PROCEEDING

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International Society of Ocean, Mechanical, Aerospace - scientists & engineers -
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About OMAse

The first Conference on Ocean, Mechanical and Aerospace for Scientists and Engineers is the conference organizes by International Society of Ocean, Mechanical and Aerospace – Scientists and Engineers – (ISOMAse).

The target for this conference is gathered the researchers involved in an area of ocean, mechanical and aerospace to share their findings and discuss the researches issue. We believe interchangeable of idea between researchers from ocean, mechanical and aerospace is important because all of the disciplines are sharing related sciences and engineering theories in their respect area. Therefore, we believe this conference will able to refresh the research experience of the people in these areas and generate impact for the cross discipline research collaboration and knowledge exchange.
Scope of OMAse

The OMAse welcomes participants from academicians, scholars, and practitioners for possible publication from all over the world that meets the general criteria of significance and educational excellence. The scope of the conference is as follows:

- Environment and Safety
- Renewable Energy
- Naval Architecture and Offshore Engineering
- Computational and Experimental Mechanics
- Hydrodynamic and Aerodynamics
- Noise and Vibration
- Aeronautics and Satellite
- Engineering Materials and Corrosion
- Fluids Mechanics Engineering
- Stress and Structural Modeling
- Manufacturing and Industrial Engineering
- Robotics and Control
- Heat Transfer and Thermal
- Power Plant Engineering
- Risk and Reliability
- Case studies and Critical reviews

The International Society of Ocean, Mechanical and Aerospace –science and engineering is inviting you to submit your manuscript(s) to isomase.org@gmail.com for the conference proceeding.
ABSTRACT

This paper is purposed to present the experiment procedures to study the hydrodynamic characteristic of new generation Round-Shape FPSO. This is because response of FPSO in wave is required to modeling correctly to ensure the safety of FPSO in dynamic condition. In this experiment, the FPSO model is constructed based on scale ratio 1:110 and it is installed in wave dynamic tank with model size mooring lines. Besides that, this paper also discussed the preparation procedures for model test which included the mooring design process and model setup before experiment. From the model experiment data, it is observed that the round FPSO which stationary by mooring lines experienced two types of horizontal motion there are wave frequency motion and slow varying motion due to drift and mooring effect. While the vertical motion of this FPSO is only experience wave frequency motion.

KEY WORDS: Round FPSO, Model Experiment; Mooring Lines Setup.
FPSO was also conducted by A. Efi et al. using RANs Method [3].

Also, Srinivasan et al (2008) proposed a non-ship-shaped circular FPSO concept which has a capability for round-the-year drilling and production operation within an Arctic Frontier Region. The structure is designed to work in clear water open-sea and in ice-covered arctic environment with least ice-management requirement. The study on this new concept including ice-structure interaction, stability, storage, constant draft, motion, moon pool resonance, moorings and station keeping, flexible riser system, disconnect turret system, offloading operation, fabrication, transportation, and installation.

Besides, Wang, Zhang, and Liu (2012) were also proposed a non-ship-shape FPSO called inverted fillet quadrangular frustum pyramid-shaped FPSO (IQFP). The hydrodynamic characteristic of this structure was studied by using three-dimensional potential flow theory in frequency domain simulation. The obtained result was compared to ship-shaped FPSO with similar tonnage. The results indicate that the IQFP has large stability margin and better hydrodynamic performance than the ship-shaped FPSO.

2.0 MODELLING RULE

In this study, the FPSO model and mooring lines in model scale are scaling follow the Froude similarity. Froude’s law of similarity is the most appropriate scaling law applicable for the free and moored floating structure experiments. The Froude number has a dimension corresponding to the ratio of \( u^2/gD \) where \( u \) is the fluid velocity, \( g \) is the gravity acceleration and \( D \) is a length of the model or prototype. The Froude number \( Fr \) is defined as \( Fr = u^2/gD \).

Let the subscripts \( p \) and \( m \) stand for prototype and model respectively and \( \lambda \) is the scale factor, then the scaling for length, speed, mass and force is shown in Table 1.

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Scaling equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>length, ( L ) (m)</td>
<td>( L_m = \lambda L_p )</td>
</tr>
<tr>
<td>speed, ( \dot{\theta} ) (m/s)</td>
<td>( \dot{\theta}_m = \lambda \dot{\theta}_p )</td>
</tr>
<tr>
<td>mass, ( m ) (kg)</td>
<td>( m_m = \lambda^3 m_p )</td>
</tr>
<tr>
<td>Force, ( F ) (N)</td>
<td>( F_m = 1.025 \lambda^3 F_p )</td>
</tr>
<tr>
<td>Mooring line stiffness in water, ( K ) (N/m)</td>
<td>( K_m = 1.025 \lambda^2 K_p )</td>
</tr>
</tbody>
</table>

3.0 TABLE AND FIGURE

In this study, the round shape FPSO model was designed and constructed to test in model experiment so the hydrodynamics characteristic of this designed model can be simulated. The round shape FPSO model was constructed based on full scale model where it diameter at designed draught is 112m. In the experiment, the round shape FPSO model was scaled down with ratio 1:110.

Upon the model complete constructed, few tests were carried out to ensure the model particulars is suited to the design. Firstly, inclining test and decay test were carried out to identify the hydrostatic particular for the round shape FPSO model. The Figure 1 showed the inclining test conducted before model experiment to check the vertical center of gravity of the structure. Besides, natural periods for the roll motions also obtained from decay test. The dimension for the models was summarized as in Table 2.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Model</th>
<th>Fullscale</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter (m)</td>
<td>1.018</td>
<td>111.98</td>
</tr>
<tr>
<td>Depth (m)</td>
<td>0.4401</td>
<td>48.41</td>
</tr>
<tr>
<td>Draught (m)</td>
<td>0.2901</td>
<td>31.91</td>
</tr>
<tr>
<td>Free board (m)</td>
<td>0.150</td>
<td>16.5</td>
</tr>
<tr>
<td>Displacement (m³)</td>
<td>0.2361</td>
<td>314249</td>
</tr>
<tr>
<td>KG (m)</td>
<td>0.225</td>
<td>24.8</td>
</tr>
<tr>
<td>GM (m)</td>
<td>0.069</td>
<td>7.6</td>
</tr>
</tbody>
</table>

4.0 INSTRUMENT OF MODEL TEST

The round shape FPSO was assumed to experience six degrees of freedom during the experiment. The linear DOF motions of the FPSO models on model size mooring are measured by theodolite camera system (Figure 2a). The theodolite camera is able to capture the positions of the reflective optical tracking markers (Figure 2b) placed on the FPSO model automatically. The Rotational DOF motions of the FPSO models are measured by gyscroscope (Figure 3) installed in the center of buoyancy of the FPSO.

A servo-type wave height measurement device (Servo 式波高計) as showed in Figure 4 is attached to the carriage which located at the position between FPSO model and wave generator to record the wave height generate by the wave generator. All the measurement design is linked to separate computer to maximize the consistency of the measuring speed. To synchronize the device and ensure all device start and stop measure the data without delay, the Wireless remove controller as showed in Figure 5 is used to given the order to start and stop all measurement devices.
5.0 MOORING SETUP

In this experiment, four model scale mooring lines attached to the Round Shape FPSO to provide horizontal restoring force to the model. The mooring lines in full scale is designed by using the catenary theory and then scaled down to model scale follow by the scaling rule explained in section 2 of this paper [F]. The mooring line profile used in this experiment is showed in Figure 6 and the segment particular is showed in Table 3. The size of these mooring lines is pre-determined before the model experiment and the suitability of the mooring lines is analyzed using numerical simulation method to simulation the mooring performance in both static and dynamic condition.

Also, due to the effect of mooring lines, the Round Shape FPSO will experience a non-linear horizontal restoring force during the experiment. The horizontal restoring force able to provide from the mooring lines to the FPSO model in order to
return the FPSO back to the original position is showed in Figure 7.

Figure 6: Mooring line profile

Table 3: Mooring line segment information

<table>
<thead>
<tr>
<th>Particular</th>
<th>Segment A</th>
<th>Segment B</th>
<th>Segment C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal Diameter (mm)</td>
<td>3.0</td>
<td>3.0</td>
<td>3.0</td>
</tr>
<tr>
<td>Type</td>
<td>Chain</td>
<td>Wire Rope</td>
<td>Chain</td>
</tr>
<tr>
<td>Segment Length (m)</td>
<td>4.0</td>
<td>9.4</td>
<td>1.4</td>
</tr>
<tr>
<td>Air Weight (kg/m)</td>
<td>0.16</td>
<td>0.0369</td>
<td>0.16</td>
</tr>
<tr>
<td>Water weight in water (kg/m)</td>
<td>0.1425</td>
<td>0.03119</td>
<td>0.1425</td>
</tr>
<tr>
<td>Water density: 1000kg/m³</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Breaking Load (KN)</td>
<td>10.79</td>
<td>5.40</td>
<td>10.79</td>
</tr>
<tr>
<td>Modulus Elasticity (GPa)</td>
<td>114.59</td>
<td>61.00</td>
<td>114.59</td>
</tr>
</tbody>
</table>

Figure 7: FPSO restoring force due to mooring effect.

5.0 EXPERIMENT SETUP

This experiment is conducted in wave dynamic tank with the length, wide and depth of 60m, 25m and 3.2m respectively. Before the experiment start, the Round FPSO model was fixed in the middle of tank by four mooring lines which connected between the fairleads located in bottom of FPSO with the anchors which sink into the bottom of tank. The position of the anchors and the arrangement of the mooring lines inside the tank are shown in Figure 8 and the view of FPSO model inside wave dynamic tank after installed with mooring lines is showed in Figure 9. Each anchor used in the experiment has the weight of 20kg in air.

Besides, to ensure the wave height measure by wave measuring device is not influence by the exist of Round FPSO, the carriage installed by the servo-type wave height measurement device is moved to the location between FPSO and wave generator where the distance between the FPSO to wave measuring device and the distance between wave generator to wave measuring device are 15m both.

Figure 8: Arrangement of mooring lines and anchor in wave basin.

Figure 9: FPSO model fixed with mooring lines in wave basin in static condition.
6.0 RESULT AND DISCUSSION

In this study, the model experiment is conducted in head sea condition with the wave period in model scale from 0.7 seconds to 2.5 seconds and the wave height is 0.04m for all selected wave periods. The example of collected time domain motions data at wave period 1.7 seconds is presented in figure 10 to 12. The Figure 10, Figure 11 and Figure 12 are presented the surge, heave and pitch motion of FPSO in time series collected from the model experiment.

From Figure 10, it is shown that the surge motion experienced two types of motions at this wave period. First, the surge motion of FPSO model in this wave period is very depend on the wave frequency motion. Due to the effect of drift force and mooring lines, the FPSO also experienced slow varying motion and this can observe from the Figure 10. In Figure 10, the wave frequency motion is represented by the blue color line while the slow varying motion can be observed by link all peak points of blue color line.

Compared to surge motion, heave and pitch motion is only experience the wave frequency motion. From the Figure 11 and 12, it is shown that the amplitude of heave and pitch motion are around 0.035m and 0.125degree. At this wave period, the heave motion is experience the resonance motion where the computed heave RAO by frequency domain calculation is around 1.7. Besides, the pitch motion amplitude calculated from the data also showed that the round FPSO model have good stability in the wave period 1.7s where the amplitude of pitch is only 0.125deg.

6.0 CONCLUSION

This paper was presented the procedures applied to study the hydrodynamic characteristic of round FPSO model by experiment method. In the experiment, the horizontal restoring force is provided by 4 model scale mooring lines scaled down from the full scale size. The examples of time series motion data collected from model experiment in wave period 1.7s and wave height 0.04m are presented in the paper. Due to the effect of the mooring lines and drift force, the surge motion of the FPSO experienced two types of motions there are slow varying motion and wave frequency motion. Besides, the sample of time series data also showed that the experiment success to capture the motion response of FPSO model due to the incoming wave.

ACKNOWLEDGMENT

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REFERENCES


The Floating Production, Storage and Offloading Vessel Design for Oil Field Development in Harsh Marine Environment

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ABSTRACT
The oil and gas exploration and production activities in deep sea are now on a steady increase globally. Therefore, it is necessary to design a cost effective and safe system for these operations. The main objective of this research is to design a Floating Production, Storage and Offloading (FPSO) vessel suitable for operation even in extreme meteorological and oceanographic conditions. In order to achieve this, the effects of extreme environmental loads on the vessel have been evaluated in terms of the maximum responses in surge, heave and pitch modes of motion. Furthermore, an interactive programme, the Principal Dimensions Programme (PD Prog) has been designed to accurately evaluate and optimise the principal particulars based on the required storage capacity and response analyses. Results show that the vessel length, which is directly proportional to the cube root of the cubic number (the overall volume), is a measure of the critical wavelength. Close to the critical wavelength in extreme metocean condition, the vessel could be subjected to several billions Newton meter of Wave Bending Moment. This design technique, in addition to the numerous useful data obtained, helps to ensure good performance during operation and so reduces downtime, and increases uptime, safety and operability of the vessel even under extreme metocean conditions.

KEY WORDS: FPSO, principal dimensions, extreme environmental loads, responses, wave bending moment.

NOMENCLATURE
w = weight/length of cable line in water
a = The horizontal component of cable tension per w
$S_{\text{min}}$ = Minimum separation of heave- and-pitch zeros

1.0 INTRODUCTION
The conceptualization and creation of floating storage vessels became imperative and feasible when the offshore oil industry began to grow in the second half of the twentieth century. The first floating storage vessels were then installed to reduce the cost of transporting oil ashore for storage before shipping it elsewhere. These first floating storage units (FSU) were tankers that stayed moored for a few days to weeks. These vessels were developed with the single point mooring system. This mooring system allows for the vessel to be positioned such that environmental impacts are minimized.

Platform operators began to look into vessels that would remain on station for periods of months to years. This type of vessel would have to be offloaded by a shuttle tanker. The logical progression was to convert mid-sized tankers into the floating, storage and offloading (FSO) vessels. These vessels however, still did not produce the oil. Thus, the oil had to be processed on a platform. Companies saw removing the platform as a way to reduce the cost of production. This led to the idea of putting production topsides on the FSO vessels. These developed into floating production, storage, and offloading (FPSO) vessels. The early FPSO vessels were tanker conversions which eventually led to drastic reduction in available fleet of tankers and so provoking the designing and building of new ones.

Generally, the needs related to the use of ship-shaped offshore units (FSU, FSO, and FPSO etc.) and their technical challenges for the development of offshore oil and gas in deep water are given by Henery and Inglis[1], Bensimon and Delvin[2] and Hollister and Spokes [3] among others.

These offshore units have proven to be reliable and cost-effective solutions for the development of offshore fields in deep waters of more than 1,000m depth, as they have successfully been applied for more than 38 years in such harsh environments.
It is noteworthy that a concrete barge with steel tanks became the first dedicated FPSO application and it was operated by Arco in the Ardjuna field in the Java Sea offshore Indonesia in 1976 [4], while the first tanker-based single-point moored FPSO facility is the FPSO Castellon for Shell offshore Spain in 1976. Since then, the application of FPSOs and other related offshore structures has grown very rapidly, and will remain a mainstay in the oil and gas industry for many years to come as they provide the flexibility and sound economics of producing and storing at the offshore well sites. Thus the oil is produced, safely stored and then directly transported to the refinery.

1.1 The Main Objectives

The purpose of this study is to design an FPSO capable of withstanding harsh metocean conditions. In other words, the research investigates the impact of harsh or extreme environmental forces on FPSO and establishes reliable methods and tools for prediction of environmental loads and structural responses. The dynamic behaviour, induced motions or responses of the vessel under the influence of these metocean forces are vital to the stability and safety of both the vessel and crew and so will be evaluated. The FPSO is to be designed for worldwide operation. To achieve this objective, the extreme metocean forces, associated extreme motion responses as well as the shear forces and bending moments for the design environment will be determined. That is, specifically, the objectives include the following:

- Predict extreme vessel motion responses associated with harsh marine environment comparison with the benign wave (North Sea and West African).
- Develop simpler methodology and programs for quick determination of design data. The design of the vessel's principal dimensions required for the development of any given oil field will be carried out based on the specified required storage capacity of the vessel.
- Evaluate the dynamic wave bending moment amidships. This is required in order to ensure that the hull girder has sufficient strength to withstand the induced stress.

2.0 DESIGN METHODOLOGY

The design of floating structures is usually carried out following a well-defined design spiral as a guide. This project therefore follows a simply-defined design spiral to accomplish the desired goal(s). The FPSO Design Spiral (FDS) usually starts with the identification of the vessel owner's requirements. The elements of the spiral include, but not restricted to the following steps: (i) Owner's Requirements, (ii) Environments, (iii) Hydrostatics, (iv) Motions, and (v) Structure. In order to meet the owner's requirements such as the required storage capacity, it is important to ensure that the right principal dimensions of the vessel are evaluated as demonstrated in the following sections.

Generally, the following preliminary design objectives are adopted for optimal design of vessels which are to be operated the harsh wave environment such as the North Sea:

(i) The storage capacity or volume must be capable of taking the output during the average interval of shuttle tanker calls plus about 3 days.
(ii) The value of the transverse metacentric height, $GM_T$, must be around 3 or more, in the fully-loaded condition.
(iii) The natural rolling period must be greater than 12 seconds. Also, the natural pitching and heaving periods must be as long as possible. Usually, a good design usually has the natural motion periods longer than the peak period of the spectrum which is exceeded for less than 2% of the time and low heave forces and pitch moments at all shorter periods. Table 1 gives the wave periods and wavelengths for four sea areas which illustrate the problems involved. For instance, the peak periods exceeded 2% of the time in the Central and Northern North Seas are 12.3s and 15.4s respectively.
(iv) In order to ensure a better motion response, the zero force frequencies for heave and pitch must be spread out as much as possible.
(v) The ratio $L/D_m$ must be less than 13 (from structural point of view).
(vi) In order to accommodate the segregated ballast and the produced water storage capacity, the underwater volume should not exceed 1.8 times the displacement. This implies that: $B/D_m \leq 1.8$.
(vii) The required external surface areas should be as small as possible, which implies low $L/B$ and $B/D_m$ ratios.
(viii) The induced motions should not exceed the levels within which the separators have been designed to operate. Conventional separators have been designed to cope with the following levels of motion: Angular motions, 0 to 7.5°; linear motions, 0 to 0.25g; periods, 3 to 15s [5].

<table>
<thead>
<tr>
<th>Areas</th>
<th>Pierson-Moskowitz</th>
<th>JONSWAP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Central North Sea</td>
<td>Tp [s]</td>
<td>Tp [s]</td>
</tr>
<tr>
<td>8.7</td>
<td>12.3</td>
<td>236</td>
</tr>
<tr>
<td>Northern North Sea</td>
<td>10.9</td>
<td>15.4</td>
</tr>
<tr>
<td>West of Shetland</td>
<td>11.3</td>
<td>15.9</td>
</tr>
<tr>
<td>Brazil</td>
<td>10</td>
<td>14.1</td>
</tr>
</tbody>
</table>

Spectral Analyses are carried out for each of the vessels with above preliminary objectives being applied as design constraints in the computer programmes written in MATLAB. The PD Programme and the WavBem have been carefully written to evaluate the optimal principal dimensions and the wave bending moment distribution using the required storage and efficiency as major inputs to the programmes.

2.1 Owner's Design Requirements

Vessels are often designed to perform specific function(s). The FPSOs are used mainly for production and storage of crude oil (and periodically offloaded to shuttle tankers for transportation.
to the refinery or market). Therefore, most vessel owners require reasonably high storage capacity and large deck area for topside installation.

Major oil fields in the Niger Delta area of Nigeria have oil reserves up to 1000 million bbls of oil. Agbami, Bonga, Forcados-Yokri, and Erha fields have oil reserves of 1000, 600, 1235, and 1200 million bbls respectively [6]. See Table 2.

Therefore, most vessel owners would require FPSOs that would be capable of storing up to 2 million bbls. Agbami FPSO has storage capacity of 2.2 million bbls. It is therefore important to have a reasonably sufficient specific storage capacity in mind as an initial design requirement. In this paper, we will be considering a storage capacity of 2 million bbls and will also be assuming that this vessel is meant for unrestricted service location. It is therefore imperative to consider in the design stage, the effects of extreme environments in which its services may be required.

Table 2: Major Oil Reserves in the Niger Delta of Nigeria

<table>
<thead>
<tr>
<th>Operator</th>
<th>Fields</th>
<th>Reserves (mmbbls)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell</td>
<td>Bonga</td>
<td>600</td>
</tr>
<tr>
<td></td>
<td>Forcados-Yokri</td>
<td>1235</td>
</tr>
<tr>
<td></td>
<td>Nembe Creek</td>
<td>950</td>
</tr>
<tr>
<td>Mobil</td>
<td>Erha</td>
<td>1200</td>
</tr>
<tr>
<td></td>
<td>Ubit</td>
<td>945</td>
</tr>
<tr>
<td>Chevron Texaco</td>
<td>Agbami</td>
<td>1000</td>
</tr>
<tr>
<td></td>
<td>Meren</td>
<td>1100</td>
</tr>
</tbody>
</table>

2.2 The Wave Environment

There are several challenging wave environments in which oil and gas exploration activities still take place. The North Sea of the United Kingdom is a very good example of such. Any offshore floating structure designed for this region can be redeployed to other locations for operation since most adverse effects of very rough, irregular and phenomenally high wave conditions might have been accounted for.

In offshore structural design, it is convenient to describe the wave environment in spectral form. The general form of the wave spectrum model is given by:

The parameters (A, B) of the Spectrum are solved in terms of the spectrum model is given by:

\[ \frac{H^2}{T^2} = \frac{124}{\alpha} \exp \left[ -496.1 (\alpha T_s)^{-4} \right] \]  

(8)

The rectangular-shaped floating production, storage and offloading vessel with length L, Beam B and draught T, (which are evaluated based on the required storage capacity as given in eqns. 1-5) is be operated in the North Sea of 100-year Return Period storm; the zero up-crossing period and significant wave height are 17.5s and 16.5m respectively. The equation of motion of this vessel is given by:

\[ \left( M_{jk} + A_{jk} \right) \ddot{x}_k + \eta_j \eta_k + C_{jk} \dot{x}_k = F_j \]  

(9)

Where: \( M_{jk} \) are the elements of the generalized mass matrix for the structure; \( A_{jk} \) are the elements of the added mass matrix; \( \eta_j \) are the elements of the linear damping matrix; \( C_{jk} \) are the elements of the stiffness matrix; \( F_j \) are the amplitudes of the wave exciting forces and moments, \( j \) and \( k \) indicate the directions of fluid forces and the modes of motions; \( \eta_k \) represents responses; \( \dot{x}_k \) and \( \ddot{x}_k \) are the velocity and acceleration terms; and \( \alpha \) is the angular frequency of encounter.

2.3 Hydrostatics

The elements of the stiffness matrix or the hydrostatic restoring force coefficients, \( C_{jk} \), are important in the station-keeping of the vessel and therefore must be carefully evaluated. In surge mode, it can be shown that the uncoupled restoring coefficient, which is largely contributed by the mooring lines, may be given by:

\[ C_{11} = w \left[ \cosh^{-1} \left( 1 + \frac{h}{a} \right) - 2 \left( 1 + \frac{2a}{h} \right)^{-2} \right] \]  

(10)

The stiffness or coefficients of restoring force and moment in heave and pitch motions can be estimated as functions of the buoyancy due to a unit length of sinking respectively.

\[ C_{33} = pgBL \]  

(11)

\[ C_{55} = Mg \times GM_L = pgLBT \times \frac{L^2}{12T} = pgB \frac{L^3}{12} \]  

(12)

2.4 The Principal Dimensions of FPSO

There are three major factors that greatly influence the size and arrangements of these different parts of the Floating Production, Storage and Offloading system and its process plants. These are: (i) Provision of sufficient oil storage capacity, (ii) Provision of enough topside area or space for process plants, accommodation, helideck and other required topside equipment and (iii) Provision of displacement and ballast capacity. These factors are directly related to (or functions of) cubic number, length-breath (x_b) and breath-depth (y_d) ratios (as variables in the analysis) respectively. The cubic number is the overall volume of the vessel and it is directly proportional to the required storage capacity. With the knowledge of the oil storage efficiency, the cubic number and the preliminary evaluation of the principal dimensions can made. The overall volume or the cubic number \( C_n \) is given by:

\[ C_n = \frac{L^3}{x_b^2 \times y_d} = \frac{B^3}{y_d/x_b} \]
\[ \frac{D^2}{(T/D)^2} = \frac{\nabla}{(T/D)^2} = \left( \frac{S_c}{C_f \times E_s} \right) \]  
From eqn. (1), it follows that:

\[ \text{The Length, } L = f_1 \left( \frac{S_o}{C_f \times E_s} \right)^{1/3} \]  
\[ \text{Breadth, } B = f_2 \left( \frac{S_o}{C_f \times E_s} \right)^{1/3} \]  
\[ \text{Depth, } D = \left( f_1 f_2 \right)^{-1/3} \left( \frac{S_o}{C_f \times E_s} \right)^{1/3} \]  
\[ \text{Draught, } T = z_m D \]  

From eqn. (17)

\[ \forall \text{is the displacement, and the new dimensionless factors are: } f_1 = \frac{x_s^2 \times y_d}{2}; \quad f_2 = \frac{y_d}{x_s}^{1/3}; \quad z_m = \sqrt{C_n} \]  

\[ S_o: \text{Required oil storage capacity in barrel (bbl)}; \quad E_s: \text{Oil storage Efficiency}; \quad C_f = 6.28981077; \text{and } 6.28981077 \text{bbl} = 1 \text{m}^3. \]

### 3.0 WAVE LOADS AND RESPONSES

#### 3.1 Surge Force and Response

In surge mode of motion, the acceleration or added mass force is out of phase with the Froude-Krilov Force. It will be wrong to add them up algebraically. Since the added mass force is very small compared to the Froude-Krilov force especially within the relevant frequency range, the surge excitation force amplitude, \( F_s \), is usually taken to be approximately equal to the amplitude of the Froude-Krilov (pressure force), \( F_{FK} \) as given in Eq. (18).

\[ \frac{F_s}{z_o} \approx 2 \left( \frac{\rho g B}{k} \right) \left( 1 - e^{-kt} \right) \sin \left( \frac{kl}{2} \right) \]  
\[ F_s \approx \rho g z_o \left( \frac{B}{k} \right) \left( 1 - e^{-\pi k l} \right) \sin \left( \frac{ml}{2} \right) \]  

Therefore, the Surge Response Amplitude Operator, \( RA_{O1}\), is:

\[ RA_{O1} = \frac{F_{s1}}{CT z_o} = \frac{2F_s}{CT} \left( \frac{\rho g B}{k} \right) \left( 1 - e^{-kt} \right) \sin \left( \frac{kl}{2} \right) \]  

\( Q_s \) is the surge dynamic amplification factor.

#### 3.2 Heave Force and Response

Assuming the vessel has a constant mass density, zero forward speed and moored in deep sea, with a sinusoidal wave propagating along the negative x-axis (head sea), the velocity potential is:

\[ \phi = g \frac{z_o}{\omega} e^{\mu x} \cos (\omega t + \mu x) \]  

The vessel is divided into strips of equal sizes and the force acting on each strip (\( dF_p \)) is the sum of the pressure force and the added mass force. These forces are integrated across the length of the vessel to obtain the expression for the heave excitation force.

\[ dF_p = \rho g B dx + A_{23}^{(2D)} a dx = -\frac{\partial}{\partial t} \left( \rho g B dx + A_{23}^{(2D)} \frac{\partial^2 \phi}{\partial z \partial t} \right) dx \]  
\[ = \zeta \left( \rho g B - A_{23}^{(2D)} g \right) e^{-\pi t} \sin(\omega t + \mu x) dx \]  
\[ F_s = \zeta \left( \rho g B - A_{23}^{(2D)} g \right) e^{-\pi t} \frac{\pi}{2} \sin(\omega t + \mu x) dx \]  
\[ = 2\zeta \left( \frac{\rho g B}{k} - A_{23}^{(2D)} \right) e^{-\pi t} \left( \frac{kl}{2} \right) \sin(\omega t) \]  

Where \( A_{23}^{(2D)} \) is the 2-D added mass in heave, while the amplitude of the heave force is given by:

\[ F_{s2} = 2\zeta \left( \frac{\rho g B}{k} - A_{23}^{(2D)} \right) \left( e^{-\pi t} \sin \left( \frac{kl}{2} \right) \right) \]  
\[ = \rho g z_o \left( \frac{B}{k} \right) \left( 1 - c_v \pi \right) \left( \frac{B}{k} \right) \left( e^{-\pi t} \sin \left( \frac{kl}{2} \right) \right) \]  

Therefore, the Heave Response Amplitude Operator, \( RA_{O2}\), defined as the heave amplitude per wave amplitude, is:

\[ RA_{O2} = \frac{F_{s2}}{z_o} = \frac{2\zeta \rho g z_o \left( \frac{B}{k} \right)}{\sin \left( \frac{kl}{2} \right) \left( 1 - c_v \pi \right) \left( \frac{B}{k} \right) \left( e^{-\pi t} \sin \left( \frac{kl}{2} \right) \right) \]  

\( Q_h \) Dynamic amplification factor in heave; \( \lambda : \text{wavelength} \); \( c_v \): virtual added mass coefficient in heave; \( \zeta \): wave amplitude; and wave number, \( k = 2\pi/\lambda \).

Both heave force, \( F_{s2} \) and response, \( RA_{O2} \) will be equal to zero when \( \left( \frac{B}{k} \right) = c_v \pi \left( \frac{B}{k} \right) \) or \( \left( \frac{kl}{2} \right) \sin \left( \frac{kl}{2} \right) \) is equal to zero. These happen at wavelengths of \( \frac{2\pi^2 B}{4} \), \( L, L/2, L/3 \) etc.

#### 3.3 Pitching Moment and Response

The amplitude of the pitching moment has also been obtained following similar procedure and it is given by:

\[ F_{s2} = \rho g z_o \left( \frac{B}{k} \right) \left( 1 - e^{-\pi t} \right) \sin \left( \frac{kl}{2} \right) \]  

So, the Pitch Response Amplitude Operator, \( RA_{O3} \), defined as the pitch response amplitude per wave amplitude, is:

\[ RA_{O3} = \frac{F_{s3}}{z_o} = \frac{2\zeta \rho g z_o \left( \frac{B}{k} \right)}{\sin \left( \frac{kl}{2} \right) \left( 1 - e^{-\pi t} \right) \sin \left( \frac{kl}{2} \right) \left( 1 - c_v \pi \right) \left( \frac{B}{k} \right) \left( e^{-\pi t} \sin \left( \frac{kl}{2} \right) \right) \]  

\( Q_p \) is the dynamic amplification factor in pitch motion.

The pitch moment, \( F_{s3} \) and its corresponding response, \( RA_{O3} \) will be equal to zero if \( \left( \frac{B}{k} \right) = c_v \pi \left( \frac{B}{k} \right) \) or \( \left( \frac{kl}{2} \right) \cos \left( \frac{kl}{2} \right) \sin \left( \frac{kl}{2} \right) \) is equal to zero. These happen at wavelengths of \( \frac{2\pi^2 B}{4} \), \( L, L/2, L/3, L/4, L/4.49 \) etc. To ensure that the vessel has a very good motion performance, these wavelengths must be well-separated from one another.

Investigations show that the minimum separation of heave- and-pitch zeros is given by:

\[ S_{min} = \min \left( L - \frac{c_v \pi B}{4}, \frac{L}{1.43} - \frac{c_v \pi B}{4} \right) \]  

The overall induced kinetic energy due to wave impact is the sum of the energies in the corresponding modes of modes of motions. For heave and pitch modes of motion, this energy depends on the values of energy coefficients, \( \varepsilon \), at various wavelengths.

\[ \varepsilon = \frac{F_{s2}}{F_{s2}} \frac{RA_{O2}}{RA_{O2}} \]  

The wavelengths at which these energy coefficients, \( \varepsilon \), tend to infinity are called critical wavelengths, \( \lambda_c \). It is important to note...
that these phenomena occur at heave zeros. That is, \( \frac{L}{\lambda cr} = N \), where \( N = 1, 2, 3, \ldots \)

### 4.0 DYNAMIC WAVE BENDING MOMENT

#### 4.1 Wave Induced Shear Force

The Shear Force at any point from the one end is the integral sum of the contributions from wave excitation force, restoring force and inertia force and damping force. The Shear Force, \( Q_x \) from one end of the vessel is therefore given by:

On the elemental strip:

\[
\begin{align*}
\text{d}F_x &= \xi_a \left( \rho g B - A_{23}^{(2D)} g \right) e^{-kT} \sin(\omega t + kx) \text{d}x \\
\text{d}F_I &= \left( \rho BT + A_{23}^{(2D)} \right) \text{d}(\eta_3 - x\eta_3) \\
\text{d}F_R &= \left[ \rho g B \text{d}(x\eta_3 - x\eta_3) \right] \\
\text{d}F_D &= B_{33}^{(2D)} \text{d}(x\eta_3 - x\eta_3) \\
Q_x &= \int_0^X \left( \text{d}F_x - \text{d}F_I - \text{d}F_R - \text{d}F_D \right) 
\end{align*}
\]

#### 4.2 Wave Bending Moment

The vertical dynamic bending moment at any point from the one end is the integral sum of the contributions from wave excitation load, restoring load, and inertia moment load and damping load.

\[
M_x = - \int_0^X x \text{d}Q_x 
\]

\[
\begin{align*}
M_x &= \xi_a \left( \frac{\rho g B}{k} - A_{23}^{(2D)} g \right) e^{-kT} \left[ kX \cos(kX) - \sin(kX) \right] \cos(\omega t) \\
&\quad + \frac{1}{2} X^2 \left( \rho BT + A_{23}^{(2D)} \right) \left( \eta_3 - \frac{2}{3} \xi_3 \right) \\
&\quad + \frac{1}{2} X^2 \left( \rho g B \right) \left( \eta_3 - \frac{2}{3} \xi_3 \right) \\
&\quad + \frac{3}{2} B_{33}^{(2D)} \omega RAO_5 \\
&\quad + \frac{3}{2} B_{33}^{(2D)} \omega RAO_5 \\
&\quad + \frac{3}{2} \rho g B RAO_5 \\
&\quad (27)
\end{align*}
\]

Let \( \frac{M_x}{\xi_a} = I_1 \sin(\omega t) + I_2 \cos(\omega t) \)

\[
I_1 = \left( \frac{\rho g B}{k} - A_{23}^{(2D)} g \right) e^{-kT} \left[ kX \cos(kX) - \sin(kX) \right] \\
+ \frac{1}{2} X^2 \left( \rho BT + A_{23}^{(2D)} \right) \omega^2 RAO_5 \\
+ \frac{3}{2} \rho g B RAO_5 \\
(28)
\]

The amplitude of the bending moment distribution per unit wave amplitude is expressed as:

\[
\left( \frac{M_x}{\xi_a} \right)_{\text{amplitude}} = \left( I_1^2 + I_2^2 \right)^{1/2}
\]

### 5.0 RESULTS AND DISCUSSIONS

From the design 1 (see Table 3) analyses (which are executed in the PD program with other subroutines described in this paper), the most probable maximum amplitudes of surge, heave and pitch motions are 13.4m, 11.3m and 7.1° respectively. The linear motions have a maximum acceleration of 0.9m/s² or 0.09g. These are all within the acceptable levels of motion within which the conventional separators can cope.

However, this vessel could experience wave bending moment up to 10 billion Newton meter in this design harsh wave condition. All the preliminary design objectives were achieved except that the minimum separation of heave and pitch zeros is small (about 10m).

| Design 1: (Note: \( S_{\text{min}} \approx 10 \)) |
|-----------------|-------|-------|-------|-------|-------|-------|
| \( L \) | \( B \) | \( T \) | \( L/B \) | \( B/D \) | \( B/T \) | \( c_v \) |
| 306.4 | 56.7 | 22.1 | 5.4 | 1.8 | 2.57 | 1.46 |
| | | | | | | |
| | | | | | | |

**Table 3: Analysis of Rectangular Block Design for Oil Storage Capacity of 2 million barrels**
Required Oil Storage Capacity and Vessel Size

<table>
<thead>
<tr>
<th>Sc</th>
<th>Es [%]</th>
<th>M [t]</th>
<th>V [m$^3$]</th>
<th>Hull Area</th>
<th>TopArea</th>
<th>TransArea</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>58</td>
<td>393356</td>
<td>383762</td>
<td>57679</td>
<td>17390</td>
<td>1789</td>
</tr>
</tbody>
</table>

BM (Predicted) | BM (ABS) | BM (DNV) |
Sagging        | 9329     | 7066    | 7040      |
Hogging        | 9982     | 7180    | 7743      |

5.0 CONCLUSIONS

A series of formulae have been given and systematically programmed for the determination of an optimal set of principal dimensions to meet a specified field output in a given harsh environment (the North Sea conditions were chosen).

The extreme responses in surge, heave and pitch motions have been evaluated and these are all within the acceptable levels of motion required for the smooth operation of the oil separators. Therefore, operational downtimes are minimized.

The critical wavelengths have been found to be prime factors of the vessel length (which is directly related to the cubic number).

There should be sufficient separation of heave and pitch zeros as this is necessary to improve the performance of the vessel. Larger separation of heave and pitch zeros also leads to the reduction in the induced wave bending moment acting on the vessel.

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REFERENCE

Theoretical Study on Effect of Turret Location on Dynamic Response of Moored Twin Hulls FPSO System

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ABSTRACT
It currently no information is available for the comparison of the dynamic behaviour of the internal turret moored for twin hull FPSO system at different loading conditions with various turret locations under the action of wave. Hence, the present study investigated characteristics for a typical turret moored FPSO system by catenary anchor leg mooring (CALM) which subjected to sea waves, in order to get insight knowledge on its dynamic behaviour due to various turret locations with different loading conditions. The comparison of the dynamics behaviour to the FPSO and it mooring lines are important when choosing potential development and optimal options. This research will analyses and highlight the optimal turret location to the new potential concepts of twin hull FPSO. It also will highlight areas where effort is best focussed to mitigate the marine risks.

KEY WORDS: Moored Twin Hulls; FPSO; Turret Location; Dynamic Response.

1.0 INTRODUCTION
Offshore structures have been in use successfully for many years and rapidly developing. They serve the same purpose in the oil and gas production as well as the storage system. There are two common the type of offshore structure which are fix structure and floating structure. These floating structures include Tension Leg Platform (TLP), Semi-submersible platform, Floating Storage and Offloading (FSO), and also Floating Production Storage and Offloading (FPSO).

Since the offshore drilling are being discovered in deep sea area the fixed structure are not practical because it have significant heave, pitch and yaw motions in large wave. Good stability characteristic as a drilling platform was make the floating structure attempt to replace traditional fixed jacket platforms. These movable structures have the maneuverability ability to be used in several fields, but the cost effectiveness is the main advantages of movable structures.

FPSO is vessel used by the offshore industry for the processing and storage of oil and gas. A FPSO vessel is transportable platform which designed to receive oil or gas produced from nearby platforms or subsea template, process it, and store it until oil or gas can be offloaded onto a tanker or transported through a pipeline. FPSOs are preferred in frontier offshore regions as they are easy to install, and do not require a local pipeline infrastructure to export oil and gas.

The vessels often take the form of traditional tankers. In addition to dedicated vessels that are designed for FPSO, oil and gas tankers can be converted to an FPSO vessel which also makes them an economical and flexible option. The vessels are moored in place either via a single point or by spread mooring which involves the vessel being anchored via multiple points on the sea floor. This allows them to operate in both deepwater and ultra deepwater environments which are designed to take into account local weather situations and can even be detached from moorings which make them ideal in extreme weather conditions.

Turret is a typical single point mooring system for FPSO. Turret also is the major component used for station keeping in harsh environments; heavy sea, high winds and strong current which the risers coming from seabed are connected. The most important function of turret is weathervaning which allows vessel to rotate freely around turret.

Floating production vessel have the principal characteristic of remaining at substantially stable position, presenting movements when they experience environmental forces such as the wind, waves and currents. In floating structure design, it is important to...
determine its motion and behavior when subjected to waves (Sharma et al. 2010).

According to the research from T.R Kannah and R. Natarajan (2006), it has been illustrated that the position of the turret location plays a vital role to determine the behavior of the FPSO and also its mooring lines forces when subjected to environmental impact. Throughout this research, it tries to investigate the influence of turret location to the dynamic behavior and the mooring lines force of the floating production system. This study is made to access the optimum operational capability of the FPSO system due to the dynamic behavior and mooring lines force condition.

This research is tried to improve the response prediction by using diffraction potential theory by involving drag effect from the Morison equation in the calculations. The main objective is too carried out the hydrodynamic response of the twin hull FPSO using the correction method that has been done by Siow et al., (2014b) thus applied to moorings and risers as top end excitation for the dynamic response analysis.

2.0 LITERATURE REVIEW

It has been forecasted that between 2013 and 2017, 91 billion dollar will be spent on floating production systems, an increase of 100 percent over the preceding five year period (Westwood, 2013). With global economic growth in 2014 projected to increase to 3.5 percent from 2.9 percent in 2013, world oil demand is forecast to rise by one million barrels per day, compared with 900 000 barrels per day in year 2013 (Cochran, 2014). This massive growth in the floating production sector has come as a result of the rapid evolution that gives the impact to the future development of FPSO.

IEA (2012) forecasted that oil and gas demand will rise from 3.3 trillion cubic meters in 2010 to 5.0 trillion cubic meters in 2035, an increase of 50 percent. This situation has driven the market for FPSO to fulfill the demand. As a result of this situation the new concept of twin hull FPSO has emerged. This novelty is achieved by joining together two FPSO which the process facilities along with the storage and crew living quarters are located on deck. This twin hull concept allows adequate space for the process facilities with the necessary space between sections of equipment to satisfy safety requirements, while providing sufficient oil and gas storage capacity.

The environmental loads on moored structures, namely due to wind, current, and waves, are of a main concern when determining vessel motions and evaluating mooring design (C. G. Soares, et al. 2005). Turret-moored FPSO systems are sensitive to the effect of waves, wind and current. In the recent years, considerable research is being carried out on turret moored FPSO system operating at offshore locations. An internal turret moored FPSO system is an attractive concept for both production facilities and offshore storage (T.R Kannah and R. Natarajan, 2006).

In an internal turret moored FPSO system, the turret structure is built inside the tanker’s hull and it is attached to the sea-bed by catenary anchor leg mooring (CALM). The spider part of the turret located at the vessel keel level includes bearings, allowing the vessel to rotate freely around its mooring legs in response to changes in environmental excitation and system dynamics. In the case of internal turret moored FPSO system, the vessel motions and mooring forces are mainly governed by the location of the turret so as to maintain optimal operating conditions.

Many research have been done up to now about the single point mooring for monohull FPSO. Thiagarajan and Finch (1998, 1999) conducted an experimental investigation of the vertical motions of a turret moored FPSO in wave using different positions of the mooring along the length of the model. The results show that the mooring location affects the vertical motions and accelerations of the FPSO. Bemittas and Papoulas (1986) conducted the study on the yaw and stability of single point mooring. Yaw of turret moored vessels in regular waves was investigated Liu et al. (1999). O’ Donoghue and Linfoot (1992) performed the model test in irregular waves and reported the effect of turret position and mooring load characteristics. Jiang et al. (1995) extensively reported the horizontal motions and mooring line loads of single point moored tanker. Cho (2012) and Cho et al. (2013) studied the motion behavior and stability of turret moored floating body and two bodies including sloshing. Recently, Seok et al. (2013) conducted the model test and stability analysis for a turret moored Floating Storage Regasification Unit (FSRU). Model tests are performed in regular waves. The results of model test show that the possibility of large yaw in irregular wave can be predicted by the regular wave tests.

Based on the review of these existing literatures, it is found that no information is available for the comparison of the dynamic behaviour of the internal turret moored for twin hull FPSO system at different loading conditions with various turret locations under the action of wave. Hence, the present study investigation has been programmed for a typical turret moored FPSO system by catenary anchor leg mooring (CALM) which subjected to sea waves, in order to get insight knowledge on its dynamic behaviour due to various turret locations with different loading conditions. The comparison of the dynamics behaviour to the FPSO and it mooring lines are important when choosing potential development and optimal options. This research will analyses and highlight the optimal turret location to the new potential concepts of twin hull FPSO. It also will highlight areas where effort is best focussed to mitigate the marine risks.

This research is targeted to propose a correction method which applicable to linear diffraction theory in order to evaluate the motion response of selected moored floating structure. The linear diffraction theory estimate the wave force on the floating body based on frequency domain and this method can be considered as an efficient method to study the motion of the large size floating structure with acceptable accuracy. The effectiveness of this diffraction theory apply on large structure is due to the significant diffraction effect exist on the large size structure in wave (Kvittem et al., 2012).

In this study, twin hull FPSO will select as an offshore structure model since this structure is one of the new concept structure used in deep water oil and gas exploration area. To achieve this objective, a programming code will develop based on diffraction potential theory and it is written in visual basic programming language. By comparing the numerical result predicted by using diffraction potential theory to experiment result, it is obtained that the motion prediction by diffraction potential theory has an acceptable accuracy mostly, except for
heave motion when the wave frequency near to the structure natural frequency (Siow et al., 2013a and Siow et al., 2014a).

As presented in a previous paper, the diffraction potential theory is less accurate to predict the structure heave motion response when the wave frequency closer to structure natural frequency. At this situation, the heave response calculated by the diffraction potential theory will be overshooting compare to experiment result due to low damping executed by the theory and then follow by the large drop which give and underestimating result compare to experiment result before it is returned into normal tendency (Siow, et al., 2013a).

In order to correct the over-predicting phenomenon made by the diffraction potential theory, the previous research was trying to increase the damping coefficient by adding viscous damping into the motion equation (Siow et al., 2014a). From that study, the viscous damping is treated as extra matrix and added into the motion equation separately. This addition viscous damping was estimated based on the equation provided by Nallayarasu and Prasad (2012).

By adding the extra viscous damping into the motion equation, it can be obtained that the significant over-predicting of heave motion when wave frequency near to the floating structure natural frequency was corrected and it is close to the experimental result compared to executed result by diffraction potential theory alone (Siow et al., 2013a). However, the under-predicting of the heave response by diffraction potential theory in a certain wave frequency region still remaining unsolved by adding the viscous damping to the motion equation as discussed in the previous study (Siow et al., 2013a).

Siow, et al., (2014b) conducted the researches which focus on the effect drag force and viscous damping in estimate the semi-submersible heave response using diffraction potential theory. To able the numerical solution to calculate the extra drag force and viscous damping, they applied the drag term in Morison equation. Accuracy of the modification solution also checked with the previous semisubmersible experiment result which carried out at the towing tank belongs to UniversitiTeknologi Malaysia (Abyn, et al., 2012a and Siow, et al., 2014b). The experiment is conducted in head sea condition and slack mooring condition for wavelength around 1 meter to 9 meters. In the comparison, they obtained that the non-agreed heave response tendency near the structure natural frequency predicted by diffraction potential theory can be corrected by involving the drag effect in the calculation

### 3.0 PARTICULARS OF TWIN HULL FPSO

As mentioned, the twin hull FPSO construct by joining together two FPSO which the process facilities along with the storage and crew living quarters are located on deck. Figure 3.5 and Table 3.1 showed the description and principal particulars of twin hull FPSO that will be consider in this research.

![Figure 1: Description of Twin Hull FPSO](image)

<table>
<thead>
<tr>
<th>No</th>
<th>Description</th>
<th>Prototype</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Dead weight</td>
<td>≈ 250000</td>
<td>tonnes</td>
</tr>
<tr>
<td>2</td>
<td>Length</td>
<td>270.7</td>
<td>m</td>
</tr>
<tr>
<td>3</td>
<td>Breadth</td>
<td>44.3</td>
<td>m</td>
</tr>
<tr>
<td>4</td>
<td>Depth</td>
<td>21.7</td>
<td>m</td>
</tr>
<tr>
<td>5</td>
<td>Fully loaded draught</td>
<td>16.7</td>
<td>m</td>
</tr>
<tr>
<td>6</td>
<td>Spacing between hull</td>
<td>30</td>
<td>m</td>
</tr>
<tr>
<td>7</td>
<td>Diameter of turret</td>
<td>21.5</td>
<td>m</td>
</tr>
<tr>
<td>8</td>
<td>Type of mooring</td>
<td>CALM</td>
<td></td>
</tr>
</tbody>
</table>

### 4.0 MODEL EXPERIMENT APPROACH

In the experimental approach the first part deals with the model preparation. In model preparation some test will perform to check the natural period of the twin hull FPSO, to confirm the KG of the structure and others. Those tests are inclining test, swing test, decay test and spring calibration test. In another way it can be said that the model preparation is to validation of the loading of the structure. The model was ballasted with additional weights to achieve the 40%, 70% and 100% DWT respectively at different turret locations as shown in Figure 3.4. The second part deals with the experiment to analyze force oscillation test and the third part is wave excitation test to analyze the force and moment acting on the structure. The last part is motion test. Wave will be used for the test. From the experiment the dynamics force of twin hull FPSO as well as its mooring line force acting on the body motion has measured at corresponding wave frequency. Part of
The motion test has provided the value to find the RAO of FPSO. The experimental approach gives the following information:

- Model preparation to determine the particular of the twin hull FPSO.
- Force oscillations test to obtain the exciting force on structure in regular waves.
- Wave excitations test to obtain the force and moment acting on the structure.
- Motion test to obtain the RAO of the twin hull FPSO in regular waves.
- Analysis of the force and moment oscillations test as well as motion test will provide the hydrodynamics behavior.

The result obtained from experiment illustrated the behavior of the structure at every particular wave frequency.

![Position of turret at midship (a)]

![Position of turret at semi-forward of midship (b)]

![Position of turret at forward of midship (c)]

Figure 2: Different Positions of Turret Structure of an Internal Turret Moored Twin Hull FPSO System.

5.0 THEORETICAL STUDY OF TWIN HULL FPSO

5.1 Diffraction Theory

According to linear potential theory, the potential of a floating body is a superposition of the potentials due to the undisturbed incoming wave $\Phi_\infty$, the potential due to the diffraction of the undisturbed incoming wave on the fixed body $\Phi_d$, and the radiation potentials due to the six body motion $\Phi_j$, the total potential can be written by:

$$\Phi = \sum_{j=1}^{6} \Phi_j + \Phi_\infty + \Phi_d$$

Also, the potential must be satisfied with boundary condition below:

- **The continuity equation,**

  $$\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} + \frac{\partial^2 \phi}{\partial z^2} = 0$$

  (2)

  holds in fluid domain.

- **Sea bed boundary condition,**

  $$\frac{\partial \phi}{\partial z} = 0 \text{ at } z = -h_0$$

  (3)

  in which $h_0$ is the distance from the origin of the earth-bound coordinate system, $O(x, y, z)$, to the sea bed.

- **Free surface boundary condition,**

  At the mean free surface:

  $$\frac{\partial \phi}{\partial z} + \frac{\partial^2 \phi}{\partial t^2} = 0$$

  (4)
Kinematic boundary condition on the oscillating body surface, on the wetted part of its oscillating hull of the structure (in its mean position):

$$\frac{\partial \phi}{\partial n} = \vec{v} \cdot \vec{n}$$ (5)

in which $\vec{v}$ is the velocity of a point on the hull of the body and $\vec{n}$ is the normal vector of the hull, positive into the fluid.

1. The body motion and diffraction potenialls have to satisfy a radiation condition which states that at great distance from the body these potentials disappear.

In regular waves a linear potential $\Phi$, which is a function of the earth-fixed coordinates $x$, $y$, $z$ and of time $t$, can be written as a product

$$\Phi(x, y, z; t) = \Theta(x, y, z), e^{-i\omega t}$$ (6)

According to Tuck (1970), a convenient formulation can be obtained by writing:

$$\Theta = -i \omega \sum_{j=0}^{7} \Theta_j \zeta_j$$ (7)

in which $j = 0$ represents the undisturbed incoming wave, $j = 1, \ldots, 6$ are associated with each of the motion modes of the body and $j = 7$ represents the diffracted wave. The space-dependent part of the velocity potential $\Theta_j$, associated with an undisturbed long-crested regular wave in water of constant depth $h$ is given by:

$$\Theta_j = \frac{\xi_0}{\omega} \frac{\cosh(k h_0 + \mu h)}{\cosh k h} \cdot e^{i k (x \cos \mu + y \sin \mu)}$$ (8)

Where,

$$\xi_0 = \text{amplitude of undisturbed wave (m)}$$
$$k = \text{wave number at shallow water (rad/m)}$$
$$\mu = \text{wave direction (rad); zero for a wave travelling in the positive x-direction}$$
$$\omega = \text{wave frequency (rad/s)}$$
$$h_0 = \text{distance from the origin, O, of the earth-fixed axes to the sea bed (m)}$$
$$h = \text{water depth (m)}$$

The complex potential $\Theta$ follows from the superposition of the undisturbed wave potential $\Theta_0$, the wave differtential $\Theta_j$, and the potentials $\Theta_j$, associated with the $j$-modes of motion of the body ($j = 1, \ldots, 6$):

$$\Theta = -i \omega \left\{ \Theta_0 + \Theta_j \xi_0 + \sum_{j=1}^{6} \Theta_j \zeta_j \right\}$$ (9)

The fluid pressure follows from the Bernoulli equation:

$$p(x, y, z; t) = -\rho \frac{\partial \phi}{\partial t}$$
$$p(x, y, z; t) = \rho \omega^2 \left( \Theta_0 + \Theta_j \xi_0 + \sum_{j=1}^{6} \Theta_j \zeta_j \right), e^{-i\omega t}$$ (10)

The first order wave exciting forces ($k = 1, 2, 3$) and moments ($k = 4, 5, 6$) in the $k$th direction are:

$$X_k = -\Re \{ \iint_{S_0} \rho n_k \, dS_0 \} = -\rho a^2 \zeta_k e^{-i\omega t} \Re \{ \iint_{S_0} (\Theta_0 + \Theta_j) n_k \, dS_0 \}$$ (11)

and the oscillating hydrodynamic forces and moments in the $k$th direction are:

$$F_k = -\Re \{ \iint_{S_0} \rho n_k \, dS_0 \} = -\rho a^2 \sum_{j=1}^{6} \zeta_j e^{-i\omega t} \Re \{ \iint_{S_0} \Theta_j n_k \, dS_0 \}$$ (12)

In which,

$S_0 = \text{mean wetted surface of the body}$
$n_k = \text{direction cosine of surface element } dS_0 \text{ for the } k\text{-mode}$

The added mass and damping (coupling) coefficients are defined as follows:

$$a_{kj} = -\Re \left[ \rho \iint_{S_0} \Theta_j n_k \, dS_0 \right]$$ (13)
$$b_{kj} = -\Im \left[ \rho \omega \iint_{S_0} \Theta_j n_k \, dS_0 \right]$$ (14)

5.2 Morison Equation

Morison equation is a semi-empirical equation for the inline force on a body in oscillatory flow. The Morison equation is used to estimate the wave loads in the design of oil platforms and offshore structures.

The Morison equation is the sum of two force components (Morison et al., 1950) an inertia force in phase with the local flow acceleration and a drag force proportional to the (signed) square of the instantaneous flow velocity. The inertia force is of the functional form as found in potential flow theory, while the drag force has the form as found for a body placed in a steady flow.

$$F = F_I + F_D |u|$$ (15)
$$F = \rho C_m V \dot{u} + \frac{1}{2} \rho C_d |u|^2$$ (16)

where, $F$ is wave force, $F_I$ is inertia force, $F_D$ is drag force, $C_m$ is inertia coefficient, $C_d$s drag coefficient, $|u|$ and $u$ is water particle velocity normal to the structure, $u$ is particle acceleration normal to the structure, $\rho$ is seawater density and $V$ is volume of the structure.

The force coefficients $C_m$ and $C_d$ are dependent on Reenoyld, $Re$ number, roughness parameter, interaction parameter and Keulegen and Carpenter, $KC$ number. The inertia term of Morison equation was used to calculate the wave exciting force and moment on the semi-submersible as the drag force was assumed insignificant (Chakrabarti, 2007). The water particle kinematics was determined by the following equations:

Horizontal water particle velocity,

$$u = \frac{\pi h \cosh k h}{T} \frac{\sinh k h}{\sinh k h} \cos \theta$$ (17)
Horizontal water particle acceleration,

\[ \ddot{u} = \frac{2\pi^2 H \cos \phi}{T} \sin \theta \]

(18)

where, \( s = y + d, \theta = kx - \alpha \), \( k \) is wave number \((2\pi/L)\), \( T \) is wave period, \( y \) is the height of the point of evaluation of water particle kinematics, \( x \) is point of evaluation of water particle kinematics from the origin in the horizontal direction, \( t \) is time instant at which water particle kinematics is evaluated, \( L \) is wave length, \( H \) is wave height and \( d \) is water depth.

### 5.3 Drag Term of Morison Equation

The linear drag term due to the wave effect on floating structure is

\[ F_D = \frac{1}{2} \rho A_{proj} C_D \phi \left( \dot{X}_d - X_d \right) \]

(19)

where, \( \rho \) is fluid density, \( A_{proj} \) is projected area in \( Z \)-direction, \( C_D \) is drag coefficient in wave particular motion direction, \( \phi \) is particle velocity at \( Z \)-direction in complex form and \( X_d \) is structure velocity at \( Z \)-direction.

In order to simplify the calculation, the calculation is carried out based on the absolute velocity approach. The structure undergoes wave motion is ignored in the calculation because it is assumed that the fluid particular velocity is much higher compared to structure velocity. Expansion of the equation 4.35 is shown as follows:

\[ F_D = \frac{1}{2} \rho A_{proj} C_D \phi \left( \dot{X}_d - X_d \right) + \frac{1}{2} \rho A_{proj} C_D \dot{X}_d X_d - \frac{1}{2} \rho A_{proj} C_D \dot{X}_d \dot{X}_d - \frac{1}{2} \rho A_{proj} C_D \phi \left( \dot{X}_d - X_d \right) \]

(20)

By ignoring all the term consist of \( \phi \) in equation [4.36] can be reduced into following format:

\[ F_D = \frac{1}{2} \rho A_{proj} C_D \phi \left( \dot{X}_d - X_d \right) - \frac{1}{2} \rho A_{proj} C_D \phi \left( \dot{X}_d - X_d \right) \]

(21)

The above equation [4.37] is still highly nonlinear and is impossible to combine with the linear analysis based on diffusion potential theory. To able the drag force to join with the diffraction force calculated with diffraction potential theory, then linear drag term is then expanded in Fourier series. By using the Fourier series linearization method, equation [4.37] can be rewritten in the linear form as follow:

\[ F_D = \frac{1}{2} \rho A_{proj} C_D \frac{H}{\pi} V_{\text{max}} \phi \left( \dot{X}_d - X_d \right) - \frac{1}{2} \rho A_{proj} C_D \phi \frac{H}{\pi} V_{\text{max}} X_d \]

(22)

where, \( V_{\text{max}} \) is the magnitude of complex fluid velocity in \( Z \)-direction. From the equation [4.38], it can sumarize that the first term is linearize drag force due to wave and the second term is viscous damping force due to the drag effect.

According to Sjöbäck (2012), the linearize term the equation [4.38] is the standard result which can be obtained if the work of floating structure performance at resonance asassumed equal between nonlinear and linearized damping term. The linearized drag equation as shown in equation [4.38] nowcan be combined with the diffraction term which calculated by diffraction potential theory. The modified motion equation is shown as follows:

\[ \left( m + m_a \right) \ddot{X}_d + \left( \frac{1}{2} \rho A_{proj} C_D \frac{H}{\pi} V_{\text{max}} \phi \dot{X}_d \right) + kx = F_p + \frac{1}{2} \rho A_{proj} C_D \phi \frac{H}{\pi} V_{\text{max}} \phi \]

(23)

where, \( m \) is mass, \( k \) is restoring force, \( m_a \) is heave added mass, \( b_p \) is heave diffraction damping coefficient, \( F_p \) is heave diffraction force calculated from diffraction potential method, \( \frac{1}{2} \rho A_{proj} C_D \phi \frac{H}{\pi} V_{\text{max}} \phi \) is viscous damping and \( \frac{1}{2} \rho A_{proj} C_D \phi \frac{H}{\pi} V_{\text{max}} \phi \) is drag force based on drag term of Morison equation.

### 5.4 Equation of the Motion for the Whole System

The motion equation for a floating structure with attached stinger can be written, in the time domain, as:

\[ \ddot{\phi}(t) = \ddot{F}_h(t) + \ddot{F}_m(t) \]

(24)

where, \( \ddot{F}_h(t) \) is hydrodynamic forces on the vessel and \( \ddot{F}_m(t) \) is Morison load on mooring lines elements.

\[ \ddot{F}_m(t) = \ddot{F}_h + \ddot{F}_d \]

(25)

where, \( \ddot{F}_h(t) \) is diffraction force calculated from diffraction potential method and \( \ddot{F}_d(t) \) is drag force based on drag term of Morison equation.

The mass matrix of the total system is defined by \( M \). The accelerations \( \ddot{\phi} \) are solved in a time stepping procedure, and the motions \( \phi \) follow from integration. If the mooring line is rigidly connected to the vessel, the motion vector contains six rigid body motion components. The Morison load on slender (tubular) elements is given:

\[ \ddot{F}_m(t) = \frac{1}{2} \rho D C_m \ddot{u} \]

(26)

where, \( C_m \) is inertia coefficient, \( C_D \) is drag coefficient, \( \ddot{u} \) is particle velocity normal to the structure, \( \ddot{u} \) is particle acceleration normal to the structure, \( \rho \) is seawater density and \( D \) is diameter of the mooring line.

### 6.0 CONCLUSION

This chapter emphasized the formula and the mathematical that has been used in this research. The formula and mathematical theory has been modeled to analyze the output of the method approach.

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