Numerical simulation of ship motion fully coupled with sloshing tanks by naoe-FOAM-SJTU solver

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Abstract

Purpose – The purpose of this paper is to verify the ability of our in-house solver naoe-FOAM-SJTU to solve the problem of exterior fluid field coupled with interior fluid field and discover the coupling effects between exterior field (ship motion) and interior field (sloshing tanks).

Design/methodology/approach – The solving equation is based on Navier–Stokes equation, by comparing two turbulence models [laminar model and Reynolds-averaged Navier–Stocks (RANS)], of which RANS model are chosen to do the simulation. A unified approach is adopted to simulate exterior and interior fields simultaneously, keeping the pressure and velocity the same in external and internal fields. By adding a new function of calculating forces on different patches, the inner sloshing moments and external wave exciting moments can be output.

Findings – The in-house solver naoe-FOAM-SJTU had the ability to simulate this problem and showed well agreement with experimental results. By considering ship motion with and without sloshing, it was figured that with the existence of sloshing tank, the ship natural frequency will be changed. When the two tank fillings are the same, there will be another roll peak appeared, which is natural frequency of sloshing tanks. Considering wave height and different filling in influence, the nonlinearity of sloshing in tank may give non-proportional response to ship motion.

Practical implications – With the ability to simulate well, the reality reference in the progress of FPSO or FLNG operation is obtained.

Originality/value – The value of this paper is a fully coupled CFD method which is adopted to solve the coupling effects, showing the ability to do the work well. It gives a referenced detailed information of inner and outer fluid field. Meanwhile, it carried out the impact pressure and damping force around the ship, which indicates the practical information in operations.

Keywords External and internal fields, Fully coupled approach, Naoe-FOAM-SJTU solver, Ship motion, Tank sloshing

Paper type Research paper

1. Introduction

Around 1960s, Abramson (1966) raised a study about a moved tank which will excite sloshing liquid in tank. He started the study about sloshing free surface for applying linear potential theory on solving the sloshing liquid in a circle container. Until 1980s, researchers...
began to discuss the problem about coupled effect on external wave and internal sloshing. Mikelis et al. (1984) carried out an experiment about ship model with membrane tanks, which lead to many researches on coupling effects on external and internal fields.

The study of coupling effects on external and internal fields comes to the ship motion with sloshing tanks in reality. When the tank is partially filled, the free surface will exist. Therefore, the ship motion which is influenced by waves will excite deformation of the free surface. In return, the sloshing liquid in tanks will have effect on ship motion. The difficulty of this problem is how to solve the external and internal fluid field simultaneously and figure out how they influence each other.

In early time, the way to solve this kind of problem in numerical method is potential theory both in interior and exterior fluid fields (interior fields refer to sloshing liquid in tanks, exterior fields refer to ship motion in waves). Rognebakke and Faltinsen (2003) applied linear potential theory to simulate two dimensional rectangular hull section in regular waves. They discussed linear and nonlinear sloshing model when solving the liquid flow in rectangular section. They pointed out the sloshing flow depended on sway motion frequency, which is dominated by wave frequencies. Malenica et al. (2003) and Molin et al. (2002) applied potential theory to analyze liquid flow in tank. However, they all solve this kind of problem separately. The motion response of ship and sloshing liquid in tank are not solved as a whole. Ship motion provides excitation to tank slosh, and the moments and forces generated by liquid in tank are added on ship. Newman (2005) started to consider this problem as a whole part. He used WAMIT model to solve ship motion with sloshing tanks in a unified approach. The inner fluid surface is included as an extension of wave free surface. This unified method enlarged the computation cost but simplified the analyzed parameters with little modification.

With the development of computational science, dispersing Navier–Stokes equation became common on the problem solving. For one hand, traditional potential theory cannot simulate when Reynolds number is high in sloshing tanks. For the other hand, the traditional potential theory did not have the ability to do the simulation when the diamond shape tanks have low filling ratios or high filling ratios (when the surface intersect with slope part of tanks, it will be difficult to solve out). To overcome these disadvantages, researchers turned to CFD method. As for meshless particle method, Shao et al. (2012, 2015) had applied an improved smoothed particle hydrodynamics method to simulate the sloshing tank and investigated the influence of baffles in rectangular tanks. For mesh method, SOLA method (Akyildiz, 2012) is first applied to simulate a 3D sloshing flow. With single-value function is used to capture the sloshing surface, the broken or roll over of fluid are not included. Therefore, volume of fluid (VOF) method (Hirt and Nichols, 1981) is considered to solve the surface in tanks. Using CFD method to solve interior tank sloshing had been studied by Saripilli and Sen, 2017a; Sen and Saripilli, 2017b; Li et al. (2012) and Jiang et al. (2015). In these works, the inner fields are analyzed by a CFD open source toolbox OpenFOAM, while external fields are analyzed by potential theories. By combining the hydrodynamic parameters of tanks with ship’s added mass and exciting forces, the coupled equations of ship motion and sloshing tanks are solved.

Recently, our work team had studied the coupling effects of ship motion and sloshing tanks. Based on the open source toolbox OpenFOAM, we developed our in house solver naoe-FOAM-SJTU. Several module had been added, such as six-DOF motion module and numerical wave tanks. Through naoe-FOAM-SJTU, Shen and Wan (2012) attempted to simulate a KVLCC ship with two LNG tanks. Following a unified approach, the fluid surface in tanks can be treated as an extension of external free surface. The advantage of the unified approach is the calculation is simplified somehow. Some local hydrodynamic parameters
such as exciting forces of tanks, forces or moments of tanks (which are seen as added interior moment of the vessel) are calculated and included directly in every time step. Using CFD method, the damping coefficient is considered and included. The disadvantages are also obvious, the time cost is high and a new function had to be added in it to carry out sloshing moments and wave exciting moments separately. Zhuang and Wan (2016, 2018) applied fully coupled CFD method to simulate a simplified LNG FPSO, verified this method is credible. These papers are mainly considered 20 per cent filling ratio and 30 per cent filling ratio, which is the same in aft and fore tank. Zhuang and Wan (2017) also added a new function to calculate sloshing moments and wave exciting moments separately, estimated the difference when filling condition is different in aft and fore tank.

In this paper, we provide a detailed data analysis of interior fluid coupled with exterior fluid in fully coupled CFD method. With the unified approach and ability to solve better results in large motion, results in CFD method have the meaning in reality. The choice of turbulence models is considered, and our unified approach is illustrated clearly. Considering both head wave condition and beam wave condition, the verification of our in-house solver naoe-FOAM-SJTU has been done. More results of roll RAO and internal sloshing moments various with wave frequencies are carried out, the features of ship motion coupled with sloshing tanks are observed. Considering wave height and filling ratios, the coupling effects are observed during the numerical simulation. Some conclusions are carried out in previous work, but without considering more wave frequencies, the conclusions cannot be generalized. The main focus of this paper is to provide results in motion RAO with and without sloshing, wave height influence, interior fluid surface profile, different filling ratio effect and exterior fluid field information, which can be a reference numerical simulation data to be considered.

2. Theory
In present work, the new function of patches calculation and fully coupled method are focused. Meanwhile, the choice of turbulence model and comparison between two turbulence models are introduced.

2.1 Governing equation
The incompressible Reynolds–Averaged Navier–Stocks (RANS) equations are adopted in this paper to investigate the viscous flow. Using dynamic deformation mesh, the governing equations are:

\[ \nabla \cdot \mathbf{U} = 0 \]  
\[ \frac{\partial \rho \mathbf{U}}{\partial t} + \nabla \cdot (\rho (\mathbf{U} - \mathbf{U}_g) \mathbf{U}) = -\nabla p_d - \mathbf{g} \cdot \mathbf{x} \nabla \rho + \nabla \cdot (\mu_{eff} \nabla \mathbf{U}) + (\nabla \mathbf{U}) \cdot \nabla \mu_{eff} + f \sigma + f_s \]  

where \( \mathbf{U} \) is velocity field, \( \mathbf{U}_g \) is velocity of grid nodes; \( p_d = p - \rho \mathbf{g} \cdot \mathbf{x} \) is dynamic pressure; \( \mu_{eff} = \rho (v + v_t) \) is effective dynamic viscosity, in which \( v \) and \( v_t \) are kinematic viscosity and eddy viscosity respectively. \( f_s \sigma \) is the surface tension term in two phases model. The choice of turbulence model will be illustrated below.
2.2 Volume of fluid method

The volume of fluid (VOF) method with bounded compression techniques is applied to control numerical diffusion and capture the two-phase interface efficiently. The VOF transport equation is described below:

\[ \frac{\partial \alpha}{\partial t} + \nabla \cdot [(U - U_0)\alpha] = 0 \]  

(3)

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

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

(4)

The boundedness is critical for VOF method and using upwind is the only implicit scheme to guarantee boundedness. However, implicit scheme is too inaccurate, for the interface is lost in multiphase. In OpenFOAM, a MULES (multidimensional universal limiter for explicit solution) scheme is applied to solve this problem. The MULES scheme substitutes equation (3) as:

\[ (\alpha - \alpha^0) \frac{V}{\Delta t} + \sum_f \phi_f \alpha_f = 0 \]  

(5)

\[ (\alpha - \alpha^0) \frac{V}{\Delta t} + \sum_f \phi_f \alpha_U + \sum_f \lambda_f^+ \phi_f^+ (\alpha_H - \alpha_U) + \sum_f \lambda_f^- \phi_f^- (\alpha_H - \alpha_U) = 0 \]  

(6)

where \( \alpha_U \) is in upwind scheme, \( \lambda_f \) is limiter, \( \lambda_f^+ \) is limiter on \( \alpha_{\text{min}}(=0) \), \( \lambda_f^- \) is limiter on \( \alpha_{\text{max}}(=1) \), \( \phi_f^+ \) is fluxes out of cells, \( \phi_f^- \) is fluxes into cells, \( \alpha_H \) is in high-order scheme. In this equation, to evaluate the limiter and advance \( \alpha \) solution, the boundedness is guaranteed. As the solution is explicit, Co number needs to be strict (usually the Co number is smaller than 0.25). The way to compress the interface and ensure conservation and boundedness can be found in detail in Zhuang and Wan (2018).

2.3 Six degrees of freedom model and patch separate

A fully 6DOF module with bodies is implemented. Two coordinate systems are used to solve 6DOF equation. We describe as the translation and rotation angles of the ship, representing motions of surge, sway, heave, roll, pitch and yaw, respectively. \((v_1, v_2) = (u, v, w, p, q, r)\) are the velocities in the earth-fixed coordinate system, which can be transformed to the body-fixed coordinate system by equations given below:

\[ v_1 = J_1^{-1} \cdot \dot{x}_1 \quad v_2 = J_2^{-1} \cdot \dot{x}_2 \]  

(7)

where \( J_1, J_2 \) are transformation matrices based on Euler angle. The forces and moments are projected into the earth-fixed system in following way:
\[ F = (X, Y, Z) = J^{-1} \cdot F_e \quad M = (K, M, N) = J^{-1} \cdot M_e \]  

(8)

To compute the sloshing forces and moments as well as wave exciting forces and moments separately, the single body is divided into several patches. In present study, the single body is divided into three patches: the hull of FPSO, the aft inner tank and fore inner tank. The wave forces are integrated on hull of FPSO patch, and inner fluid forces are integrated on aft or fore tank patches. The forces accumulated and calculated as a whole to compute for next step motion. Therefore, the new added function can export wave exciting forces and moments as well as sloshing forces and moments at the same time. The process is shown in Figure 1.

2.4 Numerical wave tank

The wave generation and wave absorbing are two essential parts in numerical wave tank. The wave generation of incoming regular wave through imposing the boundary conditions of \( \eta \) and \( U \) at the inlet. According to the linear deep Stokes water wave theory, the function of wave elevation and wave velocity which changes with time step will spread over the whole field.

Although the outlet boundary condition was adopted to let the flux flow out, the phenomenon of wave reflection can occur around outlet. Without a theoretically mature outlet boundary condition can be used, a wave absorb area is applied. The wave will decay in the absorb area (which is also called sponge layer) (Larsen and Dancy, 1983) to achieve the aim of wave absorbing. The wave absorbing is realized by adding a source term in momentum equation:

\[
- f_s(x) = \begin{cases} 
- \rho \alpha_s \left( x - x_s \right)^2 \left( u - u_{ref} \right), & x_s \leq x \leq (x_s + L_s) \\
0, & x < x_s
\end{cases}
\]

(9)

In which \( x_s \) is the starting point of the sponge layer, \( L_s \) is the length of the sponge layer, \( \alpha_s \) is dimensionless artificial viscous coefficient. The function of the coefficient is to control the strength of wave absorbing. Compared to traditional method of sponge layer, a reference
velocity $u_{\text{ref}}$ is added in equation. Usually, the reference velocity is set equal to inlet wave velocity, in order to keep mass conservation in the whole computational domain. (Cao and Wan, 2013).

2.5 Coupling model
This paper applied the fully coupled model to do the simulation. The approach is unified thus the interior free surface is solved as well as exterior free surface simultaneously. With a tunnel exists on the top of the tank, the inner space is connected with the outer space. Through the tunnel, the pressure and velocity of inner fields and outer fields keep the same, seen in Figure 2. Thus the interior fluid can be seen as an extension of exterior fluid in the simulation. With few modifications through the calculation, the approach realizes the real fully coupled model with ship motion and sloshing tanks.

2.6 Turbulence model
As the sloshing in tanks is a strongly nonlinear and in-homogeneous phenomenon, the choice of turbulence model is essential for accurate simulation. The common turbulence models in CFD methods are laminar model, RANS model and large Eddy simulation (LES) model. However, with high level cost of calculation of the LES model, only laminar model and RANS model are considered in this paper.

In laminar model, the momentum equation is solved as:

$$
\frac{\partial u_i}{\partial t} + \frac{\partial u_i u_j}{\partial x_j} = - \frac{1}{\rho} \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left( 2\nu S_{ij} \right) + f_i + \frac{1}{\rho} f_{\sigma}
$$

(10)

where $S_{ij}$ is the strain-rate tensor and $f_i$ is external acceleration.

For RANS model, the Menter’s shear stress transport (SST) $k$-$\omega$ turbulence model is used (Menter, 1994). The turbulence kinetic energy $k$ and the specific dissipation rate $\omega$ are governed as:

$$
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho U_i k)}{\partial x_j} = P_k - \beta^* \rho \omega k + \frac{\partial}{\partial x_j} \left[ (\mu + \sigma_k \mu_i) \frac{\partial k}{\partial x_j} \right]
$$

(11)

$$
\frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\rho U_i \omega)}{\partial x_j} = \frac{\gamma}{\nu_i} P_k - \beta \rho \omega^2 + \frac{\partial}{\partial x_j} \left[ (\mu + \sigma_k \mu_i) \frac{\partial \omega}{\partial x_j} \right] + 2(1 - F_1) \frac{\rho \sigma_{\omega 2} \omega}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}
$$

(12)

where $F_1$ is a harmonic function expressed as:

$$
F_1 = \tanh \left\{ \min \left[ \max \left( \frac{\sqrt{k}}{\beta^* \omega d^2}, \frac{500 \nu}{d^2 \omega} \right), \frac{4 \rho \sigma_{\omega 2} k^2}{CD_{\omega \nu} d^2} \right] \right\}^4
$$

(13)
where $v_t = \mu / \rho$ is the turbulent kinematic viscosity, and $\mu_t$ is calculated as follows.

$$\mu_t = \frac{\rho a_1 k}{\max(a_1 \omega, \Omega F_2)}$$  \hspace{1cm} (15)

$$F_2 = \tanh \left[ \max \left( \frac{2 \sqrt{k}}{\beta \omega d}, \frac{500\nu}{d \omega} \right) \right]^2$$  \hspace{1cm} (16)

Figure 3. Comparisons of computed and experimental free surface profiles
with experiments satisfied, for the laminar model are overestimated the sloshing flow and RANS model are wiped out some splashes. However, after considering the accuracy (the results of RANS model are more close to the experimental results) and more details can be given (such as vorticity around the ship), the RANS model is chosen for the simulations.

3. Comparison between theory and experiments

This section is devoted to verify our numerical methods before the analysis in the fully coupled inner and outer fluid fields. After the numerical settings displayed (Subsection 3.1), the mesh convergence (Subsection 3.2) has been done to make sure the influence of the mesh number is included. Both head wave conditions (Subsection 3.3) and beam wave conditions (Subsection 3.4) are considered to compare with experimental results.

3.1 Settings for numerical simulations

Before we start to analyze the coupling effects between ship motion and sloshing tanks, a method validation is required to verify the accuracy of our method. A series of numerical simulation were carried out to compare with the results from Nam et al. (2009), where they carried out the experiments with a simplified LNG FPSO. The LNG FPSO model is 1/100 scale of the full scale ship. The distance from the bottom of tank to the keel line is 3.3 m. The parameters of original tanks are shown in Table I. The model equipped with two LNG tanks along with the length of ship. The detailed ship parameters and arrangement of tanks can be found in Zhuang and Wan (2016, 2017, 2018).

Numerical configurations are illustrated in Figure 4. Figure 4(a) shows three patches (outer ship hull, fore tank and aft tank) of the numerical model. Based on the process shown in Figure 1, the wave forces integrated along the outer ship hull (the white patch) to output wave forces and moments, the inner fluid integrated in fore tank (the red patch) and aft tank (the green patch) to output sloshing forces and moments. Figure 4(b) and (c) gives the illustration of sloshing ratio, with 20-20 per cent in fore and aft tank (the same filling ratio in two tanks) and 82.6-23.5 per cent in fore and aft tank (different filling ratio in two tanks).

<table>
<thead>
<tr>
<th>Table I. Parameters of original tanks</th>
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<tr>
<td>Tank dimension</td>
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Figure 4. Numerical configurations of ship model

Notes: (a) Patch displacement of numerical ship model; (b) setting of filling ratio 20-20%; (c) setting of filling ratio 82.6-23.5%
Figure 5 shows the computational domain setup in this paper. The normal beam wave computational domain configuration can be found in previous work (Zhuang and Wan, 2016, 2017; 2018). The head wave computational domain is shown in Figure 5(a), in which \(-1.0L_{pp}<x<3.0L_{pp}, -1.5L_{pp}<y<1.5L_{pp}, -1.0L_{pp}<z<1.0L_{pp}\). The beam wave computational domain is set as \(-1.0L_{pp}<x<2.0L_{pp}, -1.5L_{pp}<y<1.5L_{pp}, -1.0L_{pp}<z<1.0L_{pp}\). Considered different wave frequencies when the wave length was large, the wave development needs large space. Therefore a larger computational domain is included for larger wave length condition, shown in Figure 5(b). The length of sponge layer is specified in both computational domain, to avoid wave reflection.

3.2 Mesh convergence
A mesh convergence is carried out as the mesh generation is vital for the stability and efficiency of our simulations. Following the mesh quality, the mesh convergence includes three different levels of grid numbers. The case was chosen to be in head wave with wave height equals to 0.025 m. The wave length is chosen to be 2.85 m, with zero filling conditions in ship. The cell numbers of different mesh qualities are 291w, 355w and 863w respectively. The background meshes were formed from blockMesh, and the refined mesh around the ship and free surface were generated by snappyHexMesh. Those are auto mesh generation utilities provided by OpenFOAM. The number of the grid is more than 20 in one wave height and more than 80 in one characteristic wavelength. Usually, as the wave field reflection effect of the structure is more complex and vortex around ship needs to be captured, hence the mesh near the ship is also refined. The mesh convergence is shown in Figure 6, considering accuracy and efficiency of the simulation, intermediate mesh quality is chosen, the detailed mesh generation is shown in Figure 7.

To keep the draft of the ship constant, the mass and coordinate of center of gravity needs to be adjusted in numerical simulations, shown in Table II. The treatment of changing the mass and center of gravity can avoid the error created by the mass difference in reality. With the adjustment of mass and center of gravity, it can be ensure that the motion will not be influenced by mass difference as the fluid in tanks provide the extra mass value. Meanwhile, the inertia moments of fluid in tanks are included in calculations.

**Notes:** (a) Head wave computational domain; (b) beam wave computational domain (for large wave length)
3.3 Verification in head wave cases

Figures 8 and 9 show the comparisons between several methods’ results in head wave conditions. The comparisons are among experimental results (Nam et al., 2009), linear potential theory (Gou et al., 2011), hybrid method (Jiang et al., 2012) and present work. Nam et al. (2009) provided not only results of their experiments but also results of numerical simulations, shown as the dotted line named “Nam et al.” All the results are analyzed in normalized parameters. The normalized pitch motion is given as: \( \frac{\theta L}{2A} \), in which \( \theta \) is maximum degree of pitch motion, \( L \) is the perpendicular length of ship and \( A \) is wave amplitude. The normalized heave motion is given as: \( \frac{\xi}{A} \), in which \( \xi \) is maximum magnitude of heave motion. The normalized wave frequency is given as \( \omega (L/g)^{1/2} \), in which \( \omega \) is natural frequency of wave. Under the configuration of experiments, the wave height is chosen to be 0.025 m. It can be found that the results of present work agree well with experimental results whether in zero filling conditions or in 20 per cent filling conditions.
Through the comparison between Figures 8(b) and 9, the pitch RAO shows little difference between zero filling ratio and 20 per cent filling ratio. To figure out obvious coupling effect, we turned to the beam wave cases.

3.4 Verification in beam wave cases

Figures 10 and 11 illustrate ship RAOs of different methods in beam wave conditions (90 degree incident waves). The normalized roll motion is given as: $\theta B/2A$, in which $B$ is breadth of ship and $\theta$ is maximum degree of roll motion.

Firstly, the results of zero filling condition, 20-20 per cent filling condition and 82.6 per cent-23.5 per cent filling condition are well agreed with the experimental results. Combined with the comparison in head wave cases, our method and in-house solver had the ability and accuracy to do the simulation of ship motion coupled with sloshing tanks in waves. Second, it can be found the obvious RAO difference between zero filling condition and partially filled conditions. The existence of sloshing tanks has strong influence in beam wave condition. To
figure out this kind of coupling effects more clearly, ship roll RAO with partially filled tanks are compared with that in zero filling conditions, shown in Figure 12.

It can be discovered that along with the change of incident wave frequency, the zero filling ship shows only one peak among the RAOs. This peak is the natural frequency of ship itself. When incident wave frequency is around the ship natural frequency, the ship motion response becomes very large. From Figure 12(a), when the tank filling ratio is the same in fore and aft tank, say 20 per cent, it shows two peaks (normalized frequency equals to 2 and 3.25) in RAOs. We can assume that these two peaks are natural frequency of ship with two tanks and tanks. With two tanks equipped, the natural frequency of ship itself also changed, from 2.5 to 2. The magnitude of roll RAO between zero filling condition and 20 per cent filling condition are also various, but it becomes alike when the normalized frequency equals to 3.5. This will discuss later in this paper. In Figure 12 (b), when the filling ratio is different in fore tank and aft tank, the RAOs are distinguished from both zero filling

**Figure 10.**
RAOs in beam wave condition with 0-0% filling condition

**Notes:** (a) Heave motion RAO; (b) roll motion RAO

**Figure 11.**
Roll RAOs in beam wave condition

**Notes:** (a) 20-20% filling condition; (b) 82.6-23.5% filling condition
conditions and 20 per cent filling conditions. In 82.6-23.5 per cent filling ratio case, the RAO only shows one peak. The two tanks have their own unique natural frequency, the response of these two tanks affected each other to wipe out the natural frequency of sloshing tanks. When it comes to the magnitude of roll RAO, the coupling effects between ship motion and sloshing tanks exist for 82.6-23.5 per cent filling conditions. Around the natural frequency of the ship, the value of roll RAO is obviously smaller than the ship without sloshing tanks. For the fore tank with 82.6 per cent filling ratio, the fluid in tank may rarely slosh thus the fore tank has less influence on ship motion. But for the aft tank with 23.5 per cent filling ratio, small filling ratio will excite fluid in tank violent slosh, which may provide large effects on ship motion.

4. The coupling effects in different wave heights
As we all known, large wave height provides large wave energy. Thus, the ship motion becomes violent due to large wave height. To figure out the wave height influence on ship coupled with two tanks, three different wave heights were chosen. They are H = 0.025 m, H = 0.05 m and H = 0.1 m. Meanwhile, the results carried from Gou et al. (2011) are compared with our present work. The method Gou et al. (2011) applied was linear time domain theory, they treated free surface in fully linear hypothesis. Therefore, we can consider the results from Gou et al. (2011) as the minimum wave height. The roll RAOSs of different wave heights are shown in Figure 13.

It can be seen from Figure 13, wave height has significant influence on ship coupled with sloshing tanks. Around the natural frequency of the ship coupled with tanks, the maximum roll RAO is under H = 0.025 m and the minimum roll RAO is under H = 0.1 m. With the increase of wave height, the roll RAO decreases around the ship natural frequency accordingly. When it comes to natural frequency of sloshing tanks, the roll RAO appears to be the same under H = 0.05 m and H = 0.1 m, the maximum roll RAO shows in H = 0.025 m.

An assessment of roll RAO in different wave amplitude has been made by performing the sloshing moments and sloshing free surface profiles. Figure 14 illustrates the inner sloshing moments (fore tank sloshing moment) and external moments in different wave heights. It can be seen that the moments accords with the roll RAOSs. Around the natural frequency of ship, the dimensionless moment (M/ρgLΔh, in which h is the height from fluid surface in tank to the bottom of the tank) of H = 0.025m is the largest one, even larger...
than normalized external moment (in equation of dimensionless external moment, h is draft). The sloshing in tanks has the strongest influence on ship motion when wave height is 0.025 m, and then is 0.05 m and 0.1 m. Figure 15(a) (b) and (c) shows the sloshing surface profile around ship motion resonance frequency. When wave height is 0.025 m, the surfaces in tank are linear. In 0.05 m wave height condition, the free surface performs in a nonlinear feature. It appears like a propagation wave from left bulkhead to right bulkhead. The nonlinear phenomenon is not obvious, it can only be observed two small wave peak exist when fluid propagate. When the wave height continues to increase, the sloshing in tank becomes violent due to large ship motion. Shown in Figure 15(c), the free surface of fluid in tank leaves off the lower corner of the tank and accumulates on the other bulkhead side as ship roll to the maximum degree. While the ship returned from its extreme position to its balance position, the acceleration gives fluid in tanks a push to form a more obvious propagate wave. Due to the geometry of the LNG tank, there was a climb up of fluid in tanks. After the ship reaches the other extreme position, the climbed fluid flows down on the bulkhead. Although the nonlinear phenomenon in tanks is obvious, the fluid climb on the bulkhead gently without impulsive phenomenon happened.
Notes: (a) Sloshing surface profile in $H = 0.025$ m ($\omega(L/g)^{1/2} = 2$); (b) sloshing surface profile in $H = 0.05$ m ($\omega(L/g)^{1/2} = 2$); (c) sloshing surface profile in $H = 0.1$ m ($\omega(L/g)^{1/2} = 2$); (d) sloshing surface profile in $H = 0.025$ m ($\omega(L/g)^{1/2} = 3.25$); (e) sloshing surface profile in $H = 0.05$ m ($\omega(L/g)^{1/2} = 3.25$); (f) sloshing surface profile in $H = 0.1$ m ($\omega(L/g)^{1/2} = 3.25$); (g) sloshing surface profile in $H = 0.1$ m ($\omega(L/g)^{1/2} = 3.5$)
When the frequency is larger than ship resonance frequency, the trend of the inner sloshing tank appears to be decreased except the 0.1 m wave height case. It shows a crest in natural sloshing frequency. With the incident wave frequency increases, the energy in wave decreases. Thus, the ship motion becomes small and fluid in tanks appears to be calm. When normalized frequency is 3.5, the inner tank sloshing moment is very small, which can explain roll RAO is almost the same in 20 per cent and zero filling condition in Figure 12(a).

Figure 15(d), (e) and (f) shows the fluid surface profiles in tank. If wave height is small, the slosh in tank can be seen as a linear motion. When it comes to 0.05 m wave height, the surface profile in tank performs like a sinusoidal curve. The resonance responses are not obvious in both H = 0.025 m and 0.05 m. In case of H = 0.1 m, it can be found that there exists an obvious phase difference between sloshing flow and ship motion. When the ship moves to the extreme position of its own, the peak of fluid in tanks reaches the opposite side of the roll motion. Compared with the same wave height of dimensionless frequency equals to 2.25, 2.5 and 3 (Zhuang and Wan, 2016), this only happens in 3.25 and 3.5 (Figure 15(g), which can be observed but not clearly in phase difference). We may assume since a specific frequency between 3 and 3.25, the phase difference between ship motion and sloshing fluid happens. The fluid free surface in tank under H = 0.1 m begins to turn over and nearly reach the upper slope of the tank. On the top of the fluid surface, the nonlinear phenomenon occurs. The fluid breaks and rolls down from the top.

Figures 16 and 17 show the time history of global ship motion and internal sloshing moment in different wave heights under these two wave frequencies. In Figure 16, both ship motion and inner sloshing moment time history are sinusoidal. Although we discover the nonlinearity in sloshing tanks in H = 0.1 m, the nonlinearity did not show through the sloshing moment. The ship motion time history and internal sloshing moments show a little phase difference in Figure 17. It can be assumed that there is a time delay between ship motion and tank sloshing, which is the same in Figure 15 (d)-(f). In time history of inner tank sloshing moments, we discovered there are two peaks in the curve of H = 0.1 m. The two peaks present the nonlinearity in sloshing tanks, and the impulsive feature on the bulkhead. However, the nonlinearity of sloshing tanks did not affect the characteristic of ship motion, that is, the ship motion time history is still sinusoidal. Therefore, it can be concluded that the sloshing in tanks is a local phenomenon, which can only influence the magnitude of ship motion but cannot change the feature of ship motion.

**Figure 16.** The global ship motion and internal sloshing moment time history in 2 normalized frequency (20-20%).

**Notes:** (a) Ship motion; (b) sloshing moment
To visualize the existence of damping forces around the ship model, the vortex around the ship model is shown in Figure 18. As in H = 0.025 m the ship motion is too small to capture the vortex, only H = 0.05 m and H = 0.1 m are included. First, we can find the vorticity exists around the ship bilge, which illustrates the damping around the ship cannot be ignored. Second, when we compare the value of vorticity, the large wave height has high value of vorticity, and the diversity of vortex is more obvious. When the ship motion is violent due to large wave height, the damping force will make a huge contribution to ship motion. Therefore, the damping forces around the ship are vital and cannot be ignored.

According to the setup of experiments (Nam et al., 2009), a set of pressure probes are displaced on the bulkhead, which can be found in Zhuang and Wan (2016). The time history of impact pressure on inner tank is shown in Figure 19. Although the roll RAO is large around the natural frequency of ship, there exists no impulsive pressure on bulkhead. When the incident wave frequency closes to resonance frequency in tanks, impulsive pressure appears, seen in Figure 19(b). Therefore, ship coupled with partially filled tanks should pay more attention to the wave frequency equals to resonance frequency of inner fluid in tanks. Besides, the time history of impact pressure on bulkhead provides a three-dimensional phenomenon of inner sloshing fluid. On the one hand, the sloshing in tank which is excited by ship motion is three-dimensional itself; on the other hand, the pitch motion will also affect the sloshing tank, which make this kind of phenomenon more visible. Compared with
previous work (Zhuang and Wan, 2016), the impact pressure on normalized frequency 2.25, 2.5 and 3, we can observe that when the wave height is large, the impulsive phenomenon happens in 2.5, 3 and 3.25. However, the time last of impulsive process and impulsive pressure magnitudes are the largest in 3.25, which is resonance frequency around fluid in tank.

Above all the analysis of roll RAO and sloshing moments in 20-20 per cent filling ratio under different wave heights, it can be figured that the sloshing is a frequency-dependent phenomenon. Large wave height provides the energy to fluid in tanks to move, but impulsive and strong nonlinear phenomenon happens due to incident wave frequency. The sloshing fluid is a local phenomenon, which only changes the magnitude of ship motion but cannot transform the feature of ship motion. With the increasing of the wave height, the ship motion coupled with sloshing tank is not proportional to the wave height due to the sloshing phenomenon.

5. The coupling effects in different filling ratios
As shown in Figures 11 and 12, roll RAO of different filling ratios in fore and aft tank appears to be distinguished from that of same filling ratios, the coupling effects of 82.6-23.5 per cent filling conditions are discussed. For the fore tank filling condition, the free surface in

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**Figure 19.** Impact pressure on bulkhead

Notes: (a) Impact pressure on bulkhead in H = 0.1 m ($\omega(L/g)^{1/2} = 2$); (b) impact pressure on bulkhead in H = 0.1 m ($\omega(L/g)^{1/2} = 3.25$)
tank approaches the upper slope of the tank which cannot be solved by linear potential hypothesis can be simulated accurately in our method. To figure out the coupling effects between ship motion and sloshing tanks, three increased wave heights are considered as in 20-20 per cent cases. Figure 20 shows the roll RAO various with normalized frequencies and wave heights in 82.6-23.5 per cent filling condition. The trend of RAOs in different wave height is almost alike; the largest roll RAO appears around 2.5 dimensionless frequency.

Meanwhile, the sloshing moments of fore and aft tank are analyzed in Figure 21. Unlike 20-20 per cent filling condition where fore tank sloshing moment is the same with aft tank sloshing moment (Zhuang and Wan, 2017), different filling ratios in two tanks provide their own moments separately. It can be found that in fore tank with 82.6 per cent filling ratio, the inner tank moments are very small in three wave height, even much smaller than external moments. For the filling ratio is very high in fore tank, the fluid in tank rarely provide nonlinear sloshing moments. But for aft tank with 23.5 per cent filling ratio, the inner sloshing moments are larger than fore tank inner sloshing moments apparently. Thus, the slosh in aft tank takes the main part of the coupling effects and the coupling effects depend on sloshing characteristics of 23.5 per cent filling ratio in large extend. It can be observed

![Figure 20](image_url)

**Figure 20.** Roll RAOs in beam wave condition under different wave height (82.6-23.5%)

![Figure 21](image_url)

**Figure 21.** Sloshing moments and external wave moments in beam wave condition under different wave height (82.6-23.5%)

**Notes:** (a) Sloshing moment of fore tank; (b) sloshing moment of aft tank
that in $H = 0.05 \text{ m}$, the sloshing moments of both fore and aft tanks are large. Without accurate explanation on that, we only assume the wave excited sloshing under $H = 0.05 \text{ m}$ is quite large and nonlinear. We only infer that the ship motion is not proportional to wave height due to the sloshing phenomenon, however, in different frequencies the interior sloshing moments are complex.

We only consider when incident wave frequency is around the ship natural frequency. Under this condition, global ship motion time history and fore tank moments as well as aft tank moments are displayed, shown in Figures 22 and 23. The case with $H = 0.025 \text{ m}$ has a small phase difference with the other two wave heights cases, for the other two wave heights have larger nonlinear sloshing features. The nonlinearity can be observed in the internal sloshing moments time history. The fore tank moments are small even in large wave height cases, while the aft tank moments are increased along with the increasing of wave height. In $H = 0.025 \text{ m}$, the sloshing in aft tank is almost linear; thus, there is no impulsive curve can be observed. However, in $H = 0.05 \text{ m}$ and $H = 0.1 \text{ m}$, there exist several small peaks in the crest of the curves.

Figure 24 illustrates sloshing surface profile in $H = 0.05 \text{ m}$. We need to mention that the sloshing surface profiles of the other two wave heights can be found in Zhuang and Wan (2017). Due to large ship motion, the excited sloshing in aft tank shows a strong nonlinear characteristic in tank fluid. The fluid splashes on the bulkhead and turns over to form a new free surface in tank. When the fluid reaches the bulkhead, the formed wave breaks on the bulkhead and the slope of the fluid. The sloshing in fore tank is violent but still linear. With a mirror view of fore and aft tank, the sloshing phases between fore and aft tank keep the same. In Figure 25, normalized frequency of 3 in $H = 0.1 \text{ m}$ of sloshing profile is considered. We can clearly see the violent and broken free surface in aft tank. Comparing to the same cases in 20-20 per cent, the nonlinearity of sloshing profile is not as strong as that

**Figure 22.**
The global ship motion time history in 2.5 normalized frequency. (82.6%-23.5%)
in 23.5 per cent. It may be assumed that the high filling ratio tank did not absorb the energy from wave; thus, the energy pass through to the low filling ratio tank. With much higher energy, the excited sloshing in tank is violent.

6. Conclusion
In this paper, we discussed ship motion coupled with two partially filled tanks in fully coupled CFD method. With a unified approach, we consider fluid in tanks as an extension of exterior free surface, thus the simulation can work without some interacted hydrodynamic parameters modified. Two numerical turbulence models (laminar model and RANS model) were calculated and compared to find a better way to simulate sloshing phenomenon in tanks. Comparing with the experimental profile of free surface in tank, RANS model wiped out the peak of sloshing fluid a little and laminar model overestimated the sloshing in tank. Considering with accuracy and detailed needs, RANS model was chosen. A new function was added into our method, to output the external wave force and internal fluid force. Applying our method in house solver naoe-FOAM-SJTU:
First, head wave conditions and beam wave conditions are both calculated to compare with the experimental results in several wave frequencies. The results agree well with experimental data, no matter they are zero-filling condition or partially filled conditions. This verifies the accuracy and credibility of our method. After finding that the coupling effects are small in head wave conditions, more analysis on beam wave conditions are carried out. Compared the roll RAO between ship with sloshing (20-20 per cent) and without sloshing, a change of natural frequency of ship itself occurs. Sometimes the sloshing tanks are known as anti-rolling tanks, which can reduce the ship motion to some extent. However, due to the changes of natural frequency, the existence of the sloshing tanks may increase ship motion instead. Even when frequency becomes far away from original ship resonance frequency, the vicinal frequency closes to natural frequency of sloshing tank, the ship motion will also be increased.

Second, the influence of different wave heights on the coupling effects is included. In total, 20-20 per cent filling condition and 82.6-23.5 per cent filling condition are considered. During the discover between the wave frequencies, two specified wave frequencies are included to do the far more analysis. The fluid motion in tank is a frequency-depend motion, it may not violent even in large ship motion response. But when the incident wave frequency closes to resonance frequency of fluid in tank, the slosh in large wave height will be violent and shows strongly nonlinear feature. Meanwhile, impulsive pressure on bulkhead will occur. Although the nonlinearity in sloshing tanks is obvious, it only influences the ship motion magnitude or motion phases, the ship motion is still sinusoidal. The sloshing in tanks is a local phenomenon, which wiped out all the nonlinearity during the integration.

At last, the influence of two different filling ratios is included. Besides from the same filling ratio in two tanks, those tanks influence each other, showing a more complicated phenomenon. The lower filling ratio tank takes the main part of the coupling effects, for the high filling ratio sloshes almost linearly, provides little sloshing moments on ship. Under large wave height, the low filling ratio tank moves more violent, the breaking of the free surface is more obvious. The sloshing phenomenon in different filling ratios is more complicated, compared to the sloshing phenomenon in 20-20 per cent filling ratios, we may assume that the wave energy pass through the high filling ratio to low filling ratio, make the 23.5 per cent filling condition more violent.

In the future, we will consider nonlinear or freak wave condition through ship motion coupled with sloshing tanks. The sloshing fluid cannot change the linear motion of ship, we want to figure out how it works when the ship encounter with nonlinear wave conditions.

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