AN INVESTIGATION INTO THE LIMITATIONS OF THE PANEL METHOD AND THE GAP EFFECT FOR A FIXED AND A FLOATING STRUCTURE SUBJECT TO WAVES

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ABSTRACT
The wave-induced motions of vessels moored next to a fixed object and open to the sea impact the operability of many offshore operations, and should be assessed in order to avoid accidents and catastrophes. When analysing vessels moored by a fixed object (e.g. quay-side or platform), time domain simulations have shown numeric instabilities resulting in unreliable outcomes. The origin of the numerical instability might lie in the hydrodynamic added mass and wave radiation damping. This is typically calculated using potential flow methods and influenced by the existence of standing waves in the gap between the two bodies. For certain frequencies, these give negative values, potentially causing instabilities in non-linear (coupled) time domain simulations. In these cases, the vessel can behave unexpectedly, generating energy rather than dissipating it. As such, certain simulations have been disregarded as they are unlikely to accurately represent real-life scenarios.

This paper investigates and compares added mass and damping using two different tools and studies the gap effect when conducting diffraction analysis using 3D panel methods. The work covers a literature study into potential theory, multibody analysis, Computational Fluid Dynamics (CFD) and lid techniques.

This is followed by a study conducted using both panel method and CFD analyses. The results from both approaches have been compared, showing interesting information and the necessity of researching more into the problem addressed in this paper.

KEYWORDS
Hydrodynamics, Standing Waves, Potential Theory, Multibody, Lid, Added Mass, Damping, Shallow Water, Quay-side.

INTRODUCTION
The motions of a vessel are challenging to assess when a ship is in shallow water in the vicinity of a body such as a quay wall. It has been seen that time domain simulations of this scenario can result in extraordinary and unexpected behaviour including large amplitudes and erratic motions [18].

Two approaches to calculate motions of a vessel are most common:

- Time-domain simulations have been used successfully to calculate the behaviour of moored ships in irregular waves. Time domain methods allow for non-linearity and coupling between components.
- Linear frequency-domain simulations are generally used to assess first order motions and mooring responses where the assumption of linear relationships is acceptable.

Fatilsen, Lee and Newman [7] contributed to the development of the Panel Method. According to this method, the submerged part of the vessel is divided into a finite number of panels. Potential flow theory then calculates hydrodynamic pressures on the hull. This 3D diffraction calculation derives inter alia added mass and added damping.

Discrepancies and unrealistic hydrodynamic data when using methods like the Panel Method have been seen for certain cases [18], [8]. The discrepancy between the linear predictions using potential theory and experimental data might be due to flow viscosity during the simulations or the interaction effect [8]. Researchers like Sutulo and Soares and Pinkster and Bhawsinka developed Potential Flow (PF) solvers to predict the interaction effects
as an alternative to the excessive work and high cost involved in developing a coefficient based model.

The resonance frequency of the trapped gap water is also important as it may influence the stability of the bodies. In this context, accurate prediction of the water elevation in the gap and the resonance frequency is crucial in evaluating wave added mass and radiation damping for the structure and its stability. However, it is well known that while the potential theory is capable of predicting the resonance frequency, it over-predicts the resonant wave height in the gap near the resonant frequency.

In order to fix the problem, Newman mathematically modelled a rigid damping lid on the gap surface and used the generalised mode technique to compute the lid motions [23], [24] and implemented it into WAMIT software.

A flexible lid was modelled by Chen [19]. It introduced a damping force term into the free surface boundary conditions, which was explained as energy dissipation.

Other experiments like those conducted by Pauw et al. [21] showed that there is no a priori determination of the damping coefficient unless calibrated by experimental tests. Others like Kristiansen and Faltinsen [22] applied a vortex tracking method to study the piston mode in a vertical gap and concluded saying that flow separation mainly accounted for the discrepancy between linear results and measurements, and nonlinear free surface conditions are of minor importance.

In terms of ship to ship interaction, several pieces of research have been carried out on the prediction of motions in waves. Ohkusu [26] analyzed the motions of a ship in the neighborhood of a large moored two-dimensional floating structure by strip theory. Kodan [27] extended Ohkusu’s theory to hydrodynamic interaction problem between two parallel structures in oblique waves. Van Oortmerssen [28] and Loken [29] used the three-dimensional linear diffraction theory to solve this problem. Fang and Kim [31] analyzed the motions of two longitudinally parallel barges by strip method. Fang and Chen [30] used three-dimensional source distribution method to predict the wave exciting forces, relative motion, wave elevation and drift force between two bodies in waves [17].

However, when assessing the motions of quay-side ships, not much can be found. Van Oortmerssen [8] was one of the first to state that a ship floating freely in waves in the proximity of a quayside may experience resonant sway and heave motion at multiple frequencies. F.A. Kilner [12] studied model tests on the motions of moored ships placed on long waves. Zero damping was assumed and justified for many cases.

Two main flow pattern factors affect the behaviour of a moored vessel in port and need particular consideration. Firstly, the gap between the vessel and the quayside and, secondly, the water depth / clearance below the hull of the vessel. Both of these parameters affect the water particle motions close to the vessel and will affect the outcome of any calculations.

Added mass and damping are direct results of the geometric characteristics of the vessel and the excitation frequency (i.e. wave period). The accuracy of added mass and damping coefficients directly influences the numerical simulation of vessel behaviour. The value of the added mass can be positive or negative, depending on the boundary conditions of the object [2].

The concept of added mass and first approach was introduced by Dubua in 1776 [5], who conducted experiments with pendulums in fluid. Other researchers such as Stokes and Green also contributed towards the research of added mass [5]. The evaluation of added mass and damping coefficients of an oscillating cylinder was conducted by M.Rahman and D. D. Battha [10]. Numerical and experimental results were presented showing good agreement between the two. Research into the hydrodynamic characteristics of an FPSO were conducted by Wang et.al [11]. They investigated and presented values for added mass and radiation damping.

The sign of added mass coefficients for 2-D structures was studied by M.McIver and P.McIver [5]. However, no conclusion was drawn regarding the negative values of added mass.

Many predictions have been made when calculating added mass and damping for floating bodies [20]. A research project within Heerema Marine Contractors and the MIT was conducted to determine added mass and damping coefficients of suction piles using CFD. The results were compared with model tests presented at the OMAE [20].

The issue, as detailed in a previous paper [18], originates from simulating a moored vessel in the vicinity of a straight wall (quayside) using nonlinear time domain software. Upon initial analysis, negative added mass and damping values were observed. This was not just in the cross terms of the coefficient matrix from the diffraction analyses, but also in the main diagonal for certain frequencies. When using these added mass and damping values in time domain simulations, they led to unstable calculations. As the balance was disturbed by the coupling effect of the mooring lines, the vessel seemed to generate energy and showed ever increasing motions with no sign of stabilising. It was concluded that these simulations were not representative of real-life scenarios and an alternative approach is needed to realistically represent a moored vessel alongside a fixed solid object. A robust working solution was found deriving realistic vessel motions in time domain simulations.

This paper was motivated by the fact that vessel hydrodynamic characteristics are often used as an input when running coupled time domain analyses. Wave
excitation forces, added mass, damping and restoring forces are normally imported into the numerical model. After that, the motion behaviour of the vessel is calculated based on these parameters. When no mooring lines are attached, the results are similar to calculations based on RAOs derived from diffraction. However, when simulating a vessel moored to a quayside, the coupling between the mooring forces and vessel motions needs to be considered. For certain frequencies, negative added mass and damping values were found when conducting diffraction analyses using potential theory. This fact didn’t present a problem when a spectral analysis was conducted using the data from the motions, however when conducting a dynamic analysis, the simulations became unstable. This led us to investigate the gap effect in greater depth as in the offshore industry a multibody analysis is commonly used in order to solve real problems such as lifting operations, or multi-body mooring analyses.

METHODOLOGY
The methodology of this study is summarised below:
- Modelling of a 2,500TEU container vessel for the purposes of the study
- Conducting diffraction analyses using a panel-based linear frequency-domain diffraction model. Five cases with varying boundary conditions were considered.
- CFD Analyses using the URANS methods.
- Comparison of the results for the hydrodynamic coefficients.

DIFFRACTION ANALYSES
3D diffraction calculations were performed to assess the wave motions on a vessel.
When the wave excitation forces, added mass, damping and restoring forces are known, the vessel motions follow from differential equations. They are generally presented in the form of Response Amplitude Operators (RAOs). These are used for spectral analysis of the behaviour of the vessel in waves.

The software used for this study was ANSYS-AQWA [1].

Main Formulation
Fluid is assumed to be ideal and not rotational. This allows potential theory to be used. The other major assumption is that the incident wave acting on the body is of small amplitude when compared to its length (i.e. small slope). This theory may be used to calculate the wave excitation on both fixed and floating bodies.
The potential theory used in 3D diffraction software is first order. Superposition may be used to formulate the velocity potential within the fluid domain.
This potential ‘φ’ consists of the contributions from radiation waves due to the six modes of body motion, the incident wave field and the diffracted or scattered wave field.
The two resulting problems are:
- The problem of a floating body undergoing harmonic oscillations in still water. These forces are given in terms of added mass and wave damping coefficients.
- The problem of a fixed body being subjected to a regular incident wave train. These forces are broken down into two components, Froude-Krylov and wave diffraction force components.

Following the previous assumptions, the total potential due to unit amplitude incident wave, diffraction and radiation waves may be written as:

\[ \Phi(X,Y,Z) e^{-i\omega t} = \left( \Phi_I + \Phi_d \right) \sum_{j=1}^{6} \Phi_j x_j e^{-i\omega t} \]

Where
\[ \Phi_I = \text{incident wave potential} \]
\[ \Phi_d = \text{diffracted wave potential} \]
\[ \Phi_j = \text{potential due to jth motion} \]
\[ x_j = \text{th-motion (per unit wave amplitude)} \]

Diffraction Analysis Output
3D diffraction calculations result in the following frequency dependant parameters:

- Motion Response Amplitude Operators (RAOs) at the centre of gravity - the responses in different locations are found by translating the RAO to the correct location. Rotational motions remain identical as the vessel is considered a rigid body.
- Quadratic Transfer Functions - as the convenient way to express the second order forces in the frequency domain in terms of force spectra (Pinkster, 1980). However, these are not considered in this paper.
- Added Mass Coefficients - Defined as the imaginary part of the potential due to the jth motion:

\[ A_{ji} = \frac{\rho}{\omega} \int_S \Phi_I^{im} n_j dS \]

- Radiation Damping Coefficients - Defined as the real part of the potential due to the jth motion:

\[ B_{ji} = \rho \int_S \Phi_I^{re} n_j dS \]
Active Excitation Forces - Considered as Froude Krylov and diffraction forces along the wetted surface of the body:

\[ F_j = - \int \omega \rho \Phi \, dS - \int \omega \rho \Phi_n \, dS \]

**Method for Suppression of Standing Waves**

Fictitious geometry lid elements on the free-surface between the bodies have been added to solve the problem of standing waves during diffraction calculations with a boundary integration approach.

This "lid method" is based on the implementation of a damping force at the meshed free surface in between the two bodies. This method is inherent in AQWA and based on the equations presented by Chen (2004).

Free surface boundary conditions in a conventional process are described by:

\[ \frac{\partial \Phi}{\partial z} - \frac{\omega^2}{g} \Phi = 0 \]

When adding a lid, the equation turns to:

\[ \frac{\partial \Phi}{\partial z} - (1 - i \varepsilon) \frac{\omega^2}{g} \Phi = 0 \]

In which \( \omega \) is the wave frequency and the non-dimensional parameter \( \varepsilon \) is related to the damping parameter.

**Chosen Lid**

For certain cases, a flexible virtual "lid" is added to the area between the wall and vessel. This is due to the proximity of the wall to the vessel and the generation of standing waves between the two. This can cause numerical instabilities in the software. The lid includes a damping factor set at different values depending on the considered case. An internal lid was also added in order to avoid irregular frequencies. The external "lid" is shown below.

![Figure 1: External Lid.](image)

**Viscous Damping in Roll**

Motion damping is normally caused by viscous effects such as skin friction and eddies or vortices as well as the dissipated energy from the moving structure. The Panel Method only accounts for damping generated by the radiated wave. This approximation might be accepted for pitch, surge, sway, heave and yaw. However, often little radiation damping is calculated around the roll natural period. Without the viscous effect, overestimates of roll motion prevail [4].

The Watanabe-Inoue-Takahashi [3] formula was utilised for predicting the roll damping of the hull at the design wave limit. This formula was developed on the basis of an extensive series of model tests and some theoretical considerations.

This method predicts the damping ratio as a function of the roll amplitude. In the first instance, diffraction analysis was conducted with no added damping. After that, the Watanabe-Inoue-Takahashi formula was then used to determine the total damping coefficient.

The required damping was added to the radiation component from the first diffraction analysis. This achieved the damping ratio as calculated from the formula and, as a result, 5% of the critical damping was added.

**Cases**

The first round looks forward to study the impact of adding a lid at the free surface. The following cases were considered:

1. Quay-Vessel
2. Quay-Vessel with Internal Lid
3. Quay-Vessel with Internal Lid and External Lid Damping of 0
4. Quay-Vessel with an Internal Lid and External Lid Damping of 0
5. Quay-Vessel with an Internal Lid and External Lid Damping 0.01
6. Quay-Vessel with an Internal Lid and External Lid Damping 0.02
7. Quay-Vessel with an Internal Lid and External Lid Damping 0.1
8. Quay-Vessel with an Internal Lid and External Lid Damping 0.3
9. Quay-Vessel with an Internal Lid and External Lid Damping 1

Different gap distances of 3m, 6m, 12m, 18m, 36m and 54m were tested for the described cases 1-9.
The Model

The model created included the body of the underwater part of the vessel and a representation of the quay-side as diffracting elements to be analysed using the panel method. The picture below shows a screenshot of the model:

Figure 2 Hydrodynamic Model

The proposed vessel is a 2,500 TEU container with the following characteristics:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length (m)</td>
<td>200</td>
</tr>
<tr>
<td>Width (m)</td>
<td>30</td>
</tr>
<tr>
<td>Draft (loaded) (m)</td>
<td>11</td>
</tr>
<tr>
<td>Draft (ballast) (m)</td>
<td>5</td>
</tr>
<tr>
<td>Displacement (loaded) (t)</td>
<td>50,000</td>
</tr>
<tr>
<td>Displacement (ballast) (t)</td>
<td>20,000</td>
</tr>
</tbody>
</table>

Table 1 Vessel Properties

The software calculates the hydrodynamic properties for a range of wave periods and a range of directions. The wave periods selected ranged from 0.05 rad/s (125s) to 1.6 rad/s (3.5 sec) in steps of 0.05 rad/s. The wave directions ranged from head seas to beam seas in steps of 22.5 degrees.

COMPUTATIONAL FLUID DYNAMICS ANALYSIS

This section of the paper presents the technique used for the computation of the added mass and damping hydrodynamic coefficients for the container ship in still water.

Substantial progress has been made during the last few years in dealing with the nonlinear response of a rigid ship travelling in regular waves, using potential flow analysis and Reynolds Averaged Navier-Stokes (RANS) methods [1].

The computations were carried out using the STAR-CCM+ software, using the inviscid flow option, and the 3-D model already built for potential fluid software. RANS equations form the mathematical background of the methodology used for the calculations. When using inviscid flow, the equations turns to the well-known Euler equations. In this software, a finite volume method is used to discretise the fluid domain with control volumes (CVs) corresponding to the size of computational cells. The integral form of conservation equations is discretised and applied to each cell centre. Each equation is a function of pressure and velocity at the cell centre and in the neighbouring cells. For non-linear equations, iterative techniques are applied for linearization. The mid-point method is used to compute the space integrals, implying that the surface integrals are a product of the integrand at the cell face centre and the area of the face and the volume integrals are the product of mean integrand value and the CV. Both are 2nd order accurate if the integrand is also calculated to 2nd order accuracy.

The issue of conducting CFD simulations is to compare results with the ones coming from diffraction. Forced oscillation tests have been simulated by imposing a harmonically periodic motion to the center of gravity of the structure in still water. This is explained below.

Force Oscillation Tests

The relation between the potential coefficients and the frequency of oscillation is found using forced oscillation tests. During the forced horizontal oscillation, the motion of the model is defined by:

\[ y(t) = y_a \sin(\omega t) \]

And therefore the forces are given by:

\[ F_y(t) = F_a \sin(\omega t + \varepsilon F_y) \]

On the other side the equation of motion is described by:

\[(m + a)\ddot{y} + b\dot{y} + cy = F_a(\omega t + \varepsilon F_y) \]

The component of the exciting force in phase with the sway motion is associated with inertia and stiffness, while the out-of-phase component is associated with damping.

And as

\[ y = y_a \sin(\omega t) \]
\[ \dot{y} = y_a \omega \cos(\omega t) \]
\[ \ddot{y} = -y_a \omega^2 \sin(\omega t) \]

And finally

\[(M + M_{yy})-y_a \omega^2 \sin(\omega t) + b y_a \omega \cos(\omega t) + cy_a \sin(\omega t) = F_a \cos(\varepsilon F_y) \sin(\omega t) + F_a \sin(\varepsilon F_y) \cos(\omega t) \]

When in phase

\[ a = \frac{F_a \cos(\varepsilon F_y)}{y_a} - m \]
When out of phase:

\[ b = \frac{F_0 \sin(\epsilon F_y)}{\omega} \]

Not relevant for this case:

\[ c = \rho g A_\omega \]

And the in-phase and out-phase Forces can be obtained:

\[ F_a \sin(\epsilon F_y) = \frac{2}{t \omega^2} \int_{t-T/2}^{t+T/2} F(t) \cos \omega t \, dt \]

\[ F_a \cos(\epsilon F_y) = -\frac{2}{t \omega} \int_{t-T/2}^{t+T/2} F(t) \sin \omega t \, dt \]

**Model Set Up**

The same vessel as the one used during the diffraction calculations was imported. A vertical wall was built allowing a gap of 3m between both structures and keeping a water depth of 16m.

Both water and air were modeled using the volume of fluid approach. The simulations were configured such that the High-Resolution Interface Capturing (HRIC) method. In some local areas, with small cells, this transitioned to a second-order free surface capturing scheme to maintain numerical stability. The tank used for the simulations is shown below:

The rectangular main domain extended three lengths aft and forward and sufficiently far above, fitting a 16m water depth matching the previous simulations. Blocks that allow a mesh refinement were defined for the ship and the wall.

The vertical boundary portside of the vessel was treated as a velocity inlet (See figure above). All other outer boundaries were considered pressure outlets with zero absolute velocity, except that free-slip side walls were used in the steady drift simulations.

The computational meshes were unstructured and predominantly composed of hexahedral cells with localized refinements.

The mesh included refinement of free surface in both sides of the ship making sure the whole gap was included to accurately capture the air/water interface. Prism layer meshers, Surface re-mesher and Trimer were used for all three mesh domains. Typical Meshes of 2m, 1m, 3m were tested for the ship domain. The final Mesh for the simulations was 5m (with five prism layers) for the tank and 2m (with two prism layers) for the ship, as demonstrated in the figures below. The number of elements was:

Tank: 1949282 cells, 5876591 faces  
Wall: 17263280 cells, 51179020 faces  
Subtract (Ship): 651328 cells, 1911707 faces

Pictures are shown below:
All of the simulations were solved using an implicit unsteady approach. In unsteady simulations, the time step was typically chosen to be $T/2880$, where $T$ is the motion period. Five iterations were used per time step. The unsteady motion then initiated with $t=T/4$ to avoid instantaneous changes in the sway rate.

In order to ensure a significant reduction of the equation residuals and a good resolution of the strong non-linearites, at least 5 outer-loop iterations were used.

The CFD model was validated and optimized against known parameters like the buoyancy of the ship.

**Cases**

A wave heading of 90 degrees was set up during all simulations as the parameters to be studied are in sway. The following, presenting abrupt changes in sign during diffraction analysis, were considered during the computational fluid dynamic simulations:

<table>
<thead>
<tr>
<th>Case</th>
<th>Wave Frequency (rad/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>1</td>
</tr>
<tr>
<td>Case 2</td>
<td>0.8</td>
</tr>
<tr>
<td>Case 3</td>
<td>0.6</td>
</tr>
</tbody>
</table>

Table 2 Studied Frequencies

**RESULTS**

**THE GAP EFFECT USING DIFFRACTION ANALYSIS.**

**First Round Simulations.**

As described in previous sections, a first round of simulations was conducted covering the scenarios:

- The vessel placed next to the quay with no lids covering the gap.
- The vessel next to the quay with Internal Lids in the vessel
- The vessel is placed next to the quay with an external Lid with a 0 damping factor.

This first round showed that:

- Irregular frequencies are removed when an internal lid is added.
- No significant differences are seen when there is no lid or the damping factor of the lid is 0. However, the existence of a lid, even if the damping factor is 0, slightly modifies the results for added mass and damping.

**Second Round Simulations**

**Wave Frequency Dependant Added Mass for different Gap Distances.**

The results from the analysis for the added mass of the ship for the different gap separations and damping coefficients for the lid.

The graphs show that:

- The more damped the system, the added mass tends to decrease when positive and increase when negative.
- A change in sign is produced at the same frequency for each case (Gap separation variation case) and at a smaller frequency when the gap increases.
- An abrupt change in sign is seen at 0.85, 0.75 and 0.5 rad/s for 3m, 6m and 12m Gap respectively.
- Increasing the distance seems to increasingly affect the damping coefficient assigned to the flexible lid. The system becomes sensitive to that coefficient at big gap distances.
- The system seems to be overdamped for a damping factor of 1 and 0.3 for big gap distances.
- The added mass of the ship takes values bigger than the real mass of the vessel.
- From the graphs it can be seen that the addition of lids is necessary when the gap distance takes significant values.
ADDED MASS VARIATION in SWAY 3m GAP

ADDED MASS VARIATION in SWAY 6m GAP

ADDED MASS VARIATION in SWAY 12m GAP

ADDED MASS VARIATION in SWAY 18m GAP

ADDED MASS VARIATION in SWAY 36m GAP

ADDED MASS VARIATION in SWAY (18m GAP)

ADDED MASS VARIATION in SWAY (36m GAP)
Wave Frequency Dependant Damping for different Gap Distances.

The results from the analysis for the radiation damping coming from the diffraction analyses for different gap distances between the two bodies are shown below.

The graphs show that:
- The system is sensitive to the damping coefficient assigned.
- A damping factor of 1 seems to overdamp the system.
- Damping does not decrease when the damping factor increases for certain cases (See 36m Gap)
- When added mass changes abrupt in sign, damping presents a peak.
- Unexpected behaviour is seen for the 36m Gap Case. Added mass presents negative values, damping presents a peak. However, damping turns bigger for the case in which the damping coefficient for the lid is 0.01 than when there are no lids.
Sway RAOs

The RAOs from diffraction analysis for the different separations between the two bodies are shown below.

The graphs show that:

- The results for the motions of the vessel do not seem to be impacted significantly by the existence of lids.
- An extreme variation in the damping coefficient for the flexible Lid does not seem to impact the motions much in Sway.
- Even when the system is overdamped, the motions of the vessel don’t differ much from the trend.
- When added mass and damping change dramatically, the motions seem to behave normally.
ADDED MASS AND DAMPING USING CFD FOR A 3M GAP.

Frequencies where an abrupt change in added mass and damping has been seen have been selected to be studied using CFD harmonic motion testing. The results below show a comparison at the selected frequencies for the case in which no lids are included during 3D Diffraction analysis and the CFD case explained in previous sections. The results are presented in the graphs below:

It can be noted that the graphs show clear discrepancies between the two methodologies.
DISCUSSION

This paper has researched the Gap Effect when the distance between a fixed body and a floating one is varied and when flexible lids are used to mitigate standing waves in the gap between the two bodies using the panel method.

This paper has shown the effect of the addition of flexible lids in between the two bodies and in shallow water.

Diffraction analysis is widely accepted in industry when assessing the motions of a floating body.

The diffraction analyses conducted for this paper indicates that:

1. When a multibody analysis is conducted, assessing the motions becomes a complicated and challenging problem.
2. The addition of lids is necessary when the gap reaches significant dimensions. For this particular case, when it reached a separation of 18m or more.
3. When the gap is not considerable, the addition of lids does not significantly impact the frequency dependant added mass.
4. A damping factor of 1 seems to overdamp the system.
5. An extreme variation in the damping coefficient for the flexible lid does not seem to impact the motions significantly in Sway.
6. If a coupled analysis is required, it is very important to include flexible lids during diffraction and assign the correct damping factor to them. Dynamic simulations require frequency dependant damping and added mass as main inputs. Inaccurate added mass and damping can lead to unreliable results from the simulations [18].
7. Standing waves are seen in the gap when no lids are included. Those standing waves are damped when lids are added and when the damping factor increases.
8. The shallow water effect, the interaction effect and the existence of standing waves may explain the extreme value unexpected results.
9. Towing tank tests are recommended to verify the results.
10. This particular case may demonstrate a limitation of potential theory and panel method.

From the CFD Analysis it is noted that:

1. The calibration and validation of CFD computations is more time consuming when compared to diffraction analyses.
2. Viscous effects have been accounted for during the harmonic motion test.
3. The results differ from those coming from diffraction analyses.
4. The differences in damping seen in the comparison between the Panel Method and CFD might come from the existence of standing waves in the gap during the diffraction calculations. Large waves in the gap can over-damp the system, possibly explaining the damping discrepancies.
5. The differences in added mass may also originate from the existence of standing waves. Large standing waves generate a velocity stroke, helping the fluid moving the vessel.

Currently there is no theoretical way to know how far the results from diffraction analysis are from reality. Also there is no way to know if CFD simulations are nearer the values that would be obtained conducting model tests.

It is worth mentioning that for critical operations between two bodies (for example, multi-body heavy lift activities) it is very important to know how the two bodies behave. Errors in the dynamic simulations may result in accidents during the real-world operation. This study invites further research into the proposed multibody problem as the interaction effect is increasingly required to solve real industry problems.

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