Coupled Simulation of Nonlinear Ship Motions and Free Surface Tanks

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ABSTRACT

This paper investigates the coupled nonlinear dynamics of a vessel with a free surface tank onboard. To this end, a 6-DOF ship motions simulation code is coupled with a CFD solver addressing the behaviour of the fluid in the tank. The nonlinear ship motions code is of the blended (hybrid) type, intended for the simulation of free running vessels in waves. The nonlinear CFD solver is a GPU-based 3D Weakly-Compressible Smoothed-Particle Hydrodynamic (WCSPH) solver. Numerical results are presented for the nonlinear roll motion of a vessel with and without a free surface tank in regular beam waves with different steepnesses.

Keywords: *nonlinear ship motions; sloshing; smoothed-particle hydrodynamics (SPH); GPU; anti-rolling tanks; 6-DOF simulations; blended codes; coupling*

1. INTRODUCTION

Tanks characterised by the presence of a free surface are almost invariably present onboard vessels, with different scopes: fuel tanks, ballast tanks, cargo tanks, anti-rolling devices, etc. While taking exactly into account their effect on static restoring is, nowadays, a matter of routine stability calculations, the same cannot be said when ship dynamics and fluid cargo dynamics are to be accounted for in a coupled way. Due to the complexity of the involved phenomena, a coupled dynamic approach is particularly challenging when nonlinear effects are to be considered in both ship motions and fluid dynamics in the free surface tanks.

Different approaches have been used in the past to simulate the behaviour of a vessel in presence of liquid tanks onboard. Fully linear approaches for ship motions, internal hydrodynamics and external hydrodynamics, have been developed by Malenica et al. (2003) and Kim & Shin (2008). Such approaches are very suitable for design purposes in mild sea conditions. However, when sloshing within the tanks becomes violent and/or ship motions become large, the linearity assumption become too restrictive and the underlying models fail to reproduce the actual fluid and ship dynamics. As a result, nonlinearities need to be introduced, and different authors, recognising this need in certain conditions, have tackled the problem with approaches having different levels of sophistication.

In case ship motions can be considered small enough to be treated linearly, nonlinear effects can be introduced only in the numerical solution of the sloshing problem. Approaches along this line can be found, for instance, in

(Kim et al. ,2007, Zhao et al., 2014), where nonlinear time domain potential flow approaches are used under the assumption of a free surface retaining a single-valued behaviour. However, this assumption does not allow taking into account strong nonlinear phenomena such as free surface fragmentation or wave breaking, which characterise violent sloshing. The possibility of handling complex, non-single valued, free surface dynamics was instead introduced in the work of Bunnik & Veldman (2010), where a VOF solver for the internal sloshing flow was coupled in time domain with a linear ship motions model handling the linear potential external fluid-structure interaction and the linearized rigid body dynamics.

However, there are many situations when linear approaches to ship motions are insufficient. This is, for instance the case when the interest is on the assessment of ship behaviour in severe environmental conditions, or when the interest is on typically nonlinear dynamic stability phenomena in waves (e.g. parametric roll, pure loss of stability, surf riding and broaching, large rolling amplitudes in beam waves – see IMO (2009)), or when the interest is on the simulation of the behaviour of a vessel, having free surface tanks onboard, and which is free running in waves. In all such, and other, cases, nonlinear models need to be used for simulating the dynamics of the vessel. Approaches making use of nonlinear ship motions models together with simplified models for the behaviour of the fluid in the tank can be found in Francescutto & Contento (1999) for the beam sea case, and in Neves et al. (2009) for the case of longitudinal sea and, in particular, parametric roll. 6-DOF ship motions models coupled with 1-DOF U-tube tank models have been reported by Youssef et al. (2003) and Holden & Fossen (2012).

More sophisticated models are required when nonlinear effects are to be introduced in both ship motions and in the solution of the fluid flow in the tank. Nonlinear effects in the fluid flow can become particularly relevant in case of tanks featuring large free surfaces. Along the line of increasing the accuracy of the CFD solver for the internal flow, Hashimoto et al. (2012) coupled a nonlinear 1-DOF roll motion model for the simulation of parametrically excited roll motion, with a fully nonlinear solution of the fluid flow in the tank using the Moving Particle Semi-implicit (MPS) method, which is able to take into account strongly nonlinear free surface flows. In Mitra et al. (2012) a nonlinear potential flow model was solved by FEM for the internal tank, assuming the free surface to be single valued (therefore, also in this case, free surface fragmentation, breaking and strong nonlinearities cannot be accounted for), and the coupling was done with a partially nonlinear 6-DOF ship motions model.

In this study an approach is used where a 6-DOF ship motions simulation code is coupled with a CFD solver addressing the behaviour of the fluid in the tank. The nonlinear ship motions code is of the blended (hybrid) type, intended for the simulation of free running vessels in waves. The nonlinear CFD solver is a 3D Weakly-Compressible Smoothed-Particle Hydrodynamic (WCSPH) solver, allowing the use of graphical processing units (GPUs). In the following, the simulation tool is firstly described. Then, numerical results are presented for the nonlinear roll motion of a vessel with and without a free surface tank in regular beam waves with different steepnesses.

2. SIMULATION TOOL

The tool developed in the present study is intended to be able to simulate the general case of nonlinear motions for a free running ship sailing in regular or irregular waves, with a liquid tank onboard. Since nonlinear motions and nonlinear fluid flow inside the tank are of interest, and since the tool is expected to be able to deal with the general case of a ship free running in waves, linear frequency domain approaches (Malenica et al., 2003, Kim & Shin, 2008) do not represent a relevant option for the scope of the study. Although research is ongoing (Sadat-Hossein et al., 2010, Carrica et al., 2012) regarding the use of direct computational fluid dynamics approaches for nonlinear ship motions of, possibly free running, ships in waves, the required computational time and resources are still prohibitive for practical applications.

Considering the situation, herein an intermediate approach has been followed, where the nonlinear rigid body dynamics and the ship-waves interaction is dealt with by means of a blended (hybrid) nonlinear 6-DOF approach, while the internal fluid-structure interaction, i.e. the fluid dynamics within the tank, is handled through a CFD approach based on a fully nonlinear SPH solver. The two tools are then coupled, in order to incorporate the tank effects in the solution of ship motions.

In particular, the ship dynamics is handled by the 6-DOF blended simulation code SHIXDOF ("nonlinear SHIp motion simulation program with siX Degrees Of Freedom"), under development at the University of Trieste. The code has been described and applied previously in (Bulian et al., 2012, Bulian & Francescutto, 2013) and herein some main details are reported.

The simulation approach used in SHIXDOF is a typical hybrid approach along the line of de Kat & Paulling (1989). To date, approaches of such type have been considered suitable for practical assessment of nonlinear ship motions in waves, and their suitability for such purpose has been stated also in the framework of IMO "Second Generation Intact stability Criteria" (Bulian & Francescutto, 2013, IMO, 2010, 2013). As described in some more details by Bulian et al. (2012) and Bulian & Francescutto (2013), SHIXODF solves nonlinear rigid body motions equations with respect to the ship-fixed reference system:

$$\begin{cases} m \cdot \begin{bmatrix} \underline{u}_O' + \underline{\omega} \wedge \underline{u}_O + \\ + \underline{\omega}' \wedge \underline{x}_G + \underline{\omega} \wedge \left(\underline{\omega} \wedge \underline{x}_G \right) \end{bmatrix} = \underline{F}_{ext}\left(t\right) \\ \\ \underline{\underline{I}}_O \cdot \underline{\omega}' + \underline{\omega} \wedge \left(\underline{\underline{I}}_O \cdot \underline{\omega} \right) + m \cdot \underline{x}_G \wedge \underline{u}_O' + \\ + m \cdot \underline{x}_G \wedge \left(\underline{\omega} \wedge \underline{u}_O \right) = \underline{M}_{ext,O}\left(t\right) \end{cases} \quad (1)$$

The vessel is then moved and oriented with respect to an earth-fixed reference system. The external force $\underline{F}_{ext}\left(t\right)$ and moment $\underline{M}_{ext,O}\left(t\right)$ comprise the following main effects: Froude-Krylov pressure, including hydrostatic term, calculated up to the instantaneous wetted surface of the hull (to catch geometrical nonlinearities); linear hydrodynamic radiation terms through convolution of kernel functions and infinite frequency added mass terms obtained from linear potential flow pre-calculations; instantaneous diffraction forces from linear frequency domain pre-calculations; manoeuvring forces, comprising a cross-flow model. Furthermore, it is possible to consider: constant and gusty wind effects; additional empirical damping terms (typically for, but not limited to, roll); linear/nonlinear, mooring-like springs; propulsors; lifting surfaces (rudders, fins).

In addition to the abovementioned effects, in the simulation tool developed herein, $\underline{F}_{ext}\left(t\right)$ and $\underline{M}_{ext,O}\left(t\right)$ also contain the instantaneous action, on the vessel, of the fluid in the tank. Such actions are calculated by the coupled CFD solver, which is based on the numerical solution of the 3D fluid field through a meshless Smoothed-Particle Hydrodynamics (SPH) approach.

The SPH approach has become very popular in CFD field thanks to the adaptability to complex geometries, and the capability of dealing with heavily fragmented fluids, while keeping a reasonable computational cost. The particular solver used herein is AQUAgpusph (Cercos-Pita et al., 2013, Cercos-Pita, 2015), developed at University of Madrid. To address the actually incompressible flow, AQUAgpusph uses the commonly employed weakly-compressible SPH approach (WCSPH) (Monaghan, 2005, Colagrossi et al., 2009), which is based on the solution of the Navier-Stokes equations, where an artificial weak compressibility is considered through a pressure-density state equation which provides small density variations:

$$\begin{cases} \dfrac{d\rho_a}{dt} = -\rho_a \nabla \cdot \underline{u}_a \\ \dfrac{d\underline{u}_a}{dt} = -\dfrac{\nabla p_a}{\rho_a} + \dfrac{\mu}{\rho_a}\Delta \underline{u}_a + \underline{g} \\ p_a = p_a(\rho_a) \end{cases} \tag{2}$$

AQUAgpusph solves the discretised version of (2) using the Lagrangian kernel-based SPH formalism. Other formulations can be found in order to perform truly incompressible SPH simulations (e.g. Cummins & Rudman, 1999, Souto-Iglesias et al., 2014), but the WCSPH formulation has the main benefit that a purely explicit scheme can be used to perform the integration, and hence, no linear system of equations needs to be solved in order to compute the pressure field at each time step. In order to speed up the computation, AQUAgpusph can exploit, through OpenCL, the parallel computing capabilities of graphical processing units (GPUs), if such hardware, as in the present application, is available (Cercos-Pita et al., 2013, Cercos-Pita, 2015).

To allow coupled simulations, an explicit coupling strategy has been implemented, where SHIXDOF performs the time stepping by means of an explicit integration scheme, receiving force and moment from AQUAgpusph at the beginning of each step, and passing to AQUAgpusph the updated ship and tank motions at the end of the step. With such information, AQUAgpusph simulates the fluid motion in the tank within the considered time step, while SHIXDOF waits to receive the force and moment at the beginning of the next time step. The integration in SHIXDOF is carried out by means of a 4th-order Adams-Bashforth integration scheme with fixed time step. On the other hand, AQUAgpusph integrates in time by means of a Leap-Frog method (Souto-Iglesias et al., 2006) with variable time step controlled by a Courant condition.

3. APPLICATION

An application of the developed nonlinear coupled simulation tool has been carried out using a freely available and well-known hull form geometry. This allows present results to serve as possible comparison cases for other researchers developing similar tools. Furthermore, the considered hull has been selected because experimental data regarding nonlinear rolling motion without tank were available from previous studies.

The geometry and positioning of the tank in the simulations was chosen, and constraint, to be compatible with an already existing 1:100 scale model of the hull. As a result of such choice, the positioning of the tank in the simulations is quite high above the waterline and above the centre of gravity.

Simulations have been targeted at assessing nonlinear effects on the roll response curve, coming from external hydrodynamics (ship-wave interaction) and internal hydrodynamics (ship-tank interaction). To this end, numerical experiments have been carried out, for one specific tank geometry, in regular beam waves having different steepnesses (ratio between wave height and wave length).

3.1 Sample hull and free surface tank

The simulation tool has been used for simulating the behaviour of a Series 60 hull form, in bare hull condition, and equipped with a box-shaped free surface tank. The bodyplan of the hull, together with a transversal view of the tank geometry is shown in Figure 1, while the main characteristics of the hull and the loading condition (with empty tank) are reported in Table 1. Such hull form, without any tank onboard, was used in previous numerical and experimental studies regarding nonlinear roll motion in beam regular, bi-chromatic and irregular waves (Tzamtzis, 2004, Bulian et al., 2012, Bulian & Francescutto, 2013). As a result, a certain amount of reference experimental data was available for the present study.

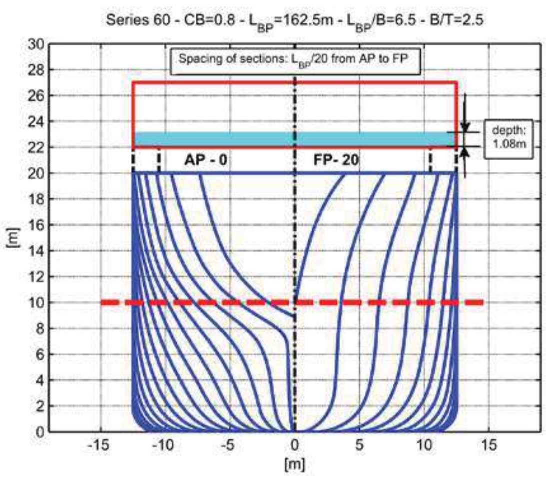

Figure 1: Hull bodyplan and tank geometry.

Table 1: Main data of hull and loading condition (without tank).

L_{BP}	[m]	162.5	Length b.p.
B	[m]	25.0	Beam
T	[m]	10.0	Draught
C_B	[-]	0.8	Block coefficient
$\overline{GM}$	[m]	1.65	Transversal metacentric height
$\overline{KG}$	[m]	8.59	Height of CoG above baseline
ω_0	[rad/s]	0.408	Roll natural frequency
$R_{xx,G}$	[m]	9.1	Dry roll radius of inertia (w.r.t. CoG)
$R_{yy,G}$	[m]	40.6	Dry pitch radius of inertia (w.r.t. CoG)
$R_{zz,G}$	[m]	40.6	Dry yaw radius of inertia (w.r.t. CoG)

The tested tank has the main characteristics reported in Table 2. The tank is longitudinally positioned at the mid perpendicular, spanning a total length of 10m in longitudinal direction (5m aft and 5m forward of the mid perpendicular). The transversal width of the tank corresponds to the ship breadth, and the fluid depth is set in such a way to obtain a first transversal linear natural sloshing mode matching the roll natural frequency of the vessel with empty tank. In such configuration, the ratio between the mass of the fluid in the tank and the mass of the vessel with empty tank is 0.83%. The increase of draught due to the additional weight loaded in the tank is 76mm, i.e. 0.76% of the ship draught without fluid in the tank. It is also to be noted that the depth to width ratio for the fluid in the tank is 0.0432, meaning that sloshing occurs in a shallow water regime. Even under purely static inclinations, the bilge corner of the tank becomes dry at a heel angle of just 4.9deg. It is herein assumed that the hull without tank has the same mechanical properties (mass, position of centre of gravity, radii of inertia) of the hull equipped with the empty tank. As a result, the indications “without tank” and “empty tank” are to be assumed, herein, as synonymous.

The righting lever ($\overline{GZ}$) curve has been calculated for the vessel without tank and with the tank. For sake of comparison, the calculation of the righting lever with the tank was carried out considering the cargo as both solid (frozen) and fluid. Results are shown in Figure 2. Since the liquid cargo is loaded high above the baseline, part of the reduction in the righting lever is due to the increase of $\overline{KG}$ (the variation of $\overline{KM}$ due to the small variation of draught, associated with the loading of the fluid in the tank, is very small). Then, the majority of the reduction in the restoring is due to the free surface effect. As a consequence of the large fluid depth to tank width ratio, free surface effects are practically linear with respect to the heeling angle until the tank bilge corner gets dry (abt. 5deg) then the overall free surface effect reduces as the heeling increases.

Table 2: Main characteristics of the tank.

Tank dimensions L_{tank} x W_{tank} x H_{tank}	10m x 25m x 5m
Longitudinal position	Centre at mid perpendicular
Height of tank bottom from baseline	22m
Fluid depth - d_{fluid}	1.08m
d_{fluid} / W_{tank}	0.0432
Filling ratio	0.2160

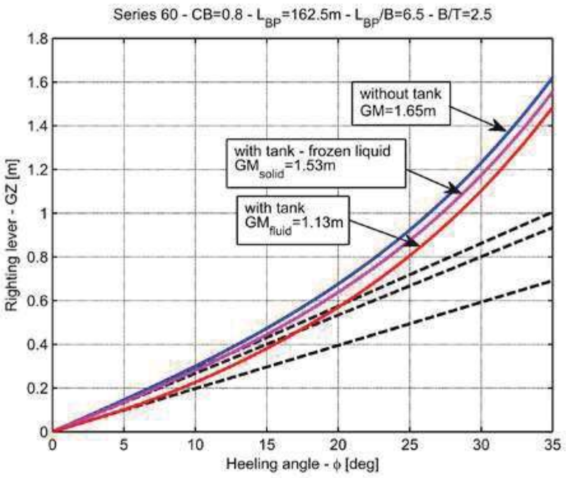

Figure 2: Righting lever curves without tank, with frozen liquid in the tank and with fluid in the tank.

3.2 Roll motion in regular beam waves without free surface tank

Before running a set of coupled simulations, the 6-DOF software tool has been compared and tuned, in terms of additional roll damping coefficients. The tuning has been performed making reference to a set of experimental data without tank for roll motion in regular beam waves at zero speed (Tzamtzis, 2004). A first tuning of the 6-DOF code on roll decay experimental data for the considered loading condition was carried out by Bulian & Francescutto (2013). Herein the tuning has been improved to achieve a better matching between simulations and experimental roll response at large forcing wave steepnesses in beam regular waves, while still keeping a good matching with roll decay data and experiments in milder regular beam waves.

In the tuning process, the drag coefficient used in the cross flow model has been kept constant to a value equal to 0.8, which is in line with typical lateral drag coefficients for quite full vessels with similar beam to draught ratios (e.g. Kijima, 2003, Faltinsen, 1990). In view of the experimentally observed behaviour of equivalent linear roll damping coefficient as a function of the oscillation amplitude from roll decays, the tuning of damping in roll was carried out through an additional empirical linear-in-velocity term ($-B_{L,add} \cdot \dot{\phi}$) and an additional cubic-in-velocity term ($-B_{C,add} \cdot \dot{\phi}^3$). Such terms have been added to the moment acting on the vessel around the longitudinal axis of the ship-fixed reference system. It is to be noted that, as a result, such additional damping terms are not independent from the considered reference system.

Simulations have then been carried out in regular beam waves, without tank, for different frequencies close to the roll natural one, and considering two wave steepnesses, 1/100 and 1/30, as in the experimental conditions (Tzamtzis, 2004, Bulian & Francescutto, 2013). In both experiments and simulations the vessel was free to drift. In the experiments the beam sea condition was maintained by manual control, while in the simulations a linear, with respect to yaw, restoring moment directed along the earth-fixed vertical axis and with spring constant equal to $5.3 \cdot 10^8 N \cdot m / rad$ was used in order to keep the heading at about 90deg. The introduction of this artificial spring leads to a yaw natural frequency, as measured from yaw decays, of $0.0673 rad/s$, which is far enough from the roll natural frequency to reduce the risk of spurious couplings. The comparison between experimental results and results from simulations is shown in Figure 3 (at ship scale). Although there is still a small overestimation of the experimental roll amplitude at the larger forcing steepness, the re-tuning of additional roll damping has led to a reduction in the difference between experimental and numerical maximum roll response at $s_W = 1/30$, from about 15% in Bulian & Francescutto (2013), to about 9% herein. The difference in the peak roll between experiments and simulations for $s_W = 1/100$ is, instead, 2%. It can be noticed that the bending towards high frequencies of the roll response curve, associated with the hardening roll restoring, is well captured by the simulations.

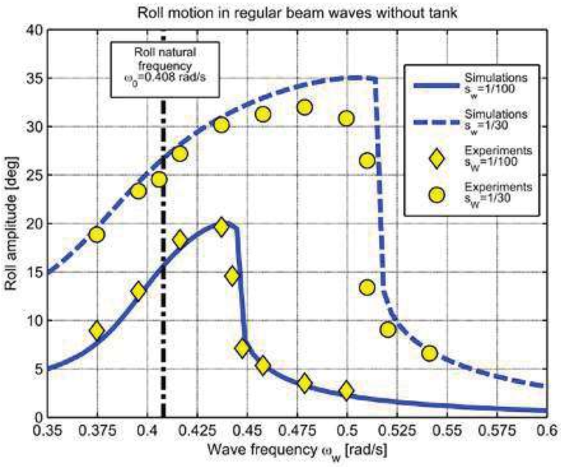

Figure 3: Roll motion in regular beam waves, without tank.

In order to analyse the obtained roll dissipation level in calm water after tuning the ship motions code, a numerical roll decay has been simulated. The resulting roll time history has then been analysed on the basis of the classical 1-DOF nonlinear model:

$$\begin{aligned}&\ddot{\phi} + 2 \cdot \mu \cdot \dot{\phi} + \beta \cdot \dot{\phi}\left|\dot{\phi}\right| + \delta \cdot \dot{\phi}^3 + \omega_0^2 \cdot r(\phi) = 0 \\ &\text{with } r(\phi) = \phi + \gamma_3 \cdot \phi^3 + \gamma_5 \cdot \phi^5 + \ldots\end{aligned} \tag{3}$$

Results from the roll decrement analysis according to the methodology described by Bulian et al. (2009), lead to $\omega_0 = 0.408 rad/s$, $\mu = 0.00340 s^{-1}$, $\beta = 0.0994 rad^{-1}$ and $\delta = 0.554 s \cdot rad^{-2}$.

On the basis of the reported results, it can therefore be concluded that the 6-DOF nonlinear ship motions code can be considered validated for the intended scope of this study.

3.3 Validation of the SPH solver

The SPH solver has been validated by simulating the SPHERIC validation test 9 (Botia-Vera et al., 2010, Bulian et al., 2010), for which data are available from https://wiki.manchester.ac.uk/spheric/ under the "Validation Tests" section.

The validation test 9 consists in a simplified 1-DOF mechanical model of a tuned liquid damper (TLD), where a rectangular tank is allowed to rotate around a fixed point under the forcing of a translating mass with prescribed oscillatory motion. The motion of the mass is rectilinear in the tank-fixed reference system. The tank is partially filled with liquid, and the resulting system is, therefore, a 1-DOF mechanical system coupled with the action of the fluid inside the tank. Such system, can also be considered as a simplified model relevant for the dynamics of a vessel equipped with a free surface tank.

The equation of motion of the coupled system, in such condition, and the values for the model parameters can be found in (Pérez-Rojas et al., 2011, Botia-Vera et al., 2010, Bulian et al., 2010) and can also be obtained from the already mentioned SPHERIC website.

Figure 4 shows a comparison between experiments and simulations carried out with a total of about 100000 particles. The very good agreement between predictions and experimental outcomes can be noticed. The SPH solver can therefore be considered suitable for the intended purpose of this study.

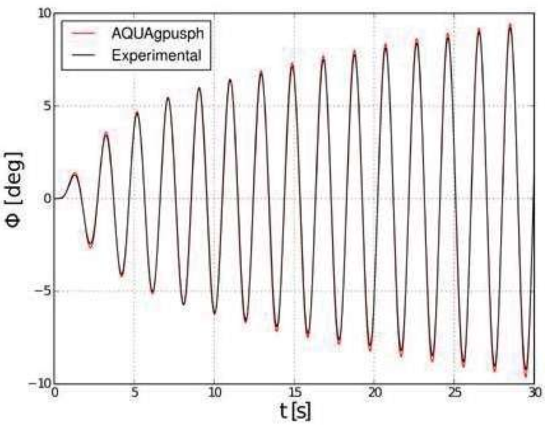

Figure 4: Roll angle of the tuned liquid damper. Comparison between SPH simulation and experiments. Amplitude of the motion for the moving mass: $100\,mm$. Forcing frequency equal to natural frequency of the dry system.

3.4 Roll motion in regular beam waves with free surface tank

After checking the capability of the 6-DOF nonlinear ship motions code to reproduce experimental data without the effect of the free surface tank, and after checking the capability of the SPH solver to properly reproduce the fluid action in a simplified 1-DOF coupled TLD model, a series of 6-DOF coupled ship-tank simulations have been carried out considering the tank partially filled with fluid as reported in Table 2.

Coupled 6-DOF simulations have been carried out in beam regular waves at zero speed, with the same numerical setup used for simulating the motion of the vessel without tank. The ship was free to drift, but rotations around the earth fixed vertical axis were partially restrained as in the case without tank. The number of particles used for the fluid discretisation in the SPH solver was set to about 12000.

The primary scope of the simulations described herein was to analyse the behaviour of the coupled system for different levels of the wave forcing. Roll motion is known to behave nonlinearly as the wave forcing increases. Similarly, it can be expected to observe a nonlinear behaviour also for the action on the vessel of the fluid inside the tank as the motion of the tank boundaries, which are forcing the fluid, increases. For the considered tank this is particularly expectable, as a consequence of the small depth to width ratio, and the associated very shallow water regime in which the tank is working. To this end, four wave steepnesses have been tested, namely: 1/80, 1/67, 1/57 and 1/50. A specific set of wave frequencies was simulated for each steepness in order to have a clear representation of the roll response curve.

To allow an assessment of the effectiveness of the tank as a passive anti-rolling device, simulations have been carried out for the same four steepnesses also for the vessel without tank.

The total length of each simulation was set to 500s, with an initial ramp of 50s on the wave forcing. A 2s pre-stabilization of the SPH solver is performed before starting each simulation. The average roll amplitude was measured considering the final part of each simulation. Herein, the roll amplitude is defined as half of the difference between maximum and minimum roll within each cycle. The roll response curves resulting from the simulations are shown in Figure 5.

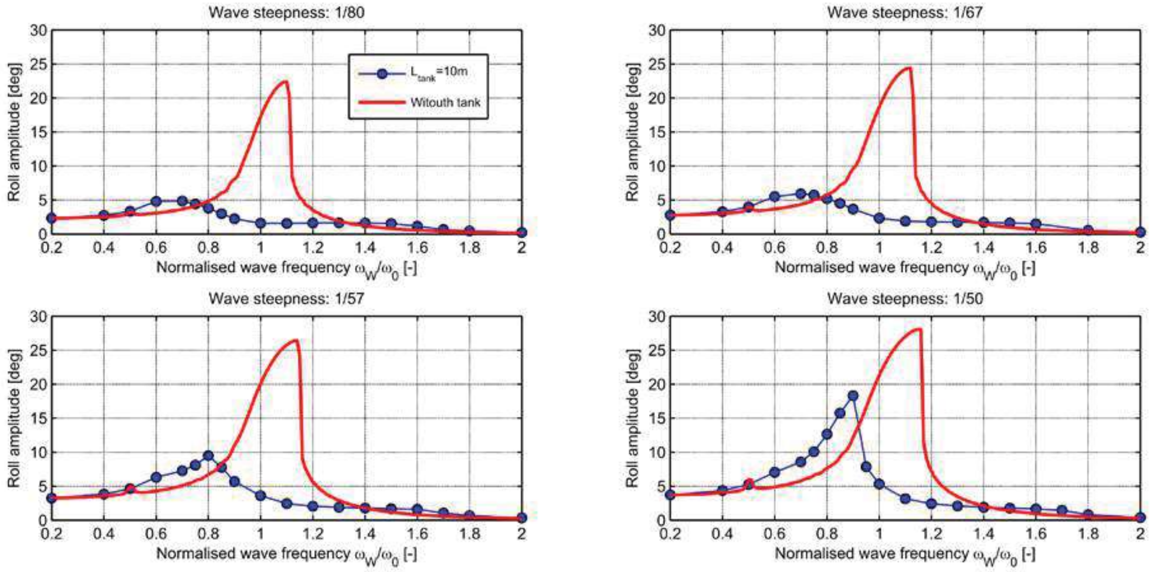

Figure 5: Roll response in regular beam waves with different steepnesses, with and without tank.

From the results reported in Figure 5, it can be noticed that the vessel without the tank follows the classical well-known nonlinear behaviour in regular beam waves, with a peak roll response which increases less-than-linearly as a function of the wave steepness, and a bending of the response curve towards the region of high frequencies, in accordance with the hardening behaviour of the roll righting moment in calm water. A small secondary peak is also visible at $\omega_W \approx 0.5\omega_0$, particularly for the largest steepnesses. Such small peak is associated with the inception of ultra-harmonic roll motions (Cardo et al., 1981), where the roll response, in addition to the harmonic at the encounter frequency, features a significant harmonic at twice the encounter frequency of the forcing.

Looking at the behaviour of the roll response with the tank partially filled with fluid, it can be noticed that the effect of the tank, as an anti-rolling device, is very significant in case of the two lowest forcing steepnesses ($s_W = 1/80$ and $s_W = 1/67$). In such cases the roll response shows the well-known double-peak shape (e.g., Field & Martin, 1976, Lee & Vassalos, 1996, Francescutto & Contento, 1999, Kim et al., 2007, and discussion by Bell in van den Bosch & Vugts, 1966): in the frequency region close to the roll natural frequency the roll motion is strongly suppressed, while, on the other hand, the roll response with the tank is larger than without the tank in the frequency regions close to the two peaks appearing at low and high frequency.

However, the roll motion has a totally different behaviour for the largest forcing steepness ($s_W = 1/50$). In such case, the maximum roll amplitude with tank is much closer to the maximum roll amplitude without tank, and the effectiveness of the tank, as an anti-rolling device, is very significantly reduced. Looking at the shape of the response curve, although a small high-frequency peak is still present, the roll response with tank approximately resembles the one without tank, with a shifting towards lower frequencies and a reduction in the maximum peak. A small but noticeable hump is still present in the frequency region corresponding to the low frequency peak for at $s_W = 1/80$ and $s_W = 1/67$, as a reminiscence of the behaviour of the roll response curve for small wave forcing. The case of forcing steepness $s_W = 1/57$ seems to represent, instead, a sort of transition case. In this case, the roll response behaves, in part, similarly to the case of small wave forcing. However the inception of the peak which will then become dominant at $s_W = 1/50$, is already noticeable at wave frequencies around $0.8\omega_0$. In the same frequency region, the transitional character of this forcing condition manifests also in time domain, as long transients before stationarity.

The behaviour described above on the basis of Figure 5 is more evident when plotting the roll response curves with tank for different steepnesses, on the same graph, as shown in Figure 6. From the response curves in Figure 6 it is also interesting to note that, similarly to the case without tank, also in case of vessel equipped with the tank the roll response curve, in the region of the low-frequency peak, tends to bend towards higher frequencies. Furthermore, it is also interesting to report that the observation of the time histories at $\omega_W = 0.5\omega_0$ indicates that, for all forcing steepnesses, the presence of the tank is able to suppress the inception of the small ultra-harmonic response which was instead observed in the simulations without tank.

Finally, Figure 7 shows the simulated fluid behaviour inside the tank for two example cases. The two cases correspond to the frequency ratio $\omega_W / \omega_0 = 0.9$ for the minimum and maximum simulated wave steepness, i.e. $s_W = 1/80$ and $s_W = 1/50$ respectively. The reported snapshots are taken at four representative time instants within the last available roll cycle, corresponding to: minimum roll, up-zero-crossing for roll, maximum roll and down-zero-crossing for roll. It can be noticed that impacts on the tank side take place at the lowest steepness. At the highest steepness, where the rolling amplitude exceeds 18deg, the fluid also impacts the top of the tank.

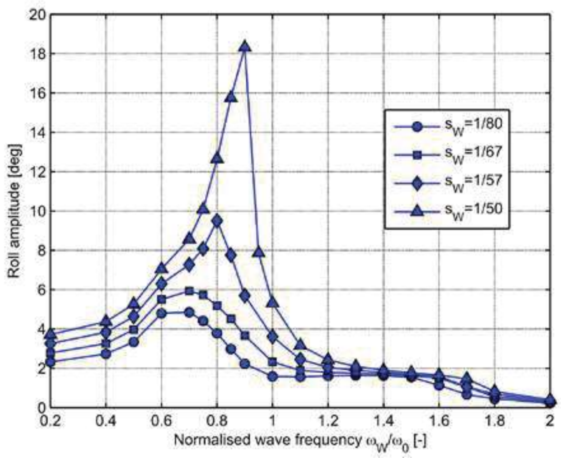

Figure 6: Roll response in regular beam waves with different steepnesses, with tank.

$\omega_W / \omega_0 = 0.9$ $s_W = 1/80$	$\omega_W / \omega_0 = 0.9$ $s_W = 1/50$
	

Figure 7: Representative snapshots of fluid inside the tank for two example cases.

4. FINAL REMARKS

Since vessels almost invariably sail with tanks partially filled by liquids, the analysis of ship motions in presence of free surface tanks onboard represents an interesting research topic, having also significant practical implications.

In this paper a time domain simulation approach has been presented where a blended (hybrid) nonlinear 6-DOF ship motions simulation code has been coupled with a nonlinear SPH solver intended to address the flow in the internal tank.

An application of the developed tool has been carried out for a Series-60 hull, with one rectangular tank meant to act as anti-rolling device. The tuning of the tank with the roll natural frequency of the vessel led to a small fluid depth to tank width ratio, and therefore a shallow water fluid regime.

The primary scope of the simulations described in the paper was to analyse the behaviour of the coupled system for different levels of the wave forcing, in order to highlight the possible occurrence of nonlinear behaviours. To this end, simulations have been carried out in regular beam waves with different steepnesses.

Before carrying out coupled ship-tank simulations, the 6-DOF ship motions code was tuned, in terms of roll dissipation, using available experimental data. Validation comparisons between simulations without tank and available experimental data indicated a good agreement. Similarly, the SPH solver was separately validated on the basis of available experimental data for a 1-DOF coupled mechanical system representing a tuned liquid damper.

Results of coupled ship-tank simulations have been reported in terms of roll response curves, with and without tank, for a range of frequencies. Outcomes from simulations have clearly shown the occurrence of nonlinear phenomena. The most notable behaviour was found to be a reduction of effectiveness of the anti-rolling tank as the wave forcing, and the consequent motions, increase. Also, simulations without tank showed the occurrence of a small ultra-harmonic roll response at wave frequencies close to half the roll natural frequency. Such type of response, instead, did not appear with the tank partially filled by fluid. Also, bending of the response curves, with and without tank, was observed, in line with the hardening restoring of the vessel.

Although the developed approach has been tested herein at zero forward speed in beam regular waves with a box-shaped tank, the software architecture is more flexible. Indeed, the 6-DOF code allows simulating ship motions in case of the vessel manoeuvring in regular/irregular waves, considering the coupling with the tank. Furthermore, the SPH solver allows taking into account more complex tank geometries (e.g. non box-shaped tanks, presence of baffles and obstructions, etc.). As a result, in addition to representing a valuable tool for research purposes, there are potentialities for this approach to be used for more practical engineering applications. Research is presently ongoing regarding the application of the present approach in different conditions and, in particular, in cases associated with different tank dimensions.

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