull grid, which follows the motions of the hull. The design of the overset grid system for the DTC is shown in Fig. 3. The radius of the background grid is 2.5 times of ship length. The hull grid resolves the flow around the ship, and the cylindrical background grid accommodates the far-field boundary conditions. The grids interpolated by Suggar++ are illustrated in Fig. 4. Local refinements are applied on the free surface and the region around the hull and the rudder. The two grids used in this paper are separately generated by snappyHexMesh, an automatic mesh-generation utility provided by OpenFOAM. This utility generates mesh on an original Cartesian background mesh, splitting hexahedral cells into split-hex cells. The view of computational mesh can be found in Fig. 5. The sizes of these grids are listed in Table 2. As the table shows, about 2 million cells are used in background grid. Most of these cells distribute around the free surface to make sure that waves do not decay before arriving at the hull.

All computations are performed with 40 processors, one of which is assigned to Suggar++ for DCI computation. The time step is $\Delta t = 1 \times 10^{-3}$ s for $\mu$ less than or equal to 90° and $\Delta t = 5 \times 10^{-4}$ s for $\mu$ greater than 90°. This makes sure the interface Courant number is less than 1, which is critical to ensure the robustness of numerical simulation and yields well-converged results. The clock time per time step is about 14 s. It is quite effective when comparing with deforming mesh technology, which spends 21 s per step. For the deforming mesh, it costs plenty of time to calculate the mesh motion at each time step.

**Table 1** Principal particulars of the DTC

<table>
<thead>
<tr>
<th>Main particular</th>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scale factor</td>
<td>$\lambda$</td>
<td>1.6365</td>
</tr>
<tr>
<td>Length between perpendiculars</td>
<td>$L_{pp}$</td>
<td>5.57 m</td>
</tr>
<tr>
<td>Molded breadth</td>
<td>$B$</td>
<td>0.80 m</td>
</tr>
<tr>
<td>Draft</td>
<td>$T$</td>
<td>0.23 m</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>$CB$</td>
<td>0.661</td>
</tr>
<tr>
<td>Displacement</td>
<td>$\Delta$</td>
<td>672.7 kg</td>
</tr>
<tr>
<td>Wetted hull surface</td>
<td>$S_w$</td>
<td>5.438 m$^2$</td>
</tr>
<tr>
<td>Longitudinal distance of the center of gravity from the aft perpendicular</td>
<td>$LCG$</td>
<td>2.74 m</td>
</tr>
<tr>
<td>Vertical distance of the center of gravity from the base line</td>
<td>$VCG$</td>
<td>0.39 m</td>
</tr>
<tr>
<td>Transverse metacentric height</td>
<td>$GM_T$</td>
<td>0.08 m</td>
</tr>
<tr>
<td>Longitudinal gyroradius</td>
<td>$r_{xx}$</td>
<td>0.31 m</td>
</tr>
<tr>
<td>Transverse gyroradius about the gravity center</td>
<td>$r_{yy}$</td>
<td>1.37 m</td>
</tr>
<tr>
<td>Vertical gyroradius about the gravity center</td>
<td>$r_{zz}$</td>
<td>1.37 m</td>
</tr>
</tbody>
</table>

![Fig. 2 View of DTC hull (top) and the rudder (bottom)](image-url)

![Fig. 3 Design of the overset grid system for DTC](image-url)
Fig. 4 Mesh views for vertical plane at $y = 0$ (red for background mesh and green for hull mesh)

(a) Global mesh (split along central fore-and-aft vertical plane)
(b) Local mesh (Bow)
(c) Local mesh (Stern)

Fig. 5 Computational mesh

![Computational mesh](image)

Table 2 Grid sizes for computations of mean drift forces in regular waves

<table>
<thead>
<tr>
<th>Hull</th>
<th>Background</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,461,919</td>
<td>2,226,624</td>
<td>3,688,543</td>
</tr>
</tbody>
</table>

RESULTS

Mean Drift Forces in Regular Waves

Figure 6 shows the coefficients of longitudinal mean drift forces, lateral mean drift forces, and mean yaw drift moments for the $\mu$ in the range of 0° to 180°. The definitions of these coefficients are as follows:

$$C_{fl} = \frac{T_l}{\rho \cdot g \cdot (B^2/L_{pp}) \cdot (H/2)^2}$$  \hspace{1cm} (6)

$$C_{fy} = \frac{T_y}{\rho \cdot g \cdot (B^2/L_{pp}) \cdot (H/2)^2}$$  \hspace{1cm} (7)

$$C_{fzm} = \frac{M_z}{\rho \cdot g \cdot B^2 \cdot (H/2)^2}$$  \hspace{1cm} (8)

where $\bar{T}_l$, $\bar{T}_y$, and $\bar{M}_z$ are the time average of longitudinal forces, lateral forces, and yaw moments, respectively. Results are compared with experimental measurements from MARINTEK for the SHOPERA Benchmark Workshop (Sprenger et al., 2016). The results of longitudinal mean drift forces obtained by potential theory (Liu and Papanikolaou, 2016) are also present in Fig. 6a.

In general, numerical results of $C_{fl}$ have the same trend as the experimental data. The comparison in Fig. 6a shows that the CFD solver is more precise than the potential method. $C_{fl}$ reaches its maximum when $\mu = 60^\circ$ as the incident angle of waves is nearly perpendicular to the forward shoulder of the hull. Consequently, the diffraction increases and the radiation also become stronger.
because of the increment of ship motions. The enhancement of diffraction and radiation makes the value of $C_{fx}$ much larger. Such diffraction and radiation effects will be discussed in the following section. In oblique waves, almost all the CFD results of $C_{fx}$ are less than the experimental fluid data (EFD) except for 30° and 150°. These errors are caused by the different arrangement between the experiment and computation. As mentioned before, the experiments restrained the hull’s surge, sway, and yaw motions by a soft-mooring system; in present computations, the surge, sway, and yaw are directly fixed. It is evident that fixing the non-restoring motion will decrease the corresponding radiation effects.

The calculated mean drift forces $C_{fy}$ and their measured values are shown in Fig. 6b. Travelling in beam waves, the ship experiences the largest value of $C_{fy}$. Like $C_{fx}$, these coefficients are
underestimated by CFD, especially for the beam waves, which is underestimated by 27.6%. The errors of CFD predictions for \( C_{f_b} \) under other wave conditions vary from 10.2% to 25.2%. In the oblique wave, the second-order force is dependent on motion, particularly on the sway motion in the beam waves. Thus, fixing the non-restoring motion will arise the drift force error related to the radiation especially when \( \mu \) is close to 90°. Consequently, the difference in \( C_{f_b} \) between CFD and EFD increases as \( \mu \) is approaching 90° (as shown in Fig. 6b).

The comparison of mean yaw drift forces \( \bar{C}_{f_{yw}} \) is shown in Fig. 6c. It shows good agreement between CFD and EFD when \( \mu < 90° \). The largest value of yaw drift force occurs when \( \mu = 60° \), since \( C_{f_y} \) reaches its maximum value and the arm of force is quite considerable. Similar to \( C_{f_b} \), the yaw moment for 60° is underestimated. The results of \( C_{f_{yw}} \) have only the same trend with EFD results, and errors are obvious in some cases.

**Time History of Forces**

The time histories of force coefficients, \( C_{f_x} \) and \( C_{f_y} \), can be defined in the same way as \( C_{f_b} \) and \( C_{f_{yw}} \) by using transient values. The results of \( C_{f_x} \) and \( C_{f_y} \) for all cases can be found in Fig. 7. The results present strong nonlinear behaviors instead of smooth cosine curves. As shown in Fig. 7c, the extremum of \( C_{f_x} \) appears three times in one period, indicating that this curve is composed of several different frequency components. For the cases of 0° (Fig. 7a), 120° (Fig. 7e), and 180° (Fig. 7g), spikes can be observed corresponding to strong wave slamming. With the heading approaching 90°, the amplitudes of both \( C_{f_x} \) and \( C_{f_y} \) increase. The peak values of \( C_{f_b} \) increase moderately and are of the same order for all heading angles. By contrast, the peak values of \( C_{f_y} \) increase considerably. For instance, the peak value of \( C_{f_y} \) for 90° is 3 times larger than that for 30° while the amplitude of \( C_{f_y} \) for 90° is 12 times larger than that for 30°. For \( \mu \) in the range from 30° to 120°, the peak values of \( C_{f_x} \) are much larger than those of \( C_{f_y} \).

**Fast Fourier Transform Results of Forces**

To analyze the linear and nonlinear effects on forces, the unsteady histories of \( C_{f_x} \) and \( C_{f_y} \) are analyzed by Fast Fourier transform (FFT). Figure 8 shows the FFT results of \( C_{f_x} \) and \( C_{f_y} \) for \( \mu \) from 0° to 60°. It illustrates that all frequency components distribute in a harmonic way and the fundamental frequency is wave frequency. When \( \mu \) increases from 0° to 60°, high-order harmonics become more obvious. For the case of 30°, the second harmonic of \( C_{f_x} \) has the largest amplitude (Fig. 8a). When \( \mu \) changes to 60°, the third harmonic of \( C_{f_y} \) has the largest amplitude (Fig. 8b). These nonlinear characteristics are mainly caused by the increment of ship motion and the enhancement of the diffraction effect, which will also be discussed in the following section.

Figure 9 shows the FFT results of \( C_{f_x} \) and \( C_{f_y} \) in the beam waves. The fundamental frequency can be captured at wave frequency. The nonlinearity becomes weaker compared with the case of \( \mu = 60° \).

Figure 10 shows the FFT results of \( C_{f_x} \) and \( C_{f_y} \) for wave headings from 120° to 180°. Again, the nonlinearity is weaker compared with the cases of \( \mu = 30° \) and 60°. The strongest nonlinearity can be observed for the \( C_{f_y} \) of 180°. It is partly caused by strong wave slamming on the stern, which will be shown later.

**Ship Motions**

The FFT results of motions for all wave conditions are obtained, but only 60° is presented in this paper since the characteristics of different wave conditions are similar. Figure 11 shows FFT results of heave, pitch, and roll for \( \mu = 60° \). The vertical axis represents the amplitude of motions at different frequency components normalized by wave amplitude (a) for heave and wave steepness (ak) for pitch and roll. The FFT results (Figs. 12a and 12b) show that only the wave frequency dominates the motions of pitch and heave. These motions are quite linear compared with the wave forces. But for the roll motion, the FFT results (Fig. 12c) show that there is a frequency component at 0.2 Hz lower than the wave frequency. According to the experimental benchmark of DTC (El Moctar et al., 2012), the natural
frequency of the roll decay motion is about 0.2 Hz. So it is evident that the roll motions are affected by both wave frequency and its natural frequency.

To investigate the motions of ship quantitatively, the heave, pitch, and roll are analyzed by transfer functions. The definition of transfer functions of motions can be given as follows:

\[ TF_3 = \frac{X_{31}}{a} \]  
\[ TF_4 = \frac{X_{41}}{a} \]  
\[ TF_5 = \frac{X_{51}}{ak} \]

where \( TF_3, TF_4, \) and \( TF_5 \) are the transfer functions of heave, roll, and pitch motions, respectively; \( X_{31}, X_{41}, \) and \( X_{51} \) are the amplitudes of the frequency component equal to wave frequency obtained by the FFT algorithm, in which the subscripts 3, 4, and 5 indicate the heave, roll, and pitch, respectively; \( a \) is the wave amplitude; and \( ak \) is the wave steepness. The results of the transfer functions are shown in Table 3.

Table 3 shows that the \( TF_3, TF_4, \) and \( TF_5 \) increase as the heading is closing to the beam waves. This can be explained as follows. In this study, \( \lambda/L_{pp} = 0.36 \), and the wave length is relatively small compared with the ship length. When the wave direction is far from the beam waves (such as 0° and 180°), the motions will be quite small, and diffraction is dominated. But when \( \mu \) approaches 90°, the wave length becomes effectively shorter in relation to the ship’s breadth, and there is an increase in the effective steepness of the waves. Consequently, \( TF_4 \) reaches its peak value at \( \mu = 120° \).

Figure 12 combines the unsteady histories of \( C_{f_h} \) and \( C_{f_r} \), heave, roll, and pitch motions of 60° into one figure, in which the relationship between wave forces and ship motions can be observed. The dashed lines in the figure indicate the moments when the \( C_{f_h} \) approaches peak value. The maximum \( C_{f_h} \) occurs when the heave reaches the highest position and the trim angle reaches the positive maximum value. Meanwhile, the curvature of
Fig. 12 Time histories of $C_{fx}$, $C_{fy}$, heave, roll, and pitch ($\mu = 60^\circ$)

<table>
<thead>
<tr>
<th>Heading angles</th>
<th>0</th>
<th>30</th>
<th>60</th>
<th>90</th>
<th>120</th>
<th>150</th>
<th>180</th>
</tr>
</thead>
<tbody>
<tr>
<td>$TF_3$</td>
<td>0.087</td>
<td>0.123</td>
<td>0.212</td>
<td>0.679</td>
<td>0.177</td>
<td>0.067</td>
<td>0.07</td>
</tr>
<tr>
<td>$TF_4$</td>
<td>0</td>
<td>0.145</td>
<td>0.188</td>
<td>0.234</td>
<td>0.312</td>
<td>0.066</td>
<td>0</td>
</tr>
<tr>
<td>$TF_5$</td>
<td>0.111</td>
<td>0.151</td>
<td>0.241</td>
<td>0.037</td>
<td>0.095</td>
<td>0.117</td>
<td>0.083</td>
</tr>
</tbody>
</table>

Table 3 Transfer functions of heave and pitch motions

$C_{fy}$ also changes at this moment. In addition, the minimum $C_{fx}$ and $C_{fy}$ occur when the heave and pitch approach negative maximum values. Therefore, the ship motions have an impact on the forces and do contribute to the nonlinearity.

Wave Patterns and Dynamic Pressure

As shown in Fig. 7c, for $\mu = 60^\circ$, the curves of $C_{fx}$ have several peak values in every period indicating the nonlinearity feature. So it is natural to analyze these moments when extremum occurs. The wave patterns and hydrodynamic pressures at these moments may illustrate the cause of the nonlinearity and the complexity of wave forces. The wave patterns and hydrodynamic pressures corresponding to three different moments (circled in Fig. 7c) are shown in Fig. 13. The following analyses on these three moments will demonstrate that the nonlinearity mainly caused by wave runups and slamming, and asymmetric wave patterns between the ship’s two sides give rise to the dramatic change of the instantaneous wetted surface.

Figure 13 shows that two large wave peaks with one trough or two wave troughs with one peak emerge periodically on the port side, while the wave pattern is quite flat on the starboard. The difference in wave pattern between the two sides causes the fluctuation of time history of wave forces and is responsible for the mean drift forces. In other words, this asymmetric wave pattern makes the wetted surface of the ship very variable and leads to a nonlinear effect.

Figure 13c shows that when $\mu = 60^\circ$, the incident direction of the wave is nearly perpendicular to the forward shoulder of the hull. In this circumstance, the wave-ship interaction grows, which makes the value of longitudinal mean drift force, $C_{fx}$, reach its maximum (shown in Fig. 6a). When a crest slams on the forward shoulder at $t = 20.3$ s (as shown in Fig. 13a), a high-pressure region can be observed around the bow, which makes the value of $C_{fx}$ reach its maximum corresponding to the first circle in Fig. 7c. This is the highly nonlinear effect arising when a steep and breaking wave impinges on the ship.

On the starboard side at 20.5 s (Fig. 13b), a negative pressure zone can be observed at the bow region while a positive pressure zone can be seen at the stern. This pressure distribution makes...
approaches the deck corresponding to the peak value of a wave crest directly impinges on the port, and the wave crest causes the largest heave motion, as shown in Table 3. The violent change of water surface extending along the whole ship keels in the simulations may cause errors in CFD results. This minimum value of $C_t$ is related to the peak value of $C_{fz}$. At that time, the positive pressure zone appears at the bow and negative pressure zone at the stern, which causes the $C_{fz}$ curve to reach its maximum value again, corresponding to the third circle in Fig. 7c. On the starboard side, the wave crest always appears at the bow paired with a trough at the stern, or vice versa, which is related to the peak value of $C_{fz}$.

While on the port side, a pair of considerable crests (troughs) always appear at the bow and the stern simultaneously. Then the notable high (low) pressure zones caused by wave crests (troughs) at the bow and the stern offset each other instead of being responsible for the extreme value of $C_{fz}$. It is evident that the starboard wave pattern determines the occurrence of the extreme value of $C_{fz}$ despite its relatively smaller wave amplitude compared with that of the port side. These asymmetric wave distributions between two sides obviously vary the instantaneous wetted surface and determine the peak value of $C_{fz}$, which is correlated with the nonlinearity of the wave forces.

The strong runup in connection with wave-ship interactions can also be observed especially for the cases of $\mu$ equal to 90° and 180°. Figure 14 shows two snapshots of wave pattern and dynamic pressure distributions for $\mu = 90°$. For this case, one wave peak or trough alternates on the port side. At $t = 28.1$ s, a wave crest directly impinges on the port, and the wave crest approaches the deck corresponding to the peak value of $C_{fz}$. At $t = 26.2$ s, a trough can be observed on the port, which causes the minimum value of $C_{fz}$. At this moment, the wave trough reaches the region where the bilge keels should be. The ignorance of bilge keels in the simulations may cause errors in CFD results. This violent change of water surface extending along the whole ship causes the largest heave motion, as shown in Table 3.

Figure 15 shows two snapshots of wave patterns in the following sea. The wave slaming occurs as the wave peak or trough alternates at the stern. At $t = 23.8$ s, a wave trough arrives at the stern following the rudder coming out of the water. At $t = 23.3$ s, a wave crest approaches the deck and directly impinges on the transom plate corresponding to the peak value of $C_{fz}$. The waves slaming on the stern periodically increase the nonlinearity of $C_{fz}$, as shown in Fig. 10a.

The results of wave patterns and dynamic pressure demonstrate the complex diffraction wave and wave slamming leading to the nonlinear instantaneous wetted surface and wave forces.

CONCLUSIONS

In this paper, the mean drift forces and moments on a ship in regular oblique waves are predicted by the in-house solver, naoe-FOAM-SJTU, developed under the framework of the open source OpenFOAM packages. Open source library waves2Foam is imposed in the solver to handle the wave generation. A DTC container ship is considered under seven wave conditions with various heading angles. The computational results of mean drift forces are compared with experimental data. The forces are underestimated in the simulation, which is caused by the disregard of nonrestoring motions. The maximum value of $C_{fz}$ is captured at $\mu = 60°$, and the peak value of $C_{fz}$ is obtained at $\mu = 90°$, which is in accordance with the EFD. The maximum values of $C_{fz}$ and $C_{fz}$ occur when the amplitude of ship motions reach maximum and wave incident angle is perpendicular to the hull. In other words, the strength of radiation and diffraction determine the magnitude of mean drift forces. The time history and FFT results of longitudinal and lateral forces show strong nonlinear property. The analyses of ship motions and wave patterns demonstrate that the nonlinearity is related to the ship motion and complex wave diffraction and wave slamming.

The results of heave, pitch, and roll motions are also obtained in the simulations. When $\mu$ approaches 90°, transfer functions will increase, leading to a more obvious radiation effect. FFT results show that wave frequency totally dominates pitch and heave motions. But for roll motion, the natural frequency of roll can be captured.

The results of wave patterns and dynamic pressures show the complex wave diffraction and slamming leading to the nonlinear instantaneous wetted surface and wave forces.

From this study, it can be concluded that the naoe-Foam-SJTU solver is able to reasonably predict the wave forces and motions, especially for the fully nonlinear instantaneous wetted surface arising from strong wave slamming, wave diffraction, and ship motions. In addition, there are some deficiencies in the present study. First, the present work neglects the nonrestoring motions, which causes an underestimation of the mean drift forces. Second, more wave conditions (e.g., wave length, height) should be considered to achieve better results on the wave effects. In future work, the mooring system can be used to restrain nonrestoring motions, and more extensive simulations can be conducted to draw more comprehensive conclusions.

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