NUMERICAL SIMULATION ON MOTION OF SEVEN CYLINDRICAL FPSO IN DIFFERENT LOADING CONDITIONS

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ABSTRACT

This paper is proposed to present the difference of motion response in wave of SEVEN’s Cylindrical FPSO when the structure is in full loaded and when it is in ballast condition. Based on the theory, the motion of the floating structure will be influenced on the weight of FPSO. Where, when the weight of the FPSO changed, the natural frequency of the motion will be changed. This will lead to shift of the peak response frequency and amplitude of the motion. To analyse the motion of the FPSO in different loading condition, the study was conducted by simulate the motion of FPSO in wave by using commercial software. The heave and pitch of the FPSO are focused. The added mass, damping coefficient, wave loading and Response amplitude operator of the FPSO in different loading condition are presented in the paper. From the simulation results, it is observed that the peak heave response of the FPSO in full loaded condition happened at lower frequency as compared to ballast condition. However, the peak pitch response of the FPSO in full loaded condition happened at higher frequency as compared to ballast condition. This result obtained is due to the displacement of the FPSO at full loaded condition is higher than ballast condition but the GM value is smaller at full load condition.

Keywords: Seven Cylindrical FPSO, Loading Condition, RAO, motion behavior.

1.0 INTRODUCTION

Nowadays, the cylindrical FPSO is being extensively used as an offshore facility in the oil and gas industry. This system has been deployed widely around the world as a unique design facility which is regarded as a promising concept for an economic oil production since it has capability for storage and wider deck that is giving better layout flexibility. Moreover, it has also the ability to move and relocate after the operation is completed and is suitable for all offshore environments meeting the challenges of the oil and gas industry.

As a floating offshore system, a cylindrical FPSO will be deployed together with slender members (moorings and risers) responding to wind, wave and current loading in complex ways. In the traditional way, the hydrodynamic interaction among the floater, moorings and risers cannot be evaluated since the floater, moorings and risers are treated separately. Moreover, this traditional method, also known as the decoupled analysis, the hydrodynamic behavior of the system is only
based on hydrodynamic behavior of the hull and ignores all or part of the interaction effects (mass, damping, stiffness, current loads) between the floater, moorings and risers.

In the project, the Western Isles Development Project (WIDP) that is located in the UKCS, Block 210/24 to the North East of Shetland will be taken as reference case. Moreover, the WIDP has shallow water conditions and has harsh environment. These two major characteristics will influence the design of the overall system of the floating offshore system. To simulate the motion of the FPSO, a cylindrical $S400$ floater was selected as the sample of FPSO model. The detailed model for each component, characterization of the environments in covering relevant load models and the simulation schemes will be presented in this paper.

This paper mainly focuses on the Hydrodynamics Diffraction simulations of the cylindrical Sevan FPSO. First, the Seven FPSO 6 degree of freedom RAO will be simulated. Due to the cylindrical shape of the FPSO, the paper will only presented the heave and pitch motion of the FPSO in different loading condition.

2.0 LITERATURE REVIEW

The vertical plane motions induced by heaving, rolling and pitching should be kept adequately low to guarantee the safety of the floating structure, risers and umbilical pipes and other important facilities use in oil production [1]. The operability and safety of floating bodies operation are greatly influenced by the relative motions between them. So, the accurate motion prediction of two bodies including all hydrodynamic interactions is important [2].

Normally motions of floating structures are analyzed by using strip theory and potential theory. A number of notable studies were carried out to solve the problem of hydrodynamic interactions between multi bodies by Ohkusu (1974) [3]; Kodan (1984) [4]; and Fang and Kim (1986) [5]. They used strip theory to analysis of hydrodynamic interaction problem between two structures positioned side by side.

Hess and Smith (1964) [6], Van Oortmerssen (1979) [7] and Loken (1981) [8] studied on non-lifting potential flow calculation about arbitrary 3D objects. They utilized a source density distribution on the surface of the structure and solved for distribution necessary to take the normal component of the fluid velocity zero on the boundary. Plane quadrilateral source elements were used to approximate the structure surface and the integral equation for the source density is replaced by a set of linear algebraic equations for the values of the source density on the quadrilateral elements. By solving this set of equations, the flow velocity both on and off the surface was calculated. Wu et al. (1997) [9] studied on the motion of a moored semi-submersible in regular waves and wave induced internal forces numerically and experimentally. In their mathematical formulation, the moored semi-submersible was modeled as an externally constrained floating body in waves, and derived the linearized equation of motion.

Yilmaz and İnceciş (1996) [10] analyzed the excessive motion of moored semi-submersible. They developed and employed two different time domain techniques due to mooring stiffness, viscous drag forces and damping; there are strong nonlinearities in the system. In the first technique, first-order wave forces acting on a structure which considered as a solitary excitation force and evaluated by using Morison equation. In the second technique, mean drift forces are used to calculate slowly varying wave forces and simulation of slow varying and steady motions. Söylemez (1995) [11] developed a technique to predict damaged semi-submersible motion under wind, current and wave. Newton’s second law of approaching equations of motion was used in the research to develop numerical techniques of nonlinear equations for intact and damaged condition in time domain.

Choi and Hong (2002) [12] applied HOBEM to analysis hydrodynamic interactions of multi-body system. Clauss et al. (2002) [13] analyzed the sea-keeping behavior of a semi-submersible in rough waves in the North Sea by numerical and experimental method. They used panel method TiMIT (Time-domain investigations, developed at the Massachusetts Institute of Technology) for
wave/structure interactions in time domain. The theory behind TiMIT is strictly linear and thus applicable to moderate sea condition only.

An important requirement in determining drilling capabilities of the structure is the low level of motions in the vertical plane (motions induced by heave, roll and pitch). Matos et al. (2011) [1] numerically and experimentally investigated second-order resonant of a deep-draft semi-submersible heave, roll and pitch motions. One of the manners to improve the hydrodynamic behavior of a semi-submersible is to increase the draft. The low frequency forces computation has been performed in the frequency domain by WAMIT a commercial Boundary Element Method (BEM) code. The code can generate a different number of meshes on the structure and calculated pitch forces.

Since demand for oil and gas is growing up, the water depth is becoming deeper and deeper, and chance of multi body operation increasing, so investigating reliability of numerical analysis method for hydrodynamic interaction is worthwhile (Hong et al. 2005) [14].

Zahra Tajali et. Al. (2011) [15] studied hydrodynamic characteristic of multi-body floating pier under the wave action by numerical method. In the research, pontoon of the floating piers is connected to each other by hinge. The research found that when the number of pontoons increases, peak frequency and peak amplitude for all motion increase. R.C. Zhu et.al (2006) [16] applies numerical methods to study the effect of gap in the multiple structural system. In that study, wave potential for incident wave and scattering wave were ignored. The motion of the structures is assumed only affected by radiated wave. The simulation showed that hydrodynamic interaction between floating structures can cause surge, sway and heave motion; however, the only sway motion shows strong interaction effect in certain resonance wave number. Besides, that study also obtained that the increase of gap width caused the resonance amplitude for added mass and damping coefficients decrease significantly.

Masashi Kashiwagi (2010) [17] was carried out a numerical investigation to compare Wave interaction Theory with the Higher Order Boundary Element Methods, HOBEM. In comparison, the wave interaction theory is able to compute integrated force and pressure distribution if the condition satisfied the Bessel Function and mathematic limitation of the theory. In addition, the research also found that reductions of distance between floating bodies will cause deviation of pressure distribution increase.

Zhu et.al. (2008) [18] was carried out a research on hydrodynamic resonance phenomena of three dimensional multiple floating structures by using time domain method. The linear potential theory in time domain was used to describe fluid motion. The research found that peak force response on floating bodies at resonance frequency is same between frequency domain technique and time domain technique. This finding proved that the linear potential theory in the time domain can be an alternative to solve a problem related to hydrodynamic interaction between floating bodies in small gap.

### 3.0 MATHEMATICAL MODEL

The simulation of the study was conducted using commercial software. The theory behind the software is diffraction potential theory. The main theoretical assumptions and limitations of the theory employed are listed below:

1. The body or bodies have zero or very small forward speed.
2. The fluid is inviscid and incompressible, and the fluid flow is irrotational.
3. The incident regular wave train is of small amplitude compared to its length (small slope).
4. The motions are to the first order and hence must be of small amplitude. All body motions are harmonic.

Since the first order potential theory of diffraction and radiation waves is used here for radiation and diffraction analysis, the linear superposition theorem may be used to formulate the velocity
potential within the fluid domain. The fluid flow field surrounding a floating body by a velocity potential is defined by:

\[ \Phi(\vec{X}, t) = a_w \varphi(\vec{X}) e^{-i \omega t} \]  

(1)

Where \( a_w \) the incident wave amplitude and \( \omega \) is the wave frequency.

The isolated space dependent term \( \varphi(\vec{X}) \) may be separated into contributions from the radiation waves due to six basic modes of body motion, the incident wave and the diffracted or scattered wave. The potential functions are complex but the resultant physical quantities such as fluid pressure and body motions in time domain analysis will be obtained by considering the real part only.

Three translational and three rotational motions of the body's center of gravity are excited by an incident regular wave with unit amplitude:

\[
x_j = u_j, \quad (j = 1, 3)
\]

\[
x_j = \dot{\theta}_j, \quad (j = 4, 6)
\]

(2)

The potential due to incident, diffraction, and radiation waves may therefore be written as:

\[
\varphi(\vec{X}) e^{-i \omega t} = \left( \varphi_1 + \varphi_d \right) + \sum_{j=1}^{6} \varphi_{rj} x_j e^{-i \omega t}
\]

(3)

Where \( \varphi_1 \) is the first order incident wave potential with unit wave amplitude, \( \varphi_d \) is the corresponding diffraction wave potential, \( \varphi_{rj} \) is the radiation wave potential due to the \( j \)-th motion with unit motion amplitude.

In finite depth water, the linear incident wave potential \( \varphi_1 \) at point \( X = (X, Y, Z) \) in Equation 3 has been given, but as a special case of unit amplitude, \( a_w = 1 \).

When the wave velocity potentials are known, the first order hydrodynamic pressure distribution may be calculated by using the linearized Bernoulli's equation.

\[
\rho(1) = -\rho \frac{\partial \Phi(\vec{X}, t)}{\partial t} = i \omega \varphi(\vec{X}) e^{-i \omega t}
\]

(4)

From the pressure distribution, the various fluid forces may be calculated by integrating the pressure over the wetted surface of the body.

\[
(n_1, n_2, n_3) = \vec{n}
\]

\[
(n_4, n_5, n_6) = \vec{r} \times \vec{n}
\]

(5)

Where \( \vec{r} = \vec{X} - \vec{X}_g \) is the position vector of a point on the hull surface with respect to the center of gravity in the fixed reference axes (FRA). Employing this notation, the first order hydrodynamic force and moment components can be expressed in a generalized form:
\[ F_{g} e^{-i\omega t} = -\int_{S_0} p^{(1)} n \cdot dS = [-i\omega \int_{S_0} \varphi(\mathbf{X}) n \cdot dS] e^{-i\omega t} \]  (6)

Where \( S_0 \) is the mean wetted surface of body.

From Equation 3, the total first order hydrodynamic force can be written as,

\[ F_j = [ (F_{ij} + F_{dj}) + \sum_{k=1}^{6} F_{ijk}^{x} x_k ] \text{ where } j = 1, 6 \]  (7)

of which the \( j \)-th Froude-Krylov force due to incident wave is

\[ F_{ij} = -i\omega \int_{S_0} \varphi_j(\mathbf{X}) n \cdot dS \]  (8)

the \( j \)-th diffracting force due to diffraction wave is,

\[ F_{dj} = -i\omega \int_{S_0} \varphi_d(\mathbf{X}) n \cdot dS \]  (9)

the \( j \)-th radiation force due to the radiation wave induced by the \( k \)-th unit amplitude body rigid motion is,

\[ F_{ijk} = -i\omega \int_{S_0} \varphi_{rk}(\mathbf{X}) n \cdot dS \]  (10)

Fluid forces can be further described in terms of reactive and active components. The active force, or the wave exciting force, is made up of the Froude-Krylov force and the diffraction force. The reactive force is the radiation force due to the radiation waves induced by body motions.

The radiation wave potential, \( \varphi_{rk} \) may be expressed in real and imaginary parts and substituted into Equation 10 to produce the added mass and wave damping coefficients

\[ F_{ijk} = -i\omega \int_{S_0} \{ \text{Re}[\varphi_{rk}(\mathbf{X})] + i\text{Im}[\varphi_{rk}(\mathbf{X})] \} n \cdot dS \]
\[ = \omega \int_{S_0} \text{Im}[\varphi_{rk}(\mathbf{X})] n \cdot dS - i\omega \int_{S_0} \text{Re}[\varphi_{rk}(\mathbf{X})] n \cdot dS \]
\[ = \omega^2 A_{jk} + i\omega B_{jk} \]  (11)
Where the added mass and damping are:

\[
A_{jk} = \frac{1}{\alpha} \int_{S_0} \text{Im} \left[ \varphi_{rk} (\overline{X}) \right] n_j dS \\
B_{jk} = -\rho \int_{S_0} \text{Re} \left[ \varphi_{rk} (\overline{X}) \right] n_j dS
\]

(12)

If a problem requires the wave loading on a fixed body, then only the active wave forces are of interest. When the body is floating, both the active and reactive fluid forces must be considered. It is also worth noting that all fluid forces calculated above are a function of the wetted body surface geometry only and are independent of the structural mass characteristics of the body.

Once the wave excitation force, damping and added mass are obtained, the motion of the body is able to calculate using the motion equation as in follow

\[
\left[ -\omega^2 (M_S + M_a) - i\omega \epsilon C + K_{\text{hys}} \right] [x_{jn}] = [F_{jn}]
\]

(13)

Where \(M_S\) is a 6M × 6M structural mass matrix, \(M_a = [A_{jm, kn}]\) and \(C = [B_{jm, kn}]\) are the 6M×6M hydrodynamic added mass and damping matrices including the hydrodynamic interaction coupling terms between different structures, \(K_{\text{hys}}\) is the assembled hydrostatic stiffness matrix of which each diagonal 6×6 hydrostatic stiffness sub-matrix corresponding to individual structure, and all off-diagonal 6×6 sub-matrices are null as there is no hydrostatic interaction between different structures.

4.0 MODEL SETUP AND ENVIRONMENTAL CONDITION

The Sevan Floating, Production, Storage and Offloading vessel (FPSO) S400 is used for the floating production and storage of hydrocarbons. It has capability to store hydrocarbons within the range from 300 to 2,000,000 bbls. The main particulars for S400 FPSO are summarized in Table 1. Two different platform drafts are specified for fully loaded and ballast conditions. Further, the 3D model and 2D model can be seen in Figure 1 and Figure 2.

![Figure 1: S400 FPSO - 3D model [19]](image-url)
Figure 2: S400 FPSO - 2D model [19]

Table 1: S400 FPSO Main Particulars [19]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter Main Hull Cylinder</td>
<td>m</td>
<td>70.0</td>
</tr>
<tr>
<td>Diameter Main Deck</td>
<td>m</td>
<td>78.0</td>
</tr>
<tr>
<td>Diameter Process Deck</td>
<td>m</td>
<td>84.0</td>
</tr>
<tr>
<td>Area Process Deck</td>
<td>m²</td>
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</tr>
<tr>
<td>Diameter Pontoon</td>
<td>m</td>
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</tr>
<tr>
<td>Height Pontoon</td>
<td>m</td>
<td>25.5</td>
</tr>
<tr>
<td>Elevation Main Deck</td>
<td>m</td>
<td>32.0</td>
</tr>
<tr>
<td>Elevation Process Deck</td>
<td>m</td>
<td>38.0</td>
</tr>
<tr>
<td>Elevation Start Flare</td>
<td>m</td>
<td>24.0</td>
</tr>
<tr>
<td>Radius of Gyration in Roll</td>
<td>m</td>
<td>22.3</td>
</tr>
<tr>
<td>Radius of Gyration in Pitch</td>
<td>m</td>
<td>22.3</td>
</tr>
<tr>
<td>Radius of Gyration in Yaw</td>
<td>m</td>
<td>32.0</td>
</tr>
<tr>
<td>Ballast Draft</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Draft</td>
<td>m</td>
<td>16.35</td>
</tr>
<tr>
<td>Displacement</td>
<td>Ton</td>
<td>70690</td>
</tr>
<tr>
<td>Freeboard to MD</td>
<td>m</td>
<td>15.7</td>
</tr>
<tr>
<td>Freeboard to PD</td>
<td>m</td>
<td>20.7</td>
</tr>
<tr>
<td>VCG</td>
<td>m</td>
<td>19.1</td>
</tr>
<tr>
<td>GM (inc correction for free surface)</td>
<td>m</td>
<td>6.5</td>
</tr>
<tr>
<td>Loaded draft</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Draft</td>
<td>m</td>
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</tr>
<tr>
<td>Displacement</td>
<td>Ton</td>
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<tr>
<td>Freeboard to PD</td>
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<td>16.3</td>
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<tr>
<td>VCG</td>
<td>m</td>
<td>18.23</td>
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<tr>
<td>GM (inc correction for free surface)</td>
<td>m</td>
<td>6.2</td>
</tr>
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</table>
In order to obtain a good result from the model, a sufficient mesh must be used. A more detailed mesh density does however also mean longer simulation times and since many simulations need to be run, as short simulation time as possible is sought after. The simulate was conducted by setting a maximum mesh element size of 5 meter and a tolerance of 2 meters on the S400 model as shown in Figure 3. Since the mesh does not seem to fluctuate very much with varying mesh size and since the values does not vary very much, the result in a mesh with 3120 elements were obtained.

![Figure 3: The mesh on the S400 model](image)

5.0 RESULT AND DISCUSSION

The calculated added mass coefficient, damping coefficient and wave loading coefficient are presented in Appendix 1. The heave RAO curve for the FPSO in ballast condition is as shown in Figure 4.

![Figure 4: RAOs for Heave motions in ballast condition](image)
From Figure 4, it can be seen that the RAO for Heave motion does not differ much as the direction of the force changes from 0° to 90°. The RAOs show a rapid increasing trend from the beginning of the simulation until it reaches the frequency of about 0.07 Hz, which is the natural frequency of the FPSO Heave motion. After that, the RAO decreases gradually to near zero.

The natural frequency has to be avoided because the RAO is about 3.6 m/m. This means that at this frequency, the FPSO Heave amplitude response will be about 3.6 times larger than wave amplitude. According to the equation of motion for surge, sway and heave, RAO is influenced by added mass, damping and restoring force coefficients. Therefore, the values of Heave RAO are much bigger than for Surge and Sway because the added mass is much bigger but damping is much smaller.

The heave RAO curve for the FPO at full load condition is presented in Figure 5. From the figure, it can be seen that the RAO for Heave motion does not differ much as the direction of the force changes from 0° to 90°. The RAOs show a rapid increasing trend from the beginning of the simulation until it reaches the frequency of about 0.067 Hz, which is the natural frequency of the FPSO Heave motion. The RAO of the FPSO at natural frequency simulated is 4.12 m/m. After that, the RAO decreases gradually to near zero.

![Figure 5: RAOs for Heave motions in full-load condition](image)

The RAO for pitch of the FPSO in ballast condition is shown in Figure 6. The RAOs show a rapid increasing trend from the beginning of the simulation until it reaches the frequency of about 0.035 Hz, which is the natural frequency of the FPSO Pitch motion. After that, the RAO decreases rapidly, increases slightly and decreases again to near zero. The Pitch RAO is largest for 0° because the FPSO is in head-sea position and smallest for 90° the FPSO is in beam-sea position. The natural frequency has to be avoided especially in head sea condition. This is because the RAO is more than 7 °/m at it natural condition in head sea. This means that at this frequency, the FPSO Pitch amplitude response will be about 7 times larger than wave amplitude.
Figure 6: RAOs for Pitch motions in ballast condition

The RAO for pitch of the FPSO in full load condition is shown in Figure 7. Similar as in the ballast condition, the RAOs show a rapid increasing trend from the beginning of the simulation until it reaches the frequency of about 0.038 Hz, which is the natural frequency of the FPSO Pitch motion. After that, the RAO decreases rapidly, increases slightly and decreases again to near zero. The Pitch RAO is largest for 0° because the FPSO is in head-sea position and smallest for 90° the FPSO is in beam-sea position. Operate the body at pitch natural frequency has to be avoided because the RAO is more than 4 °/m.

Figure 7: RAOs for Pitch motions in full-load condition
6.0 CONCLUSION

The paper discussed the wave motion of cylindrical FPSO at full load condition and ballast condition. The study was conducted using the numerical software which developed using diffraction potential theory. Based on the simulation result, it is found that the peak heave response of the FPSO is larger at full load condition and the peak heave response of the FPSO is happened at lower frequency as compared to ballast condition. However, for the pitch RAO, the peak pitch response of the FPSO at full load condition is smaller as compare to ballast condition with the natural frequency of pitch motion at the full load condition is slightly higher than in ballast condition. The pitch natural period for full load condition and ballast condition is happened at 28.6s and 26.3s respectively. This period is far from most of the average period of large ocean wave happened. Therefore, the pitch motion for this FPSO should be small at most operation condition. However, the heave natural period for full load and ballast condition is happen at 14.9s and 14.3s respectively. At this period, the resonance has larger possibility to happen. Therefore, if this type of FPSO is selected to use, then the vertical motion of the FPSO need to proper evaluate to avoid the damage to riser due to excessive motion.

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REFERENCES


**APPENDIX**

Figure 8: Added mass for Surge, Sway and Heave motions in (a) ballast condition, (b) full-load condition
Figure 9: Added mass for Roll, Pitch and Yaw motions in (a) ballast condition, (b) full-load condition

Figure 10: Damping coefficient for Surge, Sway and Heave motions in (a) ballast condition, (b) full-load condition

Figure 11: Damping coefficient for Roll, Pitch and Yaw motions in (a) ballast condition, (b) full-load condition
Figure 12: Diffraction and Froude-Krylov forces for Heave motion in (a) ballast condition, (b) full-load condition

Figure 13: Diffraction and Froude-Krylov moments for Pitch motion in (a) ballast condition, (b) full-load condition