ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

NUMERICAL SIMULATION OF UNCOUPLED HEAVE MOTION OF A MONOCOLUMN PLATFORM

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Abstract. *The monocolumn platform is an offshore platform concept that presents some advantages compared to the conventional concepts of semi-submersible and FPSO units. Particularly, the good seakeeping performance and the high storage capacity are the most attractive characteristics. However, monocolumn platforms usually show an undesirable response in heave motion. In recent years, many research projects have been carried out about the problem suggesting that the installation of an external skirt has a strong influence on damping the amplitude of the heave motion. In an attempt to develop a hydrodynamic driven design of monocolumn platforms, major oil companies have carried out experimental tests with monocolumn hulls provided with an external skirt. In order to obtain a better comprehension of the dynamics of the damping process, a numerical model for the simulation of uncoupled heave motion of a monocolumn platform was developed. In this paper, the simulation results obtained for a heave decay test of a monocolumn platform were presented and discussed. The numerical model adopted for the simulations was developed in full scale with an experimental model. The results obtained were validated against experimental data.*

Keywords*: offshore, monocolumn, damping, CFD.*

1. INTRODUCTION

The monocolumn platform is an offshore platform concept that presents some advantages compared to the conventional concepts of semi-submersible and FPSO units. Particularly, the good seakeeping performance and the high storage capacity are the most attractive characteristics. The main goal of the MonoBR project is to develop an efficient design for monocolumn platforms, based on the concept of a cone-cylinder composition of external and internal walls associated to a double bottom skirt. (Reyes *et al.*, 2009). However, monocolumn platforms usually show an undesirable response in heave motion.

Along the years, many research projects have been carried out about the problem of the undesirable response in heave motion (Aalbers, 1984; Matsuura, 1995), suggesting that the moonpool geometry has an important effect on damping the amplitude of the heave motion. Although some studies demonstrate that the geometry of the moonpool deeply affects the dynamic behavior of the platform (Torres *et al.*, 2007), it was observed that the flow inside the moonpool presents a very complex behavior, characterized by the onset of complex vortex structures (Torres *et al.*, 2008). An alternative solution to control the heave motion of monocolumn type platforms is to install external skirts near the bottom of the hull (Nishimoto, K. *et al.*, 2001). Torres *et al.* (2004) showed that it seems like the viscous effects around an external skirt located at the bottom of hull play an important role on the heave motion damping process.

In an attempt to develop a hydrodynamic driven design of monocolumn platforms, major oil companies have carried out experimental tests with monocolumn hulls provided with an external skirt. In order to obtain a better comprehension of the dynamics of the damping process, a numerical model for the simulation of uncoupled heave motion of a monocolumn platform was developed. In this paper, the simulation results obtained for a heave decay test of a monocolumn platform were presented and discussed. The numerical model adopted for the simulations was developed in full scale with an experimental model. The results obtained were validated against experimental data.

2. REFERENCE MOCOLUMN PLATFORM

The present work was developed using as reference the FPSO-MONOBR platform. The FPSO-MONOBR concept was originally developed to have low amplitude motions and to provide a centralized access to conventional steel catenary risers (Figure 1). The platform structure consists of a monocolumn configuration of a cone-cylinder composition of external and internal walls associated to a double bottom skirt. The hex-decagonal faceting is an acceptable compromise between the assurance of practically circular waterlines and constructive feasibility. Table 1 shows the main particulars of the original design.

ISSN 2176-5480

Table 1 – Main particulars FPSO-MONOBR.

Max. Outer Diameter (at bottom skirt)	118.85 m
Min. Inner Diameter	38.45 m
Hull Depth	58.00 m
Operating Draft	39.50 m
Survival Draft	28,70 m

EXPERIMENTAL HEAVE MOTION TEST

The purpose of the model tests conducted with the FPSO-MONOBR was to calibrate and validate the numerical software that was used for the full scale design computations. The model tests were conducted with a reduced scale model (1:75) that was built in aluminum and foam material (Divinycell). The platform model has an external diameter equal to 1.329 meters with a moonpool diameter of 0.513 meters. The operational draft at model scale equals to 0.527 meters. In the original configuration, the model has a skirt with an external diameter of 1.585 meters installed at the bottom of its moonpool.

Decay tests are performed in order to document natural periods and damping characteristics of the tested systems. In particular, for the heave decay test of an offshore platform, the moored model with the risers connected was pulled a small prescribed vertical distance and then released. The FPSO-MONOBR was designed to operate at a 525 m depth site and to be equipped with a spread-type mooring system, consisting of 13 mooring lines. However, due to laboratory depth limitations combined with the scale requirement, a truncated mooring was modeled in the basin. Although model tests with truncated mooring system usually give reliable frequency motion response, the vertical excursion of the truncated system may deviate from the full depth due to the difference in modeling segments to obtain the optimal horizontal excursion characteristics. In most cases, it is expected that this difference will be insignificant since the vertical stiffness from the buoy water plane area is the dominating factor.

In case of a pure free heaving body in still water, the linear equation of the motion of the center of gravity, *CG*, of the body is given by

$$(m+a)\ddot{z}+b\dot{z}+cz=0\,, \tag{1}$$

where, m represents the mass of the body, a the hydrodynamic mass coefficient, b the hydrodynamic damping coefficient, c the restoring spring coefficient and z the vertical displacement.

Equation (1) can be rewritten as

$$\ddot{z}+2\nu\dot{z}+\omega_0^2 z=0\,, \tag{2}$$

in which the damping coefficient, ν, and the undamped natural frequency, ω_0, are defined by

$$2\nu=\frac{b}{m+a}\,. \tag{3}$$

A non-dimensional damping coefficient, κ, is written as:

$$\kappa=\frac{\nu}{\omega_0}\,. \tag{4}$$

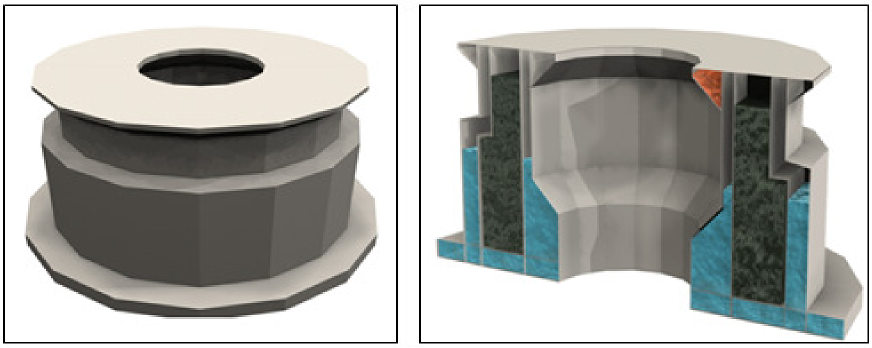

Figure 1. The original FPSO-MONOBR concept.

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After some mathematics, the solution of the Equation (1) of the heave decay motion becomes

$$z = z_a e^{-\nu t}\left(\cos\omega_z t + \frac{\nu}{\omega_z}\sin\omega_z t\right). \tag{5}$$

The logarithmic decrement of the motion is directly defined as

$$\nu T_z = \ln\left(\frac{z(t)}{z(t+T_z)}\right). \tag{6}$$

Considering that $\omega_z^2 = \omega_0^2 - \nu^2$ for the natural frequency oscillation and also the damping is not of significance, it is possible to neglect ν^2. This leads to $\omega_z \approx \omega_0$, and, consequently, $\omega_0 T_z \approx \omega_0 T_z = 2\pi$. The non-dimensional damping coefficient is now given by

$$\kappa = \frac{1}{2\pi}\ln\left(\frac{z(t)}{z(t+T_z)}\right). \tag{7}$$

Figure 2 shows the normalized heave decaying curve obtained for the FPSO-MONOBR model. It can be observed from Figure 2 that the non-damping coefficient of the system, κ_{exp}, equals to

$$\kappa_{\exp} = 0.036\ ,$$

since $z(t)$ = -0.00893 m, when t = 1.4 s, and $z(t+T_z)$ = -0.00712 m, when t =4.2 s (for T_z = 2.8 s).

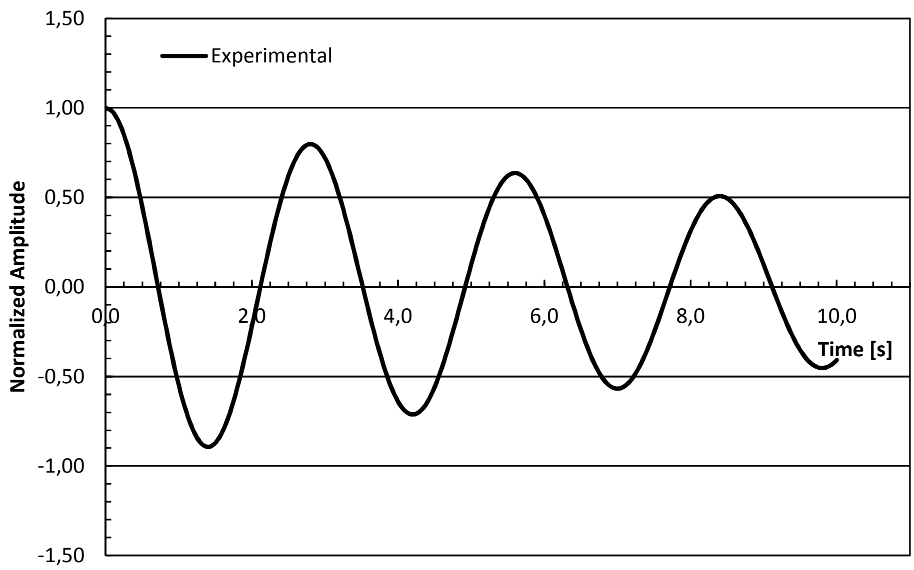

Figure 2. Experimental heave decaying curve obtained for the FPSO-MONOBR model.

3. NUMERICAL MODEL

Computations were conducted on a 2D domain taking the advantage of the axial symmetry of the monocolumn structure. A slice type geometry was developed for the domain with an angle of rotation about the moonpool vertical axis of 1.0 degrees. A cartesian (x, y, z) coordinate system was used, with the Z coordinate taken as the vertical direction vertical and the bottom of the domain representing the x = 0 cm coordinate (Fig. 3). The numerical model was developed in full scale with the experimental model, having geometric similarity with it.

ISSN 2176-5480

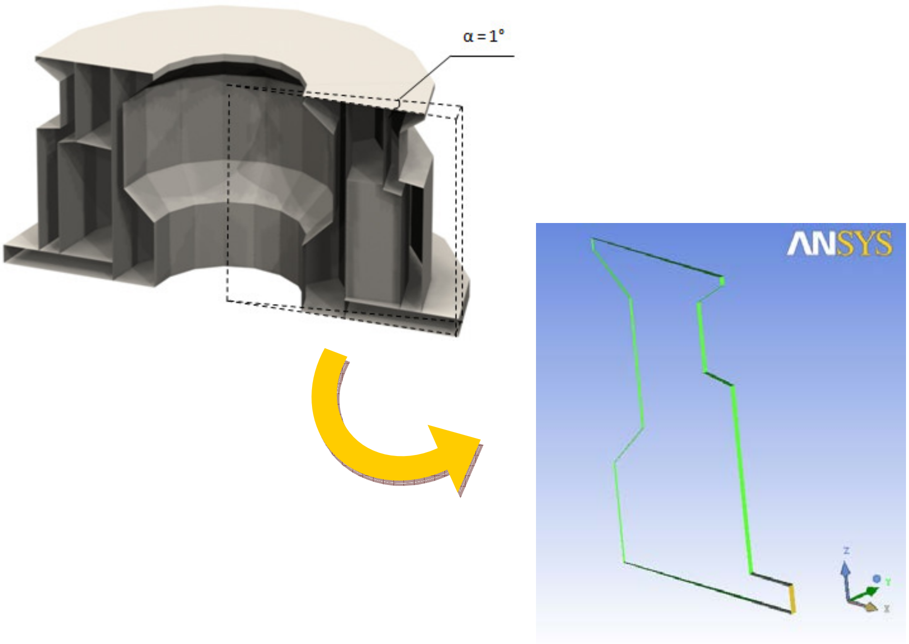

Figure 3. 2D numerical domain, slice type geometry.

The domain was discretized by a structured mesh consisting of hexahedrons. Extensive domain independence tests were performed showing a great influence of the domain configuration in the performance of the numerical model. In order to reduce the computational effort, it was tried to adopt a reduced domain configuration by means of using truncated values for the domain extension in relation to the actual figures observed in the experimental test. Figure 4 shows the dynamic pressure field calculated for a reduced domain configuration with an extension equal to 40% of the actual value. It can be seen an expressive deformation of the dynamic pressure field in the nearby of the farfield, even though the domain extension is more than 15 times the model diameter. Once observed the significant influence of the domain configuration on the simulation results , the dimensions of the experimental basin (50 m extension, 10 m depth) have been fully adopted for the configuration of the domain.

Extensive grid-independence tests were also performed resulting in a final non-uniform, body-fitted mesh with approximately 1,000k hexahedral elements. The mesh was particularly refined in the near wall region so as to completely resolve the inner turbulent and viscous sub-layers. A fine grid mesh was also adopted around the skirt in order to capture the complex vortex formation process that takes place around it (Figure 5). As a result of the grid-independence test, a fine grid with y^+ varying from 0.5 to 2.0 was obtained.

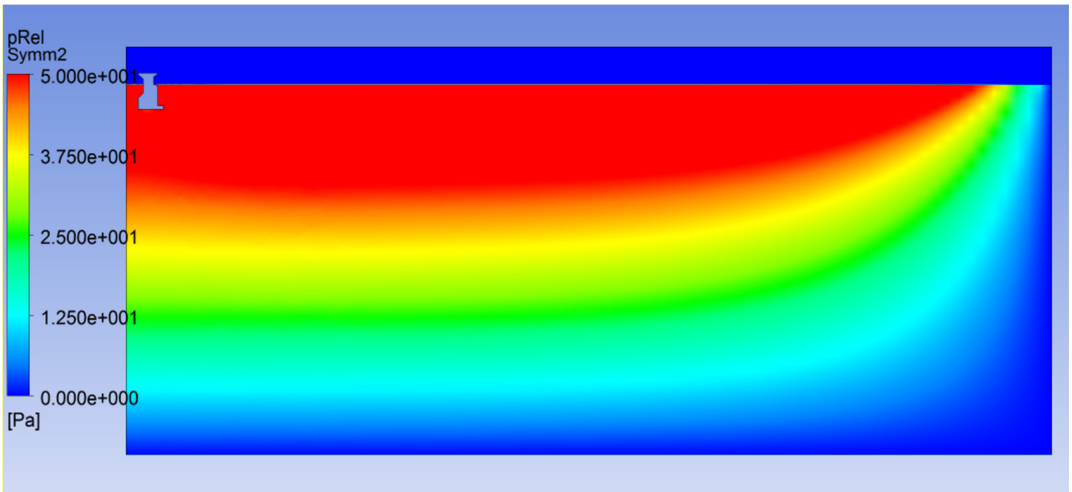

Figure 4. Deformation of the dynamic pressure field in the nearby of the farfield.

ISSN 2176-5480

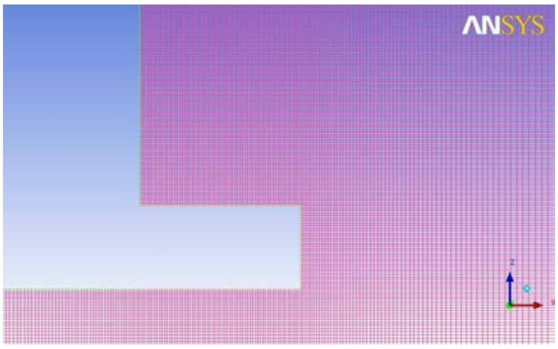

Figure 5. Detail of the fine grid mesh adopted around the skirt.

The cell-aspect ratios in the region around the internal skirt/bottom of the hull were 1.0/1.0. At the bottom and the outer regions of the domain, a relatively coarse mesh was used, with cell-aspect ratio of 50/1. A linear transformation law was adopted to cell-aspect ratio variation. The boundary conditions at the hull, outer edge and the symmetry planes are immediately defined. The dynamics of the air and the water flows at the top of the domain were modeled adopting an opening type boundary condition.

The equations governing the problem were solved using the well-known code ANSYS CFX, release 14. The code solves the Reynolds averaged Navier-Stokes equations (RANS) through a finite volume approach. The solution procedure is based on a fully implicit discretization of the governing equations. In the present work, the well-known eddy viscosity model SST (Shear Stress transport) κ–ω Based Model was adopted for the computation of the turbulent properties. The Volume of Fluid (VOF) method was adopted for the simulation of free surface flow. The Rigid Body solver offered by CFX was assumed for the computations of the interacting forces between the water and the hull.

The κ–ω model considers that the turbulent viscosity, ν_τ, is related to the turbulent kinetic energy, κ, and the turbulent frequency, ω, through the expression

$$\nu_\tau = \kappa/\omega \tag{2}$$

The κ–ω based model formulation has become very popular over the last few years for its apparent superior performance for the treatment of near wall conditions. The κ–ω model does not require the introduction of the typical non-linear dumping functions present in the κ–ε model and, for this reason, should be more accurate and robust. As a matter of fact, the κ–ω model can be resolved with a near wall resolution of $y^+ < 2$.

The two transport equations for the κ–ω model can be written as

$$\frac{\partial \kappa}{\partial t} + \overline{u}_i \frac{\partial \kappa}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\upsilon + \left(\frac{\upsilon_t}{\sigma_\kappa}\right)\right)\frac{\partial \kappa}{\partial x_i}\right] + P_k - \beta' \kappa\omega \tag{3}$$

$$\frac{\partial \omega}{\partial t} + \overline{u}_i \frac{\partial \omega}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\upsilon + \left(\frac{\upsilon_t}{\sigma_\omega}\right)\right)\frac{\partial \omega}{\partial x_i}\right] + \alpha \frac{\omega}{k} P_k - \beta\omega^2 \tag{4}$$

where $\overline{u}$ represents the mean flow velocity.

The κ–ω model constants are given by

$$\alpha = 5/9$$
$$\beta = 0.075\ ;\ \beta' = 0.09$$
$$\sigma_\kappa = 2\ ;\ \sigma_\omega = 2$$

The Shear Stress Transport (SST) κ–ω Model accounts for turbulent shear stress transport by considering

$$\upsilon_\tau = \frac{\alpha\kappa}{\max(\alpha\omega, SF_2)} \tag{5}$$

where F_2 is a blending function and S is an invariant measure of the strain rate.

The blending function F_2 is given by

$$F_2 = \tanh(\arg_2^2)\text{, with} \tag{6}$$

$$\arg_2 = \max\left(\frac{2\sqrt{k}}{\beta' \omega y}, \frac{500\upsilon}{y^2\omega}\right) \tag{7}$$

4. NUMERICAL RESULTS

The numerical simulations covered the first 10.0 s of the experimental tests. A time-step of 0.01 s was considered adequate for the discretization of the flow in time. A residual target of 10^{-6} (RMS) was adopted for the convergence criteria, that should be achieved in a maximum number of 20 iterations. A sequence of velocity field plots showing the velocity field around the monocolumn is presented in Figures 6 and 7. The sequence of plots presented comprehends the time interval in which are observed the higher motion amplitudes.

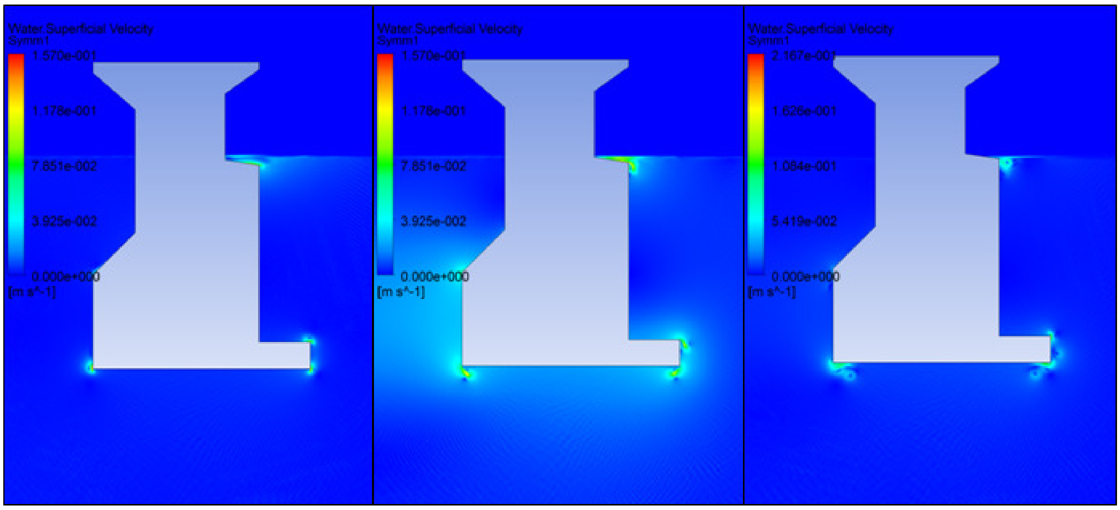

Figure 5. Velocity field around the monocolumn: 1.5, 2.0 and 2.5 s.

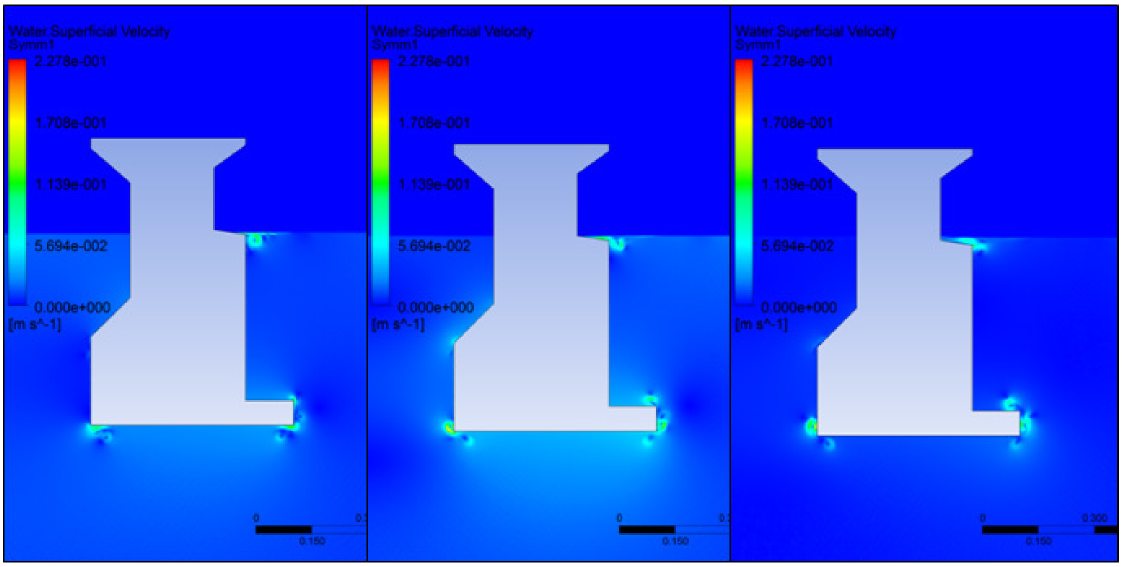

Figure 6. Velocity field around the monocolumn: 3.0, 3.5 and 4.0 s.

ISSN 2176-5480

It can be seen that complex vortex structures are formed around the skirt, as well as near the inner bottom edge of the moonpool. The vortexes structures grow until the vertical motion of the model reaches its maximum displacement. As expected, once the direction of the vertical motion of the model is changed, the direction of vortex rotation is inverted. In the course of the oscillations, a higher amount of energy is transferred to the vortexes that have been formed, resulting in larger structures. It is important to notice that becuse the small amplitude motion imposed to the model the vortexes initially formed did not vanish. These vortexes structure actually acts as a buffer of damping energy.

In Fig. 7 the numerical results obtained for the normalized heave decaying curve are compared to the experimental data. As can be seen, a good agreement between the numerical and the experimental results was achieved for the frequency motion response: $T_{Z,exp}$ = 2.80 s, $T_{Z,num}$ = 2.78. These results suggest that the numerical model was able to reproduce the physics of the flow around the model for the case that the damping is not of significance, where $\omega_z \approx \omega_0$ applies. This is the case of the heave decay test adopted as reference, since $\kappa_{\text{exp}} = 0.036 << 0.2$.

On the other hand, the numerical results for the amplitude response did not show a good adhesion to the experimental values. The discrepancies are more pronounced for the later oscillations. This behavior is a predictable consequence of an underestimation of the damping forces acting on the hull. Based on Equation (7), the numerical damping coefficient is equal to

$$\kappa_{\text{num}} = 0.025 \,,$$

since $z(t)$ = -0.00924 m, when t = 1.39 s, and $z(t+T_z)$ = -0.00788 m, when t =4.18 s (for T_z = 2.78 s).

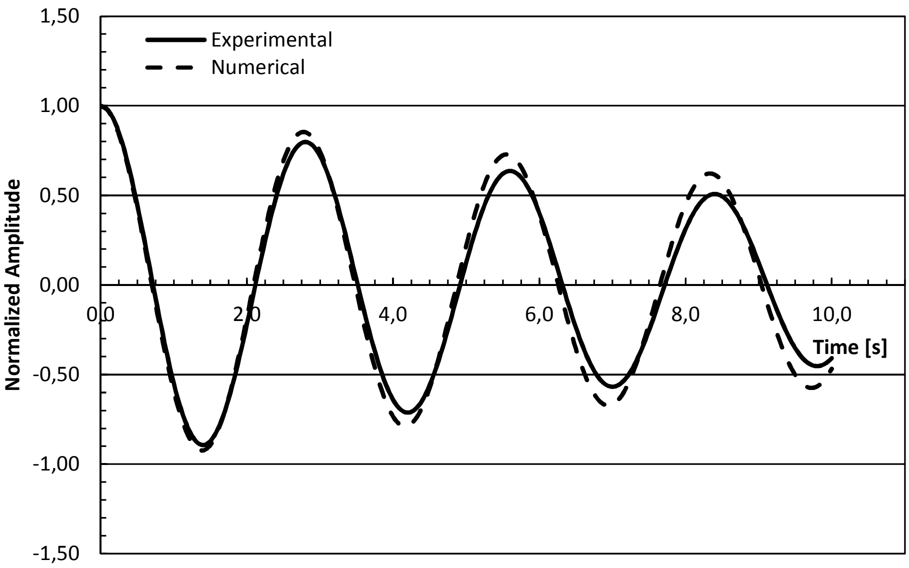

Figure 7. Comparison between numerical and experimental heave decaying curves.

As can be seen in Figures 5 and 6, the energy dissipation in the heave decay motion is governed by the vortex formation process occurring around the bottom edges and, particularly, around the skirt. The higher the vortex structure the higher dissipation of energy. It is well known that the energy associated with a vortex structure is related to the magnitude of the velocity field. The installation of a skirt in a floating device has the aim to cause a remarkable effect on its decaying motion. Not only the amplitude of the motion was influenced but also the period of oscillation is modified by the presence of the skirt.

In experimental heave decay tests, the model is usually deflected to an initial vertical displacement in still water and then released. To reduce the possibility of inducing secondary motions, as pitch for example, the initial vertical displacement is usually small. The decay test adopted as reference for this study was performed with an initial displacement of 0.01 m (model scale). Considering that the viscous effects originated by the skirt and the inner bottom edge are a function of the model vertical velocity, such a small initial displacement will result in low damping forces, whose estimation is very sensitive to numerical errors. It must be also noted that part of the amplitude discrepancies may have been caused by the uncertainties associated with the modeling of the contribution of the mooring and the risers system to the damping of the heave motion in model scale.

ISSN 2176-5480

4. CONCLUSIONS

A numerical model was developed in order to obtain a better comprehension of the dynamics of the damping process of the heave motion of a monocolumn platform. The simulation results obtained for the heave decay test of a reference monocolumn platform were presented and discussed. It was observed that a good agreement between the numerical and the experimental results was achieved for the frequency motion response, suggesting that the numerical model was able to reproduce the physics of the flow around the model for the case that the damping is not of significance. However, the numerical results obtained for the amplitude response did not show a good adhesion to the experimental valuesIt was discussed that the discrepancies observed in the amplitude response may be the result of low damping forces caused by the small initial displacement of the model, whose estimation is very sensitive to numerical errors. Further investigations must be conducted in order to enhance the performance of the numerical model in computing the amplitude response in heave motion.

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