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# Experimental and numerical investigation of sloshing under roll excitation at shallow liquid depths

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ABSTRACT

This paper investigates sloshing at shallow-liquid depths in a rectangular container by using experimental and numerical methods. A motion platform is used to perform a prescribed periodic rotational motion to excite the liquid sloshing at a range of frequencies and filling levels. Simulated free-surface elevation is compared with the experimental results for a selection of cases. The wave mechanisms at the chosen fillings are studied by combining numerical methods and the experimental results. We find that the simulated free-surface elevation is in close agreement with experimental results inside the resonance zone. But at frequencies above the bifurcation point, with several overlapping waves, the deviation is increasing. The bifurcation point is determined for a range of filling levels through observation. The numerical results provide important information about sloshing mechanisms at these depths. Complex interaction between the bottom, the lower layer and the wave influences the amount of dissipation before the wave hits the wall. The existing theory seems to be too conservative in predicting the occurrence of hydraulic jumps in the upper limit.

## 1. Introduction

Sloshing can be characterized as the motion of liquids in containers or vessels. Sloshing can occur with more than one immiscible liquid, as studied in Rocca et al. (2002) and La Rocca et al. (2005). There are several applications in which sloshing may occur. The basic problem sloshing presents is estimating the hydrodynamic pressure distribution, forces, moments, and natural frequencies. Extensive work on how to approach it analytically can be found in Ibrahim (2005) and Faltinsen and Timokha (2009). The former focuses on space applications, while the latter focuses on sloshing within the maritime field. Naturally, a moving ship in waves is subject to sloshing. Within the maritime field there are tanks of different applications, but in general all marine vehicles have some kind of tank installed on-board. Examples are roll-stabilizer tanks or cargo tanks carrying different type of liquids. Impact at shallow depths can cause great damage to the tank. The resonance zone extends to frequencies higher than the calculated first natural frequency. Depending on the depth, frequencies below the first natural frequency result in a bore (Olsen and Johnsen, 1975; Peregrine and Svendsen, 1978). By increasing the frequency, it is possible to cause the bore to travel all the way from one wall to the other. Further increase of the frequency results in a narrow region where a steep solitary wave travels the entire tank length without breaking, and this results in severe impact on the side walls. This paper aims to characterize wave patterns in a sloshing tank under roll and to identify frequencies that cause severe impacts at low fillings.

There are many methods to analyse sloshing and several studies use experimental or analytical approaches to examine the subject. Olsen and Johnsen (1975) characterized sloshing at these depths and performed a limited number of tests with forced roll. Armenio and Rocca (1996) investigated sloshing with forced and free oscillations at shallow depths under rolling motion. The roll amplitude varies between 1.0° and 4.5° with two different depths. They compare results using the shallow water equations (SWE) form of the Navier-Stokes equations and RANS equations, and they found higher accuracy with RANS than with SWE. La Rocca et al. (2000) and Mele and Armenio (1997) also performed theoretical and experimental analysis of sloshing in a rotating container at intermediate depths. A fully nonlinear model is defined by applying the variational method. A technique to select the most energetic modes from experimental tests is presented. The comparison between experiments and theory shows good agreement.

To investigate the detailed flow conditions in sloshing, both with and without internal structures, computational fluid dynamics (CFD) provide promising capabilities. Viscous dissipation is accounted for and wave breaking regimes can be modelled with good accuracy. The literature offers several studies. Gómez-Goñi et al. (2013) compared

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two CFD tools using the volume of fluid method (VOF) and a multi-modal sloshing model by Ansari et al. (2011). They found good agreement between the two CFD codes, but the multi-modal method over-predicted the wave amplitude in some conditions. Bai et al. (2015) used a finite difference CFD model to simulate a full scale LNG tank undergoing realistic ship motions. To capture the free surface, a level-set method is employed. To validate the simulations, both longitudinal and rotational motions are considered. Few cases that compared free-surface elevation are considered. The comparison of pressure data showed acceptable agreement. Zhao and Chen (2015) implemented a finite-analytical NavierStokes (FANS) flow solver in conjunction with a new coupled level-set and VOF method (CLSVOF). Impact pressure from simulations are compared with experimental data and shows good agreement. They also compared the method to a level-set method with global mass conservation and found that the CLSVOF method showed an significant reduction in the relative mass change.

An increasing trend is to use large-eddy simulations (LES), but this comes at a computational cost, which increases with higher Reynolds number. Liu and Lin (2008) simulated sloshing in a three-dimensional tank using LES with the Smagorinsky subgrid scale model. However, their comparisons only concern non-linear sloshing conditions, excluding the resonant case. The cases with violent sloshing were not validated against experimental data.

Another method that has proven promising in predicting wave motions in sloshing is smooth particle hydrodynamics (SPH). Iglesias et al. (2004) performed SPH simulations of passive anti-roll tanks. The phase-lag for the roll moment is compared to experimental results for different fillings and roll amplitudes. Bouscasse et al. (2013) investigated shallow depth sloshing for sway. They compared experimental results with a $\delta$-SPH scheme and found that the method proved to be robust and reliable in studying violent free-surface flows. An extensive experimental program is presented, with several amplitudes and fillings. An additional classification of the wave patterns by Olsen and Johnsen (1975) are presented. Delorme et al. (2009) and Bulian et al. (2010) also compare SPH simulations and experiments.

The numerical model, REEF3D (Bihs et al., 2016; Bihs and REEF3D, 2016) which we use in this work is based on discretization and solving the Reynolds-averaged Navier-Stokes equations (RANS). The novelties of the RANS simulations are the use of level-set method and the improved turbulence boundary condition at the free surface. The model has been extensively used for wave hydrodynamics problems (Chella et al., 2015), ocean wave energy (Kamath et al., 2015) and sediment transport problems (Afzal et al., 2015). Forced sloshing within the proximity of the first mode natural frequency as well as free sloshing is simulated and compared to experiments performed at the lab facility at the Norwegian University of Science and Technology (NTNU) in Ålesund. Together with the high-order numerical treatment of the governing equations, this leads to high-quality simulation results, as shown through the comparison with the measured data in the first part of the paper. In the second part, investigation of the resonance zone at shallow depth sloshing is performed. The mean wave amplitude and bifurcation point is determined for a range of fillings. The combination of RANS simulations and experimental observations has led to an improved representation of the sloshing hydrodynamics.

## 2. Numerical model

### 2.1. RANS equations

The governing equations are the incompressible RANS equations given in tensor notations valid for two and three dimensions:

$$\frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\nu + \nu_t)\left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right)\right] + S_i \tag{1}$$

Where $u$ is the velocity, $p$ the pressure, $\rho$ the density and $\nu$ and $\nu_t$ are the viscosity and turbulent eddy-viscosity respectively. The last term are the body forces. Since we are using a tank-fixed coordinate system, source terms in addition to gravity must be accounted for to represent the equations in a non-inertial global system. The motion is harmonic, and the $x$- and $z$-terms are given by:

$$\begin{aligned} S_x &= \ddot{\theta}(z - z_m) + \dot{\theta}^2(x - x_m) - g sin\theta, \\ S_z &= -\ddot{\theta}(x - x_m) + \dot{\theta}^2(z - z_m) - 2\dot{\theta}\bar{u} - g cos\theta. \end{aligned} \tag{2}$$

where the $z$-component is the vertical direction and $x$ is in the longitudinal direction of the tank. As the motion is planar, no source term is added for the $y$-component. $\theta$ is the rotational angle. $\dot{\theta}$, $\ddot{\theta}$ is angular velocity and acceleration respectively. The coordinates $x_m$ and $z_m$ are center of the rotational point, and therefore $x - x_m$ is the distance from the rotational point to the center of the tank fixed coordinate system. The second last term of the $z$-component is the *Coriolis* acceleration.

### 2.2. Turbulence

Modelling turbulence in sloshing, or general free surface flow with large density ratios, is complex. To calculate the velocities and pressure of Eq. (1), an expression for the eddy-viscosity is needed. The two-equation, $k$-$\omega$ turbulence model is used to close the set of equations (Wilcox, 1994). These are the kinetic turbulent energy and the specific dissipation rate of turbulent energy, $\omega$. The equations can be written (Wilcox, 1988):

$$\frac{\partial k}{\partial t} + \bar{u}_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\nu + \frac{\nu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k - \beta_k k\omega \tag{3}$$

$$\frac{\partial \omega}{\partial t} + \bar{u}_j \frac{\partial \omega}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\nu + \frac{\nu_t}{\sigma_\omega}\right)\frac{\partial \omega}{\partial x_j}\right] + \frac{\omega}{k}\alpha P_k - \beta\omega^2 \tag{4}$$

$P_k$ is the turbulent energy production term. $\sigma_k$, $\sigma_\omega$ are standard coefficients in the model, both with values of 2 in this case. $\beta_k$, $\beta$ and $\alpha$ are empirical constants, with values 9/100, 3/40 and 5/9 respectively. The RANS model overproduces the turbulent energy in highly strained flows. This gives unrealistically large values for the eddy-viscosity. Menter (1994) noted that the stress intensity ratio scales with the ratio of turbulence production to dissipation. Typical stress intensity ratios can be found from experiments in certain type of flows. In order to avoid overproduction of turbulence in highly strained flow outside the boundary layer, the turbulent eddy-viscosity, $\nu_t$, can be bounded through the limiting formulation (Durbin, 2009):

$$\nu_t = \min\left(\frac{k}{\omega}, \sqrt{\frac{2}{3}}\frac{k}{|\mathbf{S}|}\right) \tag{5}$$

where $|\mathbf{S}|$ is the rate of strain.

The rough wall function by Schlichting (1979) is applied to solid boundaries (Bihs and REEF3D, 2016):

$$u^+ = \frac{1}{\kappa} ln\left(\frac{30y}{k_s}\right). \tag{6}$$

$u^+$ is the dimensionless wall velocity, $\kappa$ is a constant equal to 0.4 and $k_s$ is the equivalent sand roughness. Near the wall it is assumed that the turbulent production is equal to the dissipation of turbulent energy. This gives the following expression for the specific turbulent dissipation:

$$\omega_{wall} = -\frac{C_\mu^{3/4} k_w^{1/2} u_w^+}{\Delta y_p} \tag{7}$$

where $\Delta y_p$ is the distance from the wall to the respective cell.