Sloshing effect, Fluid Structure Interaction analysis

C L Dumitrache and D Deleanu
Maritime University of Constanta, Faculty of Electromechanics, Department of General Engineering Sciences, Mircea cel Batran street, No. 104, Constanta, 900663, Romania

E-mail: ldumitr@yahoo.com

Abstract. The sloshing effect of the liquids is considered to be an important factor in area such as cargo ships (tankers) which transporting petroleum liquids or other liquids (water inside of ballast tanks). Sloshing can produce big problems so is essential to apply the prevention measures. For example in water ballast tanks, the sloshing motion of sea water inside of ballast tanks will bring large fluid pressures and impact loads, which might destroy the structure of tanks. It is known that ships with larger tanks, less internal structure, and more free surface area, the number of failures attributed to sloshing liquids have increased. Nowadays it is recognized that the ballast tanks which may be partly filled are the crucial group of tanks onboard. The problem of an assessment of liquid sloshing effect is more important than ever because of the obligatory ballast water management requirement. Our paper is an original fluid structure interaction analysis during sloshing phenomenon. We are using ANSYS CFX to simulate the behavior of two fluids, (seawater and air from the ballast tank), requires itself the multiphase flow model. Due to the movement of the ballast/cargo tanks is complex involving pitching and rolling of ship, it is clear that the simulation is done with the ANSYS CFX transient module. Because this complex simulation requires massive resources calculation we consider a simplified CAD models for a part of the oil chemical ballast tank structure.

1. Introduction
Nowadays, Fluid-Structure Interaction (FSI) modelling is a computational modelling technique in which the Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA) models are linked together so that the results of each model impart forces on the other and create FSI model. With this model two challenges are introduced [1]:

• The first challenge that arises when coupling CFD and FEA models together is coupling the two independent mesh domains together while still accounting for the differences in mesh formulation and motion. There is fundamental differences between the mesh utilized in FEA models (the Lagrangian mesh), which deforms as a function of mass motions, and the mesh utilized in CFD models (the Eulerian mesh), which is fixed at all points in space and time [2];
• The second challenge is to transfer data between domains in a manner that mitigates fluctuations, instabilities, and non-physical phenomena at the domain interfaces. These instabilities arise from the data transfer methods, the mass effect, and magnification of instabilities or shock waves at the interface.

Specialized studies in the field of FSI have shown the major importance of sloshing phenomenon in ships stability [3] and thus major regulations have been introduced in the IMO (International Maritime
Organization) which is responsible to implement and amend different codes as per types of ships, shipbuilding, maritime security goods or cargoes, cargo operation, the safety of the crew, training etc. 

For this reasons the stability measures such as righting arm curves and metacentric heights are in common use. The righting arm curves shall be corrected according to the IMO regulations (ISC - International Code for Intact Stability) using one of two accepted methods:

- correction based on the moment of inertia of tank’s horizontal projection (simple pendulum model);
- correction based on the actual moment of fluid transfer calculated for each angle of heel.

Both methods of free surface correction calculation consider the quasi-static attitude towards the sloshing phenomenon only. They do not consider the location of tanks within the hull of a ship and the location of the rolling axis. However, the main advantage of currently applied compulsory corrections is the simplicity of their calculation. The formula recommended by IMO for calculating free surface correction takes into account the fact that liquid surface is always flat and depends only the angle of ship’s heel not time.

The phenomenon of sloshing and its effects are related to the movement of free surface of a liquid inside of container. They are studied with great interest in industrial fields such as maritime and oil exploitation and refining. In the article [3] the researchers present the behaviour of the free liquid surface subjected to the dynamic loading which may result in liquid spilling or tank wall damage. This paper deals with the seismic design of the open cylindrical liquid storage tank with the aim to determine overall response of the liquid to an earthquake, dynamic properties (natural frequencies and modes of oscillation), maximum vertical displacements over tank radius.

In maritime industries it is known that the liquid sloshing effect is produced in partly filled ships tanks. For the ships that are moving, on the tanks partially filled, the energy transmitted from the sea waves acts from the outside, energies that influence the movement of liquids inside of the tanks [4, 5]. Both the liquid motion and its effects are called sloshing. The interaction between ship’s tank structure and water sloshing inside the tank consists in the constant transmission of energy [4, 6].

In this article the authors present the original transient and FSI analysis of the sloshing phenomenon which was performed over a time period of 16 seconds by moving the tank at a velocity \( v = 3 \text{m/second} \) in the direction of \( O_x \), corresponding to the pitching motion of the ship, also taking into account the expressions represents the height of the seawater column in the ballast tank, the water density, the volume of fluid moved by the sloshing effect and the relative pressure exerted by the fluid on the walls of the ballast tank.

2. The computational model

For the computational model we are using the three dimensional Navier–Stokes equations in conjunction with the continuity, volume fraction, and energy equations [1, 7]:

\[
\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = \nabla p + \nabla \cdot (\vec{t}) + \rho \vec{g} + \vec{F} \tag{1}
\]

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = S_n \tag{2}
\]

\[
\sum_{i=1}^{n} \alpha_i = 1 \tag{3}
\]

\[
\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = \nabla \cdot (k_{eff} \nabla T - \sum_j h_j \vec{J}_j + \tau_{eff} \cdot \vec{v}) + S_h \tag{4}
\]

where \( t \) is time, \( \rho \) is density, \( \vec{v} \) is the velocity vector, \( \nabla \) is the derivative in three-dimensional space, \( p \) is pressure, \( \vec{t} \) is the stress tensor, \( g \) is gravity, \( F \) is external body forces, \( S_n \) is a mass source term, \( \alpha \) is the fluid volume fraction, \( E \) is the total fluid energy, \( k_{eff} \) is the effective thermal conductivity of the fluid, \( T \) is the temperature, \( h \) is the enthalpy, \( J \) is the diffusion flux, and \( S_h \) is a volumetric energy source.
For the equations (1) through (4) respectively, we are utilizing a pressure based solver for subsonic incompressible flow, along with the k-epsilon turbulence model. For the calculation of pressure field we are using the equations (1) and (2) which represent the momentum and continuity equations.

Additionally, because the model contains two fluids with a discrete interface, the volume fraction equation must be solved to conserve species, and the mass balance equation must be evaluated to conserve the overall mass of the system, equations (2) and (3) respectively.

The equations (5) and (6) define k and epsilon turbulence model, and equation (7) define the turbulent viscosity:

\[
\frac{\partial}{\partial t} \left( \rho k \right) + \frac{\partial}{\partial x_i} \left( \rho ku_i \right) = \frac{\partial}{\partial x_j} \left( \mu \left( \frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \right) \right) + G_k + G_b - \rho \varepsilon - \rho \varepsilon' + S_k
\]

\[
\frac{\partial}{\partial t} \left( \rho \varepsilon \right) + \frac{\partial}{\partial x_i} \left( \rho \varepsilon u_i \right) = \frac{\partial}{\partial x_j} \left( \mu \left( \frac{\partial \varepsilon}{\partial x_i} + \frac{\partial \varepsilon}{\partial x_j} - \frac{2}{3} \frac{\partial \varepsilon}{\partial x_k} \right) \right) + C_{\mu} \frac{\varepsilon}{k} \left( G_k + C_{\mu} G_b \right) - C_2 \rho \varepsilon^2 + S_\varepsilon
\]

\[
\mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}
\]

where \( k \) is turbulent kinetic energy, \( u \) is velocity, \( \mu_t \) is the turbulent viscosity, \( G \) is generation of turbulent kinetic energy, \( \varepsilon \) represents fluctuation due to compressibility, \( S \) is a user-defined source term, \( \varepsilon \) is the rate of dissipation, and \( C_{\mu}, C_{\mu}, C_\varepsilon, \sigma_k, \sigma_\varepsilon \) are constants with all of the associated subscripts \( i, j, k, \) and \( t \) representing direction and time references. Each of these equations is defined for the fluid present in each control volume prescribed by the fluid mesh. If multiple fluid species or a volume of fluid model is evaluated, this set of equations will be evaluated for each fluid in the domain.

For the calculation of deformation, stress, strain, and forces across each node in the solid domain we are using the three-dimensional strain displacement, nodal displacement, and stress equations:

\[
[B] = [\varepsilon][N]
\]

\[
[\varepsilon] = [B][\delta]
\]

\[
[\sigma] = [E][\varepsilon]
\]

Where \( B \) is the strain displacement, \( \partial \) is the four-dimensional gradient (time and space), \( N \) is element shape function, \( \varepsilon \) is strain, \( D \) is nodal displacements, \( \sigma \) is stress, and \( E \) is modulus of elasticity.

### 3. The CAD models of ballast tanks

The original CAD models, made in NX Siemens, presented in the previous work were considered [8]. The dimensions of the ballast tanks are identical, one is simple and the other is optimized and has internal baffles that will influence the phenomenon of sloshing (figure 1). Due to the large volume of calculation, the dimensions of these tanks are much simplified and are part of the structure of ballast tanks from an oil chemical tanker with the dimensions presented in the previous work [8].

In the FSI analysis, the existence of welded joints was not taken into account, but the design of these tanks using NX Siemens took into account the existence of stress concentrators that can create very dangerous situations. It is known that in the construction of the tanks, naval steels are used which are in fact some low carbon alloy steels with mechanical properties such as plasticity, much improved. These steels are fine grained and in our FSI analysis made with ANSYS, Structural Steel with mechanical properties Yield Strength = 250 MPa and Ultimate Tensile Strength = 460 MPa was taken into account.

The discretization operation was performed with the Triangle Surface Mesher option, which generated for the simple tank a number of 561382 nodes and 280640 elements, and for the optimized tank a number of 639241 nodes and 320122 elements (figure 1).
The dimensions for the tanks are in millimeters and identical for the two models. It can be seen that this tank consists of two symmetrical vertical side faces, marked "symm1" (the face in the immediate vicinity of the viewer) and "symm2", located at a distance of 3 meters defined as boundaries limits "symmetry" type, and the winding surfaces of the tank, comprising the vertical wall coinciding with the position of the ship's keel and other unsymmetrical surfaces defined as boundaries limits "wall" type, all of them are marked with "wall".

4. Fluid Structure Interaction (FSI) analysis

In the previous paper [8] we presented the transient modeling of the sloshing effect using CFX ANSYS knowing that this modeling involves the interaction of two phases, the air-liquid inside the tanks and the amount of seawater reaches 0.6 m from the height of the tanks.

In figures 2(a), 2(b) are presented the average values of pressure exerted by the fluid during the sloshing effect on the three surfaces "symm1", "symm2", "wall" defined as boundary conditions. It can be seen from figure 2(a) that immediately after the start of the sloshing the pressures have negative values and after an interval of 2 seconds the pressure fields vary as some harmonic oscillations (waves) that are hard to amortize over time. In figure 2(b) the pressure fields are very quickly depreciated because the obstacles inside the tank greatly reduce the sloshing, and after 4 seconds the pressure fields become constant.

Figure 1. The mesh structure for simple ballast tank and optimized ballast tank.

Figure 2. Average pressure exerted by the fluid on: a-simple ballast tank; b-optimized ballast tank.
Different graphs were drawn representing the pressure exerted by the fluid at different time intervals of the sloshing phenomenon [8]. In figure 2(a) we observe the effect of harmonic oscillation of the fluid pressure, due firstly to the large free surface of fluid, while in figure 2(b) the sloshing effect is diminished due to the obstacles interposed inside the tank.

At very small time intervals from the beginning of the sloshing phenomenon, the pressure exerted by the fluid on the tank surfaces is negative. Next we will perform FSI analysis in situations when the pressure reaches the highest values as follows:

- for the simple tank, the pressure exerted on the surface "symm1" is 3 Pa and acts after two seconds from the beginning of the sloshing effect (figure 2(a), figure 3);
- for the optimized tank, the pressure exerted on the "wall" surface is almost 3 Pa and acts after one second from the beginning of the sloshing effect (figure 2(b), figure 5).

![Figure 3. Total pressure at 2 sec from CFX module at simple ballast tank.](image)

![Figure 4. Total pressure at 2 sec imported in transient structural module for simple ballast tank.](image)
The CAD design of these tanks did not take into account the existence of welded joints, and their design took into account the existence of stress concentrators that can create very dangerous situations.

The import of the pressure field from the CFX ANSYS module (figure 3, figure 5) into the ANSYS Transient Structural module (figure 4, figure 6) was done after previously defined separation surfaces between the air-liquid mixture and the surfaces that are “washed” by the fluid mixture inside the tanks.

In our FSI study the moments were taken into consideration when the pressure values exerted by the fluid reached the maximum values during the sloshing effect; that is for the simple tank the pressure field was imported on the surface “symm1” at the time of 2 seconds (figure 4), and at the optimized tank the pressure field was imported into the “wall” surface at the time of 1 second (figure 6).

It is worth mentioning that during the sloshing effect the pressure fields act cumulatively on all the surfaces defined by us as a boundary condition, and in this FSI study we only considered the pressure fields that reach maximum values only on certain surfaces. It can be deduced that the pressure values are not high because they are produced by a low speed of the ship and a relatively small amount of water that is in the ballast tanks. The situation would have been completely different if these tanks had been larger, the quantity of water would have been higher and the speed of the ship would have been

Figure 5. Total pressure at 1 sec from CFX module at optimized ballast tank.

Figure 6. Total pressure at 1 sec imported in transient structural module for optimized ballast tank.
higher. All of the above would have required considerable computational effort and higher computational resources.

Figure 7 shows the total deformations and as expected they are larger at the edge of the board more precisely at the starboard side extremity. The zero deformation values are in the area of the keel vessel, where we applied a "fixed constrained" surface.

![Figure 7. Total deformations.](image)

The values of equivalent stresses shown in figure 8 clearly show the influence of the tensile concentrators on the crossings from one surface to the other through an angle of 90 degrees. As mentioned above, the existence of welded joints was not taken into account, but these crossings can anticipate when dangerous situations may occur. The maximum values presented in these figures with that red indicator appear to be suspended in the air; here we must specify that in all the figures where the tanks are presented, the images are actually longitudinal sections. In fact, the ballast tanks are closed compartments that have some "mouths" for access and visitation.

![Figure 8. Equivalent Stresses.](image)
5. Conclusions
In this paper we presented a study based on the influence of the air-water ballast pressure on the metallic structure of the ballast tanks. The ballast tank models presented here are identical in size, the first tank has a large free surface, while the second tank has been optimized to reduce the sloshing effect.

The sloshing effect is produced by moving of ballast tanks with a velocity \( v = 3 \) meters/second on the Ox direction, which corresponds to the pitching motion of the ship. This motion corresponding to a velocity of 5.825 knots will make the water contained in the ballast tanks remain inertial resulting in an imbalance in the fluid mass and will form an internal wave which itself is the effect of sloshing.

In conclusion, the effect of sloshing has a great influence when the ship is in the underway because the cargo tanks are empty, and the ballast tanks are fully loaded. For various reasons some ballast tanks may be partially filled (ex. the sides tanks that provide ship’ trim), and the sloshing effect that will occur may influence the stability of the ship. In this situation it is very important how these tanks were built. In this tanks, the mechanical stresses induced are cyclic (may have maximum and minimum values, repeated) and can produce the fatigue stresses that is located in the welded joints of the ship' hull. In this situation the quality of the welded joints plays a decisive role.

6. References
[1] Sederstrom, Donn R 2016 Methods and Implementation of Fluid-Structure Interaction Modeling into an Industry-Accepted Design Tool, Electronic Theses and Dissertations, https://digitalcommons.du.edu/etd/1197.
[2] Souli, M and Benson, D. J 2010 Arbitrary Lagrangian-Eulerian and Fluid-Structure Interaction: Numerical Simulation, (USA: John Wiley & Sons, Inc.), 1 edition, 320 pages, ISBN-13 978-1848211315.
[3] Sivý M., Musil M., Chlebo O. and Havelka R. 2017 Sloshing effects in tanks containing liquid MATEC Web of Conferences 107 00069 DYN-WIND'2017 doi: 10.1051/matecconf/2017107000 Bratislava Slovakia.
[4] Krata P 2013 The Impact of Sloshing Liquids on Ship Stability for Various Dimensions of Partly Filled Tanks, TransNav, The International Journal on Marine Navigation and Safety of Sea Transportation 7(4) 481-489 doi: 10.12716/1001.07.04.02 Gdynia Maritime University (Gdynia, Poland).
[5] Zeineb S., Chokri M., Zouhaier H. and Khifa M. 2010 Standing wave induced by free liquid sloshing in rectangular tank Proceedings of the International Renewable Energy Congress (Sousse, Tunisia) doi: 10.2478/jok-2013-0005
[6] Akyildiz H. and Unal E. 2005 Experimental investigation of pressure distribution on a rectangular tank due to the liquid sloshing Ocean Engineering 32 www.sciencedirect.com
[7] Raouf A. Ibrahim 2005 Liquid Sloshing Dynamics, Theory and Applications, Cambridge University Press, 1 edition, ISBN –I3-978-O-521-83885-6
[8] Dumitrache C L, Deleanu D 2019 Sloshing effect, design and optimisation of water ballast tank, Sea-Conf 2019 The 5th International Scientific Conference, “Mircea cel Batran” Naval Academy, Constanta, Romania, IOP Conf. Series: Journal of Physics Conference Series 1297, doi: 10.1088/1742-6596/1297/1/012003