Enhancement of Performance of Semi-Submersible Behaviour on Heave Motion by Adding Heave Plates

by

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14950

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Civil Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfillment of the requirements for the
BACHELOR OF ENGINEERING (Hons)

Approved by

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(Dr. Montasir Osman Ahmed Ali)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK

JAN 2015

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

HO MIN TZOONG

ABSTRACT

This paper presents the results from a study on the enhancement of the heave performance of a semi-submersible in deepwater by adding heave plates. Heave motion Response Amplitude Operators (RAO) of a conventional semi-submersibles is first obtained through the construction of transfer functions and validated in the current study. The effects of the heave plates in terms of elevation, sizes and shapes on the heave motion RAO are then presented and discussed. The study reveals that the elevation of the heave plates has a significant impact on the heave performance of the semi-submersible. Parameters such as sizes and shapes also affect the effectiveness of the heave plate as an additional mechanical damping devices. Based on these results, recommendations are then made on how to enhance the heave performance of the semi-submersible. Heave responses of the semi-submersible with the added heave plates is later compared with the conventional semi-submersible. The comparison result indicates that with the added heave plates, there is a considerable improvement in the heave performance of the semi-submersible.

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CHAPTER 1
INTRODUCTION

1.1 Background Study

In water depth greater than 120 metres, floating vessel is a better alternative compare to fixed structures for offshore drilling operations as they are seemed more practical and feasible in that environment. In Books LLC (2010), it stated that in the early 1950s, Monohull ships such as CUSS I were used for offshore drilling activities in deepwater. However, it was found out that these ships would experience a significant heave, pitch and yaw motions in large waves and thus, the oil and gas industry required more stable drilling platforms in order to venture into deepwater oil drilling.

Later on, semi-submersible concept for the drilling industry mobile offshore fleet was introduced and it garnered much attention from the industry since the semi-submersible is less affected by the wave loadings compare to a normal ship. Since then, the number of semi-submersible operates in offshore drilling increased greatly. A semi-submersible consists of topside/deck, vertical columns and pontoons. The vertical columns connect the pontoons and the topside. It gains its buoyancy from ballasted and waterproof pontoons placed below the ocean surface and wave action.

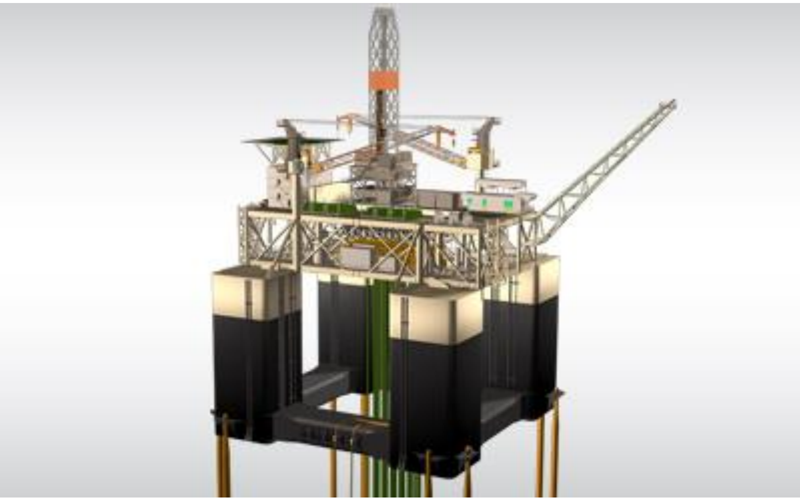


Figure 1: Configuration of Semi-Submersibles

Chen, Mei and Mills (2007) indicated that since semi-submersible has a large footprint and low centre of gravity, it has a good stability and hence, it can ensure its topside to be located high above the sea level and kept well away from the waves. Semi-submersible also can hold an abundance amount of flexible risers or steel catenary risers (SCRs) due to its availability of spaces on the pontoons. Besides that, the design of semi-submersible helps in decreasing the costs as well as the scheduling time with its quayside topsides integration. Its initial investment is also reasonably low. Moreover, a semi-submersible vessel is capable of changing from a deep to a shallow draft by taking out the ballast water from the hull.

Nevertheless, semi-submersible has a few flaws on its design if it is compared with the spar or tension leg platform (TLP). The most significant one is that it has a higher heave responses due to its shallower draft. Due to this flaw, it is unsuitable for a dry tree riser arrangement and it mostly only used as a short-term drilling vessel or a permanent wet-tree application production platform. In the dry tree riser arrangement, the tree is placed above the water by using tensioning devices such as mechanical tensioners or supporting buoyancy cans. This dry tree riser system has a huge benefit in terms of costs effectiveness for the well completion, drilling/workover and intervention during the lifespan in the offshore production facility as discussed by (Chen, Mills & Mei, 2007). Furthermore, the large heave motion will also increase fatigue damage on the SCRs and thus, more fatigue resistance design of SCRs, which has a higher cost, is required.

The interface between the vessels and the risers is one of the most the most essential conditions for floating production system and in order to for the semi-submersible to become a feasible dry-tree floating solution for deepwater development, its heave motion has to be greatly reduced. The ability in addressing this issue will surely result in a revolutionary advances in the offshore development field.

1.2 Problem Statement

As the oil and gas industry ventured into the deep frontier in order to tap the remaining oil and gas in the world, the need for an effective and efficient vessel without any water depth limitation has become more acute. The keys for an efficient and effective vessel lies on the types of riser system it used on the platform. Nowadays, dry tree riser is preferable in the industry due to its significant economic advantage and high efficiency. Based on current technology, dry tree riser is only restricted to spars and tension leg platform due to its design sensitivity to vessel motions. Significant heave motion will make it impossible for the risers to be tensioned through the connections to the deck or buoyancy cans. Therefore, a conventional semi-submersible is not opt to become a viable vessel for dry tree riser in the deepwater field, unless its heave response can be greatly reduced. One of the approaches in reducing the heave motion is by adding flat plates to the semi-submersible. These plates will act as heave inertia-driven damper which will add damping to the semi-submersible and consequently, reducing its heave motion under wave loads. Nevertheless, depending on the location or size of heave plates used, they may result in different effects towards the heave performance of the semi-submersible.

1.3 Objectives

The primary objective of this study is to enhance the heave performance of a semi-submersible by adding heave plates. In order to achieve this goal, some secondary objectives have to be defined and set clear. These objectives are:

1. To conduct numerical modelling on the heave response of a conventional semi-submersible by using a software.
2. To determine the effect of heaves plate on the heave response of the semi-submersible in terms of its elevation, size and shape.

1.4 Scope of Study

This scope of study is constricted within the following boundaries:

i. The semi-submersible is subjected to regular waves and unidirectional.
ii. The study is limited to head sea condition (wave-heading at 180°)
iii. The hydrodynamic analysis of the study is in term of frequency domain analysis only.
iv. Current and wind effects are neglected in this study.

CHAPTER 2
LITERATURE REVIEW

2.1 Chapter Overview

This chapter will discuss on the past research with a correlation to the dynamic analysis of floating structure especially semi-submersible. First, the origin concept of semi-submersible is introduced. Next, studies related to motion analysis of a semi-submersible is presented. This is then followed by the discussion on the design concept of semi-submersible with heave plates. Finally, a critical review on the related research topics is presented.

2.2 Evolution of Semi-Submersible

The first development of the semi-submersible is in 1962 by the Bruce Collip of Shell. Originally, semi-submersibles were used for drilling operations, but, they were also used for other momentary tasks such as accommodation and installation. Lim and Ronald (2000) cited that the benefits of semi-submersibles only became obvious in the late 1980s and early 1990s and thus, there were a lot of research ongoing that time in order to convert semi-submersibles to production for deepwater field. A scrutiny on the evolution of the semi-submersible can be done based on the new riser types, construction methods, hull configurations and the increasing production rates. (Refer Figure 1 for the production rates of semi-submersible per year)

Argyll FPF was the first semi-submersible floating production platform which was converted from the Transworld 58 drilling semi-submersible in 1975 for the Hamilton Brothers North Sea Argyll oil field. Whereas, the first purpose-built production semi-submersible platform was only built in 1986 for the Balmoral field, UK North Sea. Most of the production semi-submersibles, approximately 85%, were converted either from different operation such as drilling and accommodation or from another field location (reuse) up until 1994 onwards, only there was more demand for the new-build production semi-submersibles (Lim & Ronald, 2000). It was found out that a new-build semi-submersible may be the better choice for major operations especially in a harsh environments due to its longer service life and lower maintenance costs as well as the greater flexibility for the operator to design its hull and topsides for a certain field. At present, there are only eight purpose-build semi-submersibles operating in the world and have reached the sixth generation. They can be differentiated accordingly to their age, deck load, water depth capacity and environmental rating.

In both Brazil and North Sea, production semi-submersibles have been recognized as a popular solution for deepwater development throughout the first 25 years of their service. With this, it is proven that the semi-submersibles have advantages in niche areas. The semi-submersibles can still be further revolutionize until there is a possibility that they might become an economic alternative to the spar in deepwater, especially in the Gulf of Mexico.

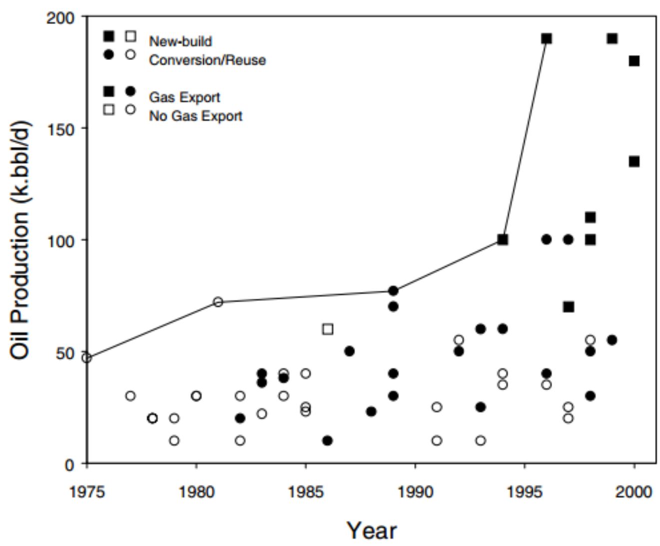


Figure 2: Production Rates of Semi-Submersibles per year [Lim & Ronald (2000)]

2.3 The Motion Analysis of Semi-Submersible

Floating structures such as semi-submersible have six degrees of freedom of motions which basically are in linear (surge, heave, sway) and angular (pitch, yaw, roll) form. Other than the environmental effects, the motion responses in the semi-submersible are governed by its own mass properties and geometric parameters.

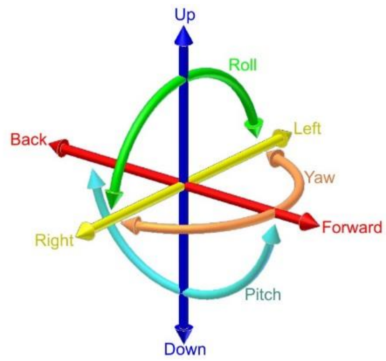


Figure 3: Six Degrees of Freedom

The most important procedure in designing offshore structure is the hydrodynamic analysis of the structure at the defined environment. Chakrabarti (1987) even claimed that the hydrodynamic force calculation for design is a very complicated task as it involves an interaction between waves and structure at a complex environmental conditions. It has always been the designers' preference for them to scrutinise the environmental forces and resulting motion of the offshore structures under regular sea conditions despite the fact that the ocean waves are of a random nature. Chakrabarti (1987) considered this method as design wave approaches and it is based on three parameters which are the period (T), the water depth (d) and the height of wave (H).

Wave theories and wave force formulations are applied in the design wave approach in order to determine the wave kinematics and dynamics and subsequently, the wave force acted on the offshore structure. Chakrabarti (1987) further discussed that the suitability of the application of these wave theories on different type of region; shallow, intermediate or deep can be determined based on their analytical validity and experimental validity.

Theory	References	Exactly satisfies			
		DE	BBC	KFSBC	DFSBC
Linear wave theory	(Wiegel, 1964; 1964; Sarpkaya and Isaacson, 1981)	X	X	—	—
Third-order Stokes	(Skjelbreia and Hendrickson, 1961)	X	X	—	—
Fifth-order Stokes	(Skjelbreia and Hendrickson, 1961)	X	X	—	—
First-order cnoidal	(Laitone, 1960; Wiegel, 1960)	—	X	—	—
Second-order cnoidal	(Laitone, 1960)	—	X	—	—
Stream function numerical wave theory	(Dean, 1965)	X	X	X	—

DE = Differential Equation
BBC = Bottom Boundary Condition
KFSBC = Kinematic Free Surface Boundary Condition
DFSBC = Dynamic Free Surface Boundary Condition

Figure 4: Boundary Value Problem of Water Wave Theories [Dean (1968)]

The different type of boundary conditions that can be satisfied by which wave theories are shown in Table 3 by Dean (1968). Most of the theories fail to satisfy the nonlinear boundary conditions at the free surface expect for the stream function theory. The rest of the theories can only satisfy the differential equation and the bottom boundary condition. Dean (1968) performed an analysis to check how well these wave theories satisfy the two free surface boundary conditions; kinematic and dynamic at different regions of the non-dimensional wave parameters. Based on the results, it is shown that the Airy and Cnoidal (first-order) theories are more suitable in shallow water whereas the Stokes nonlinear theory is more suitable in deeper water. According to the analytic validity, the stream function theory is the most applicable throughout the entire range of d/T^2 values as shown in Figure 4.

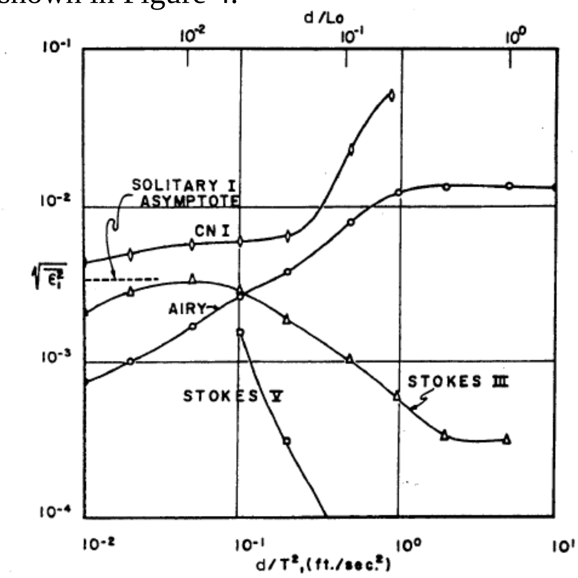


Figure 5: Dimensionless error $\sqrt{\overline{\varepsilon_1^2}}$ in kinematic free surface boundary condition, H/H_b = 0.25; all wave theories. [Dean (1968)]

Besides that, Chakrabarti et.al (2007) also performed an analysis on a truss pontoon semi-submersible concept in deepwater by using linear diffraction theory and also linear part of Morrison Equation. The results that he obtained from those methods are then compared with the experimental result. Based on that, he concurred that both computation method bear almost similar results as the experimental, except when the wave periods are at higher level at which the Morrison Equation will shows a slight difference; approximately 10% higher.

Based on the results obtained from the design wave approach, motion analysis of the offshore structure is then able to be conducted to find the structure-response such as the motion vector, velocity vector and acceleration vector of the structure. The structure-response is usually expressed in terms of Response Amplitude Order (RAO) or spectrum and it is one of the most important criteria in determining the level of efficiency of a floating vessel in its operating state. Hassan, Jaswar and Siow (2013) cited in their journal that low level of vertical plane motions induced by heave, roll and pitch is an important requirement for a unit to have a good drilling capabilities. This is due to the fact that large heave motion will only increase the total costs and cause wastage in terms of drilling time. Moreover, it will also compromise the safety of the risers and umbilical pipes on the structures as stated by Zhang and Li (2009) in their paper. Hence, the ongoing research and studies on the methods in reducing the heave motion of the floating structures like semi-submersibles gained much attention in the oil and gas industry.

In short, effective and efficient interaction between the motion responses of floating structures and surrounding fluid is an essential requirement to engineering design as it affects the workability, time and total costs.

2.4 Design Concept of Semi-Submersible with Heave Plates

Basically, the enhancement of the natural period in heave motion can simply be done by increasing the mass of the structures, for example, adding heave plates onto the structures. This concept is well-known in the construction of spar platform. Rho, Choi and Lee (2002) had experimented scaled models of spar platform with/without damping plate in a wave tank and demonstrated that the spar platform with damping plate has shown better performance than standard spar in heave motion at resonance.

Halkyard et.al (2002) presented a new concept of semi-submersible known as DPS 2001. The concept is based on integration system of a semi-submersible and a truss spar. The upper hull is made up of a conventional semi-submersible whereas the lower hull consists of a heave plate and a truss that provides support to the plate. The lower truss/heave plate can be retracted during fabrication and transportation and lowered during operation mode. This concept is similar to the T- semi concept which stated by Yu, Chen and Cui (2013) in their paper. However, for T- semi concept, it involved a multiple of heave plates within the lower truss. The concept that proposed by Chakrabarti et.al (2007) also utilizes multiple heave plates in its design. The concept which recognized as Truss Pontoon Semi-Submersible (TPS) is also a hybrid of a semi-submersible and a spar and rather than a single lower truss under the hull, TPS has four truss columns. At the bottom of each columns, there are heave plates attached to it.

The study conducted by Halkyard is then further improved by Zhu, Ou and Zhai (2011) as they introduced a conceptual design of a semi-submersible with a moveable heave plate. This concept is known as MHS platform and it is based on a turned mass damping (TMD) system which consisted of elastic connectors and dampers between the hull and heave plate. From the findings, the heave performance of MHS platform is better than DPS 2001 due to the TMD system. However, it is still a conceptual design and hence, more time is still needed for them to conduct studies in order to access other problems posed by the conceptual design such as its fatigue and strength.

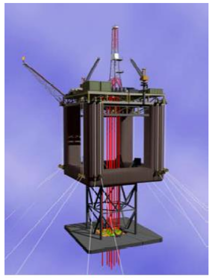
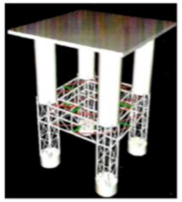
Besides T-semi concept, there is another concept, named E-semi which also depends on the hydrodynamic interaction between the hull and heave plate in order to reduce the motion (Yu, Chen & Cui, 2013). For E-semi concept, there is only one extendable heave plate located under the hull. Xie, Xie and Jiang (2012) have also introduced a new concept known as Deepwater Tumbler Platform (DTP) which is almost similar to E-semi concept in terms of its design configuration. The only difference is that the heave plate in DTP is placed within a lower tier pontoon (LTP) which connected to the hull by four telescopic columns.

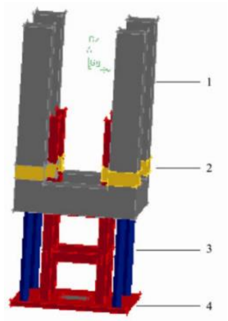
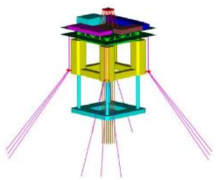
Chen, Mei and Mills (2007) have also performed parametric studies on the effect of elevation and dimension of heaves plates on the semi-submersible heave response. From the study, the heave motion increases when a square type heave plate is placed near or above the heel elevation. Only when the same heave plate is placed at an elevation about 60 metres below or more below the semi keel, the heave motion decreases considerably. Nevertheless, this method is deemed as unpractical due to the limitations in the design of semi-submersible. Chen, Mei and Mills (2007) claimed that in order for the semi-submersibles to be able to carry the heave plates at any elevation, a new structural component is necessary to be attached to the semi-submersibles for supporting the plates' weight.

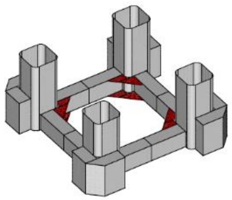
On the other hand, Haslum and Faltinsen (1999) suggested that an increasing system damping can decrease the heave response to wave frequency. For instance, Tao, Lim and Thiagarajan (2004) made a modification on the hull shapes with larger damping in consideration of the suppression of heave resonant response. Based on the study done by Srinivasan, Chakrabarti and Radha (2005), the concept of both hydrodynamic mass and separated-flow damping are used to control the heave motion of a large floating platform designed by themselves.

Instead of focusing on the method of increasing the added mass and system damping to reduce the heave motion, Zhang and Li (2009) declared that the natural period and heave are mainly dependent on the volumetric ratio of pontoon compare to the specific geometric configuration. The added mass coefficient and damping coefficient also do not bring any significant effect on the heave response. Taking into consideration the fact that both volumetric ratio of pontoon and additional mass can affect the motion characteristics of a semi-submersibles, Kyoung et.al (2013) developed a concept of dry tree semi-submersibles with the application of low heave motion and vortex induced motion (VIM) response. The main features of this concept is that it has column step that allows the displacement from the pontoon to be redistributed to the column. In addition, pontoon plates are also placed at the corner of the pontoon and column in order to provide an additional mass as well as structural rigidity to the lower hull. It was seen that the hull motion performance really increase with the paired pontoon plates without any major changes on the hull design and fabrication cost

2.5 Critical Review

No.	Author	Year	Title	Methodology	Findings	Remark
1  Fig 1 *DPS 2001-4*	Halkyard. J, Chao. J, Abbott. P, Dagleish. J, Banon. H & Thiagarajan . K.	2002	A Deep Draft Semisubmersible with a Retractable Heave Plate	Heave plate (200’ X 200’) was located at the lower hull and supported by truss.	Conceptually, DPS 2001-4 is able to accommodate the dry tree riser due to its lower heave motion compare to a conventional semi-submersible.	This conceptual design is still new and thus, the effect of heave plates is not studied thoroughly.
2  Truss pontoon semi-submersible platform (TSP)—prototype. 	Chakrabarti. S, Barnett. J, Kanchi. H, Mehta. A & Yim. J.	2007	Design Analysis of a Truss Pontoon Semi-Submersible Concept in Deep Water	Truss Pontoon Semi-Submersible (TPS) is a hybrid of conventional semi-submersible and a spar with an addition of heave plates at the bottom of the truss columns. The total number of heave plates used are 8 with a dimension of 1458.3 sq ft. The excitation forces is then calculated using wave diffraction theory and Morrison Equation. The motion responses obtained is compared with the results from wave tank model test (1: 50).	Based on the comparison of the two methods, they all yield almost the same results as the experimental result. However, at higher wave periods, Morrison equation results are slight higher than the other methods by 10%.	Morrison Equation can be used to analyse TPS system without affecting much of the results quality.

3 	Chen.C.Y., Mei. X. & Mills.T.	2007	Effects of Heave Plate on Semisubmersible Response	A various dimensions of heave plates (100’ X 100’ , 173’ X 173’ and 240’ X 240’) is added to a deep draft semi-submersible at various elevations (20’, 0’, -20’, -60’, -100’, -200’ and -300’ with regard to the keel).	Different location or draft of heave plates will result a different effectiveness. An increase in the heave plate size will also further reduce the heave motion.	The heave plate size used for the study is too big and unfeasible.
4 	Zhu.H, Ou. J & Zhai. G.	2011	Conceptual Design of a Deep Draft Semi-Submersible Platform with a Moveable Heave-Plate	A concept that based on turned mass damping (TMD) known as Moveable Heave Plate Semi-Submersible (MHS). The connectors and dampers located between the hull and heave plates are elastic in heave motion but rigid in other motion. The motion responses obtained is then compared with the deep draft semi-submersibles with retractable heave plate (DPS).	MHS platform is proven to have a smaller heave motion compare to DPS platform due to TMD system that help to decrease the hull heave motion.	This study is only a conceptual design and thus, there are still many problems concerning the structures such as its strength, fatigue and etc.
5 	Xie.B, Xie. W & Jiang. Z.	2012	A New Concept of a Deepwater Tumbler Platform	Deepwater Tumbler Platform (DTP) is based on a concept of deep draft semi-submersible hull with a lower tier pontoon (LTP) connected by four telescopic columns. Within the LTP, there is a large heave plate with a dimension of 31.2 m X 31.2 m.	The heave RAOs of DTP are lower compare to conventional semi-submersible for fatigue wave periods of 15 s and less. Meanwhile, its heave RAOs are as same	It is only based on numerical modelling and thus, there is no experimental results for DTP. Economic analysis in fabrication and

					level as spar in survival condition.	installation stages should be performed.
6 	Kyoung.J, O'Sullivan. J, Kim. J.W. and Lambrakos. K.	2013	Dry Tree Semisubmersible Application with Low Heave Motion and Vortex Induced Motion (VIM) Response	Heave and VIM Suppressed (HVS) semi-submersible has a feature of a column step and a narrow pontoon. At each of the junction of the pontoons and column, there is a triangle pontoon plate with a width of 12 m. Hydrodynamic analysis is done by using MLTSIM and Computational Fluid Dynamic (CFD) also used to optimize the design.	Without any major changes to the design and fabrication cost, the paired pontoon plates provide additional hydrodynamic mass, damping and structure rigidity which reduces the heave motion in the body system.	The location and geometry shape of the pontoon plate can still be optimized for a better performance of hull motion.

All of the discussed literature approached the heave motion problem in semi-submersible through the method of increasing the heave natural periods and system damping by introducing additional plates to the structure and from their findings, it is proven that the motion characteristics of a conventional semi-submersibles is significantly improved with the use of heave plates. However, none of the literature has conducted a thorough study on the effects of the heaves plate in terms of its shape toward the heave performance of the semi-submersible. Therefore, besides the elevation and sizes of the heave plates, the effect of different shapes of heave plates on the heave performance of the semi-submersible is also determined and discussed in this study. This is the gap of this study compared to those previous studies.

CHAPTER 3
METHODOLOGY

3.1 Design Parameter

Since this study is based on numerical modelling, the dimensions of a conventional semi-submersible and the environmental data for this study should be obtained from one of the literatures. This is to ensure the end result of this study can be validated by comparing it with the result from the literature. Below are the dimensions and environmental used for this study:

Table 1: Conventional Semi-submersible Dimensions and Its Loadings

Parameter	Unit	Value
Water Depth	m	1710
Topsides Weight	ton	20,000
Square Column size	m x m	16.2 x 16.2
Column Length	m	45.72
Pontoon Size	m x m x m	60.5 x 18.9 x 6.1
Draft	m	30.48
Freeboard	m	15.24

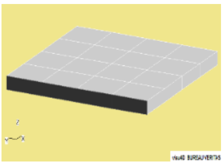
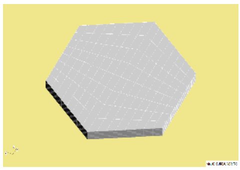
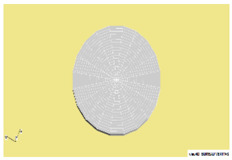
Table 2: 100-yr Extreme Environments in Gulf of Mexico

100- year Environment	Wind at 10m (m/s)	Significant Wave Height /Hs (m)	Peak Period /T_p (s)	Maximum Wave Height / H_{max} (m)	Current at surface (m/s)
Hurricane Max. Wax Condition	40.3	13.8	14.2	23.8	1.1
Hurricane Max. Wind Condition	42.1	13.3	13.9	22.9	1.1
Loop Current Condition	9.8	2.9	8.4	4.9	2.2

3.2 Design Parameters of the Heave Plates

The design parameter of the heave plates that will be used for this study is shown in the Table 3 below. The heave plates are attached on each column of the semi-submersible.

Table 3: Design Parameters of the Heave Plates

Parameter	 Square Plate	 Hexagonal Plate	 Cylinder Plate
Size (m)	22	$d_1 = 29, d_2 = 33.5$	29
	26	$d_1 = 30, d_2 = 34.6$	30
	29	$d_1 = 31, d_2 = 35.8$	31
Thickness (m)	1.2	1.2	1.2
Area (m^2)	484	728.33	661
	676	779.43	706.86
	841	832.25	754.77
Volume (m^3)	580.80	873.996	793.20
	811.20	953.32	848.23
	1009.20	998.70	905.72
Mass (kg)	4646400	6989306	6340991
	6489600	74822469.49	6785840.13
	8073600	7989603.97	7245769.30

Elevation (m)	28, 24, 20, 0, -30, -60 (measured from the semi keel)

3.3 Application of Wave Theories

The common wave theories that are used in the design of offshore structure are Linear Airy wave theory, Stokes second- and third-order theory, Stokes fifth-order wave theory, cnoidal theory and stream function theory. Chakrabarti (1987) stated that the validity of these wave theories in two different areas; analytical and experimental can determine the ranges of suitability of the application of these theories.

The chart in the Figure 3.1 is obtained through the studies by Dean (1968) and LeMehaute (1970). Since these wave theories depend on the three-dimensional parameters, d, H, and T, the regions can be described in terms of H/T^2 and d/T^2.

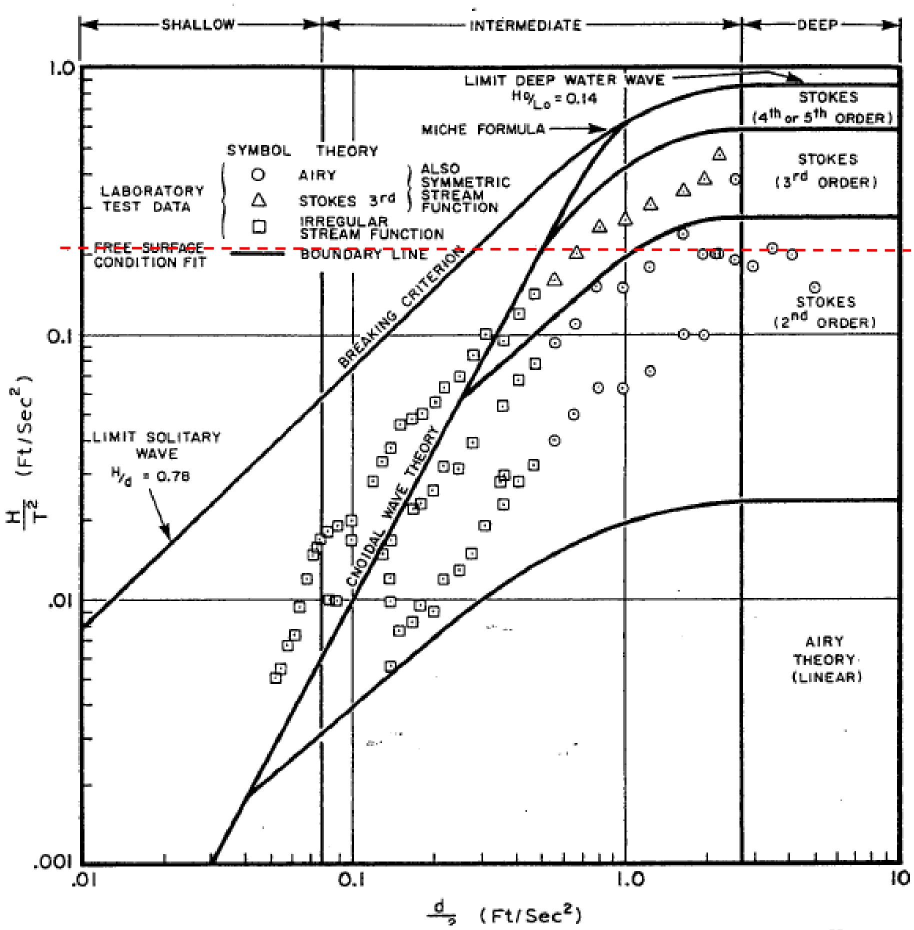


Figure 6: The Region of Application of the Various Wave Theories in terms of $\frac{H}{T^2}$ *and* $\frac{d}{T^2}$*.*
[Dean (1968) and LeMehaute (1970)]

Based on the Gulf of Mexico waves

H = 45.2 ft, T = 14.2 s , d = 5610 ft

$\frac{H}{T^2} = \frac{45.2}{14.2^2}$ $\frac{d}{T^2} = \frac{5610}{14.2^2}$

$= 0.22$ $= 27.82$

Based on the chart, the applicable wave theory is the Stokes 2nd Order. However, linear airy wave theory is found to be valid as well for this study according to the experimental data obtained from Chakrabarti (1980). The experimental data superimposed on the plot has shown that the linear theory is applicable beyond its analytic validity. Since HydroSTAR is based on first- and second- wave diffraction-radiation theory in which the linear air wave theory is used, it is concluded that linear airy wave theory is more suitable to be applied and used for determining the wave kinematics in this study.

3.3.1 Linear Airy Wave Theory

Linear Airy Wave Theory is a wave theory derived from the assumption that the wave height is small compared to the wave length or water depth which allows the linearization of the free surface boundary conditions by excluding any wave height terms that are beyond the first order. Instead at the oscillating free surface, this assumption permits the free surface conditions to be fulfilled at the mean water level. A rough and fast estimation of wave characteristics and their effects can be obtained through the linear theory. This approximation has a high accuracy for small ratios of the wave height to water depth (shallow water) and wave height to wavelength (deepwater). Moreover, the estimation of the several second order-nonlinear properties of surface gravity waves and their propagation can be done from their results.

$$k = 2\pi/L \quad (1)$$

$$\omega = 2\pi/T \quad (2)$$

$$\theta = kx - \omega t \quad (3)$$

The basic equations in the linear airy theory are as shown. The wave number (k), wave frequency (ω) and phase angle (θ) are found through Equation (1), (2) and (3) respectively. These values are used in Equation (3), (4), (5), (6), (7), (8) and (9) in order to find the horizontal particle velocity (u), vertical particle velocity (v), horizontal particle acceleration ($\dot{u}$), vertical particle acceleration ($\dot{v}$), horizontal displacement (ξ) and vertical displacement (η) respectively.

$$u = \frac{\partial \phi}{\partial x} = \frac{\pi H}{T} \frac{\cosh ks}{\sinh kd} \cos \theta \tag{4}$$

$$v = \frac{\partial \phi}{\partial y} = \frac{\pi H}{T} \frac{\sinh ks}{\sinh kd} \sin \theta \tag{5}$$

$$\dot{u} = \frac{\partial u}{\partial t} = \frac{2\pi^2 H}{T^2} \frac{\cosh ks}{\sinh kd} \sin \theta \tag{6}$$

$$\dot{v} = \frac{\partial v}{\partial t} = -\frac{2\pi^2 H}{T^2} \frac{\sinh ks}{\sinh kd} \cos \theta \tag{7}$$

$$\xi = -\frac{H}{2} \frac{\cosh ks}{\sinh kd} \sin \theta \tag{8}$$

$$\eta = \frac{H}{2} \frac{\sinh ks}{\sinh kd} \cos \theta \tag{9}$$

3.4 Wave Force Computation

One of the most basic tasks in the design of the offshore structure is the computation of the water wave forces on the structure and yet it is also one of the most complicated task since it involves the complexity of the interaction of waves with the structure. The computation of the wave forces on offshore structures can be done in three different ways:

I. Morison Equation
II. Froude-Kyrlov Theory
III. Diffraction Theory

In this study, only the diffraction theory will be discussed since the software used to conduct the numerical modelling (HydroSTAR) is based on first- and second-wave diffraction- radiation potential theory.

3.4.1 Diffraction and Radiation Potential Theory

The diffraction and radiation potential theory is applicable in the wave-force computations when the structure's size is comparable to the wave length. The velocity potential can be used to describe the regular wave acting on floating bodies and it is usually written in respective to the flow direction and time as shown below:

$$\Phi\ (x, y, z) = \text{Re}\ [\mathit{\Phi}\ (x, y, z,)e^{iwt}\ ] \quad (10)$$

$$\mathit{\Phi}\ (x, y, z) = \frac{g\varsigma_a}{iw}\ \{\Phi_o(x, y, z) + \Phi_7(x, y, z)\} + \Sigma_{j=1}^{6}\ iwX_j\Phi_j(x, y, z) \quad (11)$$

Where,

g	: Gravity acceleration (9.81 m/s^2)	ς_a	: Incident wave amplitude
X_j	: Motions amplitude	Φ_o	: Incident wave potential
Φ_7	: Scattering wave potential	j	: Direction of motion
Φ_j	: Radiation wave potential due to motions		

As shown in the above equation, the total wave potential, $\mathit{\Phi}$ in the system is sum of potential of the incident wave, Φ_o, scattering wave, Φ_7 and radiation wave, Φ_j. An assumption of the same phase and amplitude for both the incident wave and scattering wave is made. Nonetheless, radiation wave potentials are influenced by each type of the motion responses of a floating structure since the total potential for radiation wave for the single body is the summation of the radiation wave generated by each type of body motions such as roll, pitch, yaw, surge, sway and heave.

The wave potential must be satisfied with boundary conditions as below:

$$\tilde{V}^2\Phi = 0 \quad (12)$$

$$\frac{\partial\Phi}{\partial z} + k\ \Phi \qquad \text{at } z = 0\ (k = \frac{w^2}{g}\ ) \quad (13)$$

$$\frac{\partial\Phi}{\partial z} = 0 \qquad \text{at } z = 0 \quad (14)$$

$$\Phi \sim \frac{1}{\sqrt{r}}\ e^{-ik_0 r} \text{ should be 0 if r } \infty \quad (15)$$

$$\frac{\partial\Phi_7}{\partial n} = -\ \frac{\partial\Phi_o}{\partial n} \text{ on the body boundary} \quad (16)$$

3.5 Motion Computation

The following equation is used to describe the motions of floating bodies and it is derived from the Newton's Second Law.

$$[M_{ij} + A_{ij}] \ddot{x} + [B_{ij} + B_v] \dot{x} + K x = F (t) \quad (17)$$

Where,

M_{ij} : Inertia matrix of the body
A_{ij} : Added inertia matrix of the body
B_{ij} : Damping Matrix
B_v : Viscous damping
K : Stiffness Matrix
x : Motion vector of the body
$\ddot{x}$: Acceleration vector of the body
$\dot{x}$: Velocity vector of the body

F (t) : Excitation wave force coming from the diffraction problem solution

Once the motion vector of the body (x) is solved, the Response Amplitude Operator (RAO) transfer function can be defined through frequency domain analysis and thus, expressing RAO as:

$$\text{RAO} = \frac{x}{\varsigma_a} \quad (18)$$

3.6 Model Configuration

In order to proceed with the motion computation, an input of centre of gravity (CG) was required as well as the radius gyration in x, y and z axis. Moreover, the draft of the model was designed within the acceptable range during operating state. Based on PTS 34.19.10.30, the minimum air gap should be 1.5 m and provision of 0.5 m should be made for seabed subsidence. The air gap is the distance between the underside of the lower part of the cellar deck and the maximum extreme storm case crest elevation.

The CG of the semi-submersible was calculated using Equation (18):

$$\bar{x} = \frac{\sum_{j=1}^{N} m_n x_n}{\sum_{j=1}^{N} m_n} \quad (18)$$

Where, x_n = distance of CG of section n from the reference point and m_n = mass of steel at section n

As for the radius of gyration, it was calculated using the equation below:

$$k = \sqrt{\frac{I}{M}} \quad (19)$$

Where, I = total mass moment of inertia and M = total mass of structure

Each geometries has their own specific way to calculate their moment of inertia as they differ from one another in terms of centroid in respect to the rotary axis. Below are the equations used to calculate the mass moment of inertia in this study:

I. Rectangular

$$I_x = \frac{1}{12} m(y^2 + z^2) + me^2 \quad (20)$$

$$I_y = \frac{1}{12} m(x^2 + z^2) + me^2 \quad (21)$$

$$I_z = \frac{1}{12} m(x^2 + y^2) + me^2 \quad (22)$$

Where, m = mass of plate, e = eccentricity in respect of centre of gravity and x,y and z are the length of the plate in the respective direction

II. Thin Circular Disk

$$I_x = I_y = \frac{1}{4} mr^2 + me^2 \quad (23)$$

$$I_z = \frac{1}{2} mr^2 + me^2 \quad (24)$$

Where, r = radius of plate

III. Hexagon

$$I_{hexagon} = I_{rectangular} - 4 * (I_{regular\ triangular\ prism})$$

$$I_x = [\frac{1}{12} m\left(y^2 + z^2\right) - 4 * [\frac{1}{24} m\left(A^2 + 2z^2\right)]] + me^2 \quad (25)$$

$$I_y = [\frac{1}{12} m(x^2 + z^2) - 4 * [\frac{1}{24} m(A^2 + 2z^2)]] + me^2 \quad (26)$$

$$I_z = [\frac{1}{12} m(x^2 + y^2) - 4 * \left(\frac{mA^2}{12}\right)] + me^2 \quad (27)$$

Where, A = shorter side length of the regular triangular prism

For the model to be deemed as stable, it has to satisfy the following conditions:

i. The weight of the unit must be equal to the weight of liquid displaced

ii. 'B' and 'G' must be in the same vertical line.

iii. 'G' must be below 'M'

iv. GM > 0

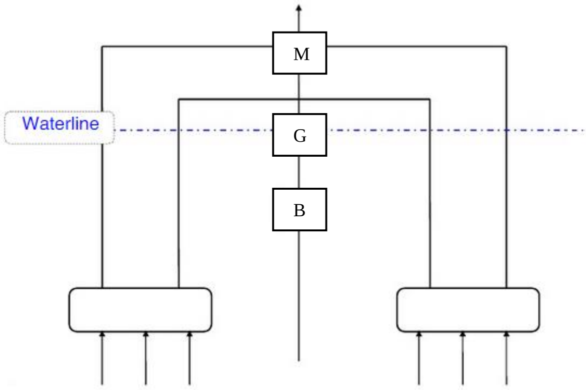


Figure 7: G, M and B of the Semi-Submersible

Where, M = metacentre, G = centre of gravity and B = centre of buoyancy

In addition, the stiffness of the model the total stiffness is consisted of the hydrostatic stiffness and mooring line stiffness. The hydrostatic was calculated by HydroSTAR whereas the mooring line stiffness was expressed in the matrix form shown as below:

$$K = \begin{bmatrix} k_x & 0 & -k_x\delta \\ 0 & \rho g A_s & \rho g \Delta GM \\ -k_x\delta & 0 & k_x\delta^2 \end{bmatrix} \tag{28}$$

Where, k_x = mooring line stiffness, δ = distance of CG from fairleads, ρ = density of seawater, A_s = water plane area, Δ = water displacement by volume and GM = metacentric height.

3.7 Numerical Modelling

With the development in computational technology, numerical modelling now can be done by using a software called as HydroSTAR. HydroSTAR is an incredible 3D diffraction/radiation potential theory 3D panel software for wave-body interactions that includes multi-body interaction, effects of forward speed and dynamic effects of liquid motion in tanks.

The dynamic responses of the semi-submersibles with/without heave plates were evaluated numerically. All the variables that effect on the responses of the semi-submersible were identified and the relationship between these variables were studied based on the rational assumptions and approximations made. The model was designed as rigid bodies and taking consideration of the six degree of freedom; surge, sway, heave, roll, pitch and yaw. Nonetheless, this study was focused more on the heave motion as stated in the objectives.

The numerical modelling was started off with the mesh generation of the semi-submersible model. The input file was the dimensions of the semi-submersible such as the column size and length and pontoon size. The mesh was made up of the patches in which was formed through the connection of four nodes. The right hand rule was applied in this numerical modelling. The nodes of the panel were arranged accordingly to this rule to ensure the execution of normal vector and it was in the right direction.

An .HST file was generated once the mesh input file was completed and read by the HydroSTAR. The .HST file was then read and few properties of the model were generated such as reference point and centre of buoyancy. The mesh of the model was then verified in the hschk module in terms of the consistency of the normal vector orientation, panels with null area, panels over the surface, panels at free surface, overlapped panels and holes (neighbour-absences).

The next stage was the establishment of the sea parameter which includes the wave range, incident wave angle and water depth. This input was saved as .RDF file and then, radiation and diffraction computation was started. The mechanical properties of the model such as the mass, centre of gravity, stiffness matrix and radius of gyration was also later established and used as an input file in the hsmec module. The outcome of this module determined whether the model is balanced or not from its motion computation.

Once the motion computation was completed and the model was deemed as balance, the second order mean drift loads were calculated. The near-field, middle-field and far-field formulations were defined as the input in this hsdft module. In the final phase, the construction of the transfer functions was generated in order to get the response of the semi-submersible in terms of response amplitude operators (RAO) in frequency domain and the results were presented in tabular form and graphically. Figure 8 shows the flow of the numerical modelling in HydroSTAR.

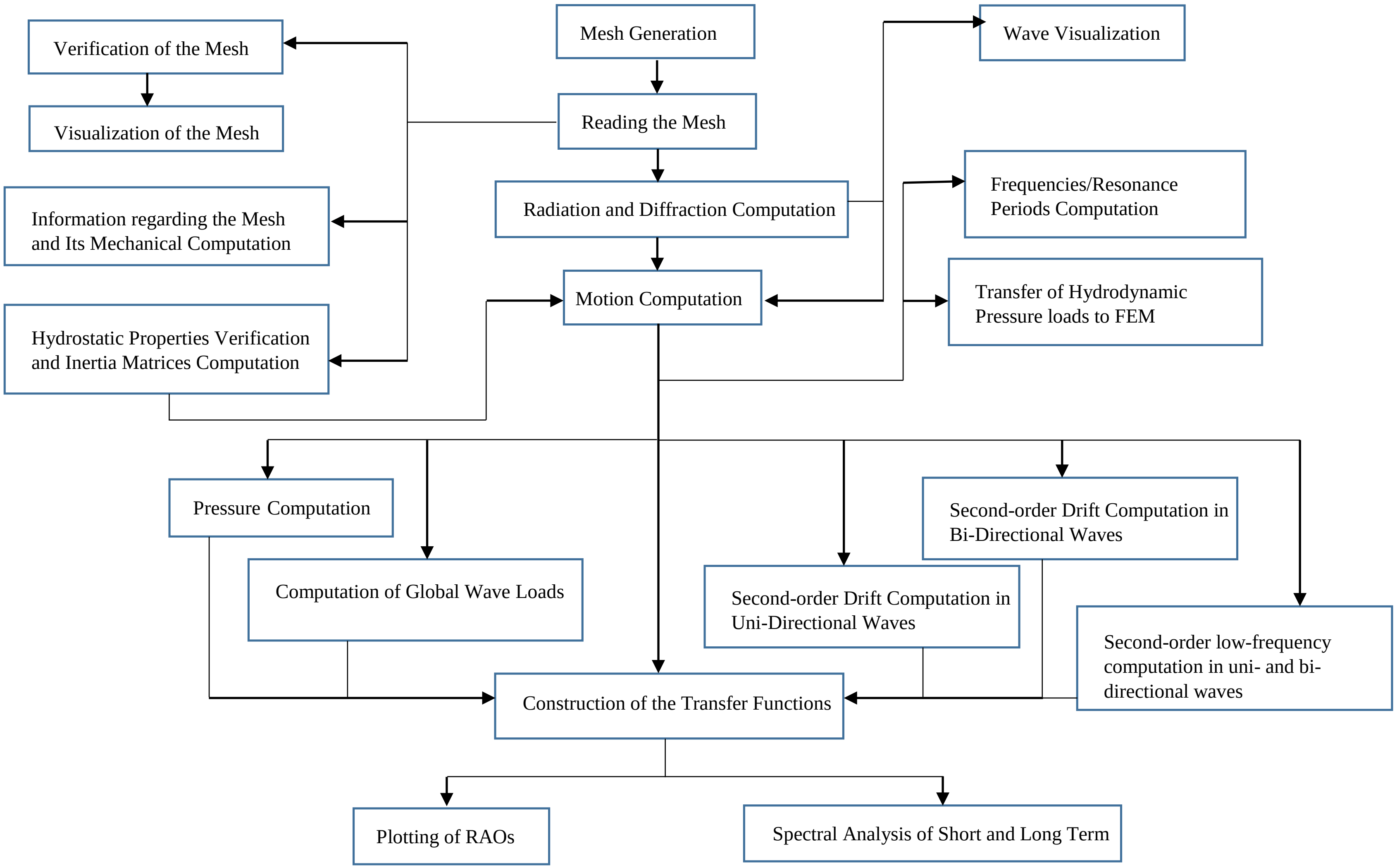


Figure 8: Numerical Modelling Flowchart

CHAPTER 4
RESULTS AND DISCUSSIONS

4.0 Chapter Overview

This chapter includes the general data of the semi-submersible and also the validation of the numerical method used in this study. The effect of the heave plates in terms of elevation, sizes and shapes towards the heave motion RAOs of the semi-submersible are also shown and discussed.

4.1 Modified Design Parameter

In order to minimize the inconsistency and errors during the mesh generation, the initial dimensions of the semi-submersible has been modified to a more suitable values as shown in the Table 5. The table also contains the detailed information regarding the semi-submersible such as total mass, draft, CG, radius of gyration and the mooring line.

Table 4: General Data of the Semi-submersible

Parameter	**Unit**	**Value**
Square Column size	m x m	20 x 20
Column Length	m	50
Total Mass	tonne	106600
Pontoon Size	m x m	60 x 20
Draft	m	35
Freeboard	m	15
Centre of Gravity	m	23.09
Centre of Buoyancy	m	11.73
Metacentric Height (GM)	m	20.697
Radius of Gyration: Rxx	m	43.27
Ryy		45.78
Rzz		45.78
Mooring Line Stiffness	kN	1294

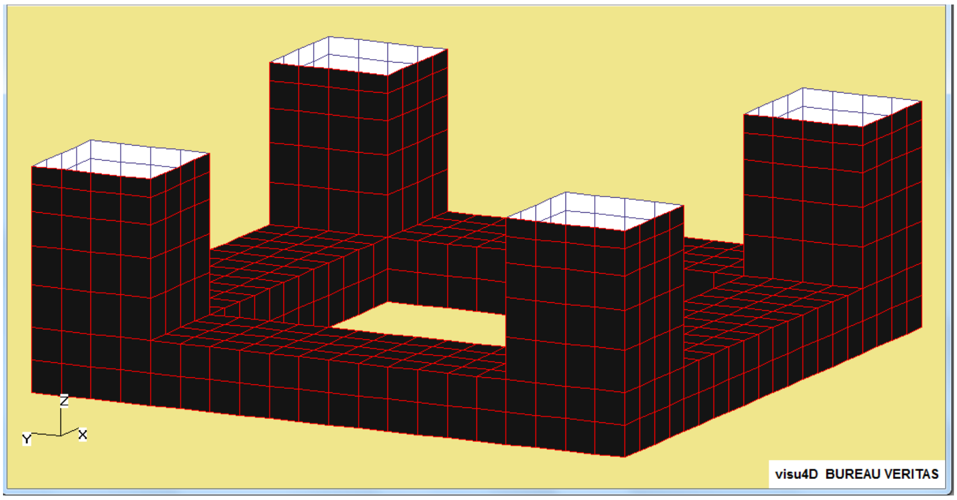


Figure 9: Semi-submersible Model

The conventional semi-submersible model was subjected to a regular wave of a frequency from 0.1 rad/s until 2.1 rad/s with 0.1 rad/s increment each time in the water depth of 1710 m and the motion RAOs of the semi-submersible in terms of heave, surge and pitch are obtained and shown in the Figure 10, 11 and 12 respectively.

4.2 Validation of Numerical Method

The motion of the semi-submersible were obtained through frequency domain computation which involved a simple iterative method. Based on the Figure 10, the motion RAO of the semi-submersible at its maximum when the semi-submersible is subjected to wave frequency of 0.2 rad/s; 1.14 m/m.

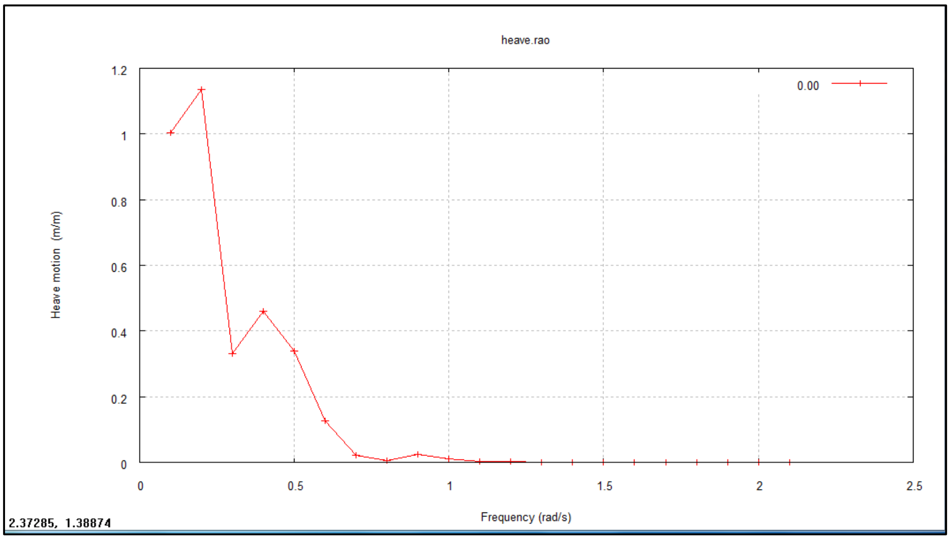


Figure 10: Heave Motion RAO (Frequency)

Moreover, it is shown that in all the figures, the motion RAOs decreased rapidly (almost nearing zero) when it is in the ranges of 1.5 rad/s and 2.1 rad/s. The reason is that higher wave frequency has a lower wave period and as a result, shorter wavelength and this short wavelength only causes a relatively low pressure forces to the floating structure.

The raw data from the Figure 10 were then used to form a motion RAO vs period graph as shown in the Figure 11 using the equation below:

$$T = \frac{2\pi}{\omega} \tag{29}$$

With this, the result of this study can be compared with the results obtained by Chen, Mei and Mills (2007) for validation purpose. Their study was performed using MULTISIM which is a time domain platform motion simulation program. Hence, some discrepancies are expected from this comparison.

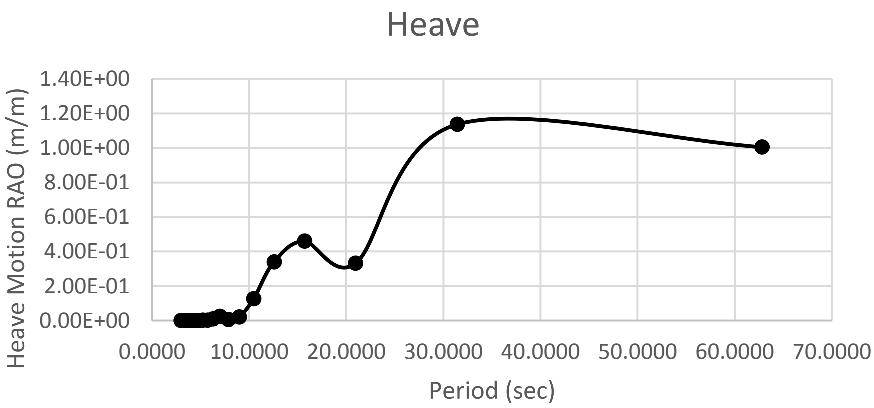


Figure 11: Heave Motion RAO vs Period

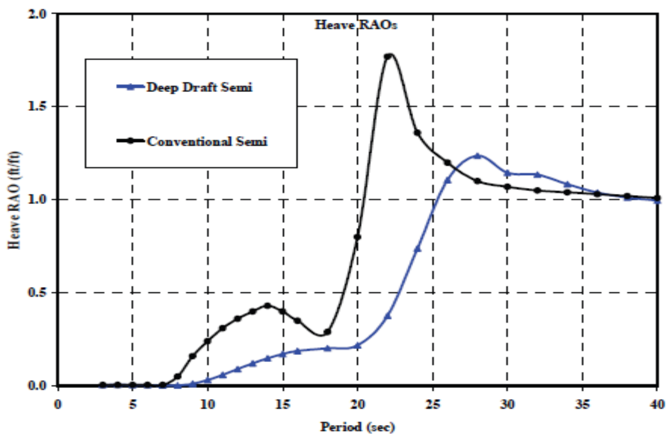


Figure 12: Heave Motion RAO [Chen, Mei & Mills (2007)]

Based on the Figure 11 and 12 it is shown that there is a similar pattern of the heave motion RAOs between the results of this study and the results obtained by Chen, Mei and Mills (2007). Nevertheless, there is a difference of 36% in terms of its value. This may be resulted from the minor alteration of the dimensions of the pontoons and columns of the semi-submersibles. The dimensions used in this study is slightly larger than the dimensions used to generate the conventional semi-submersibles in the study conducted by Chen, Mei and Mills (2007). Therefore, the submerged volume and the total mass of the model in this study were also larger. Subsequently, the inertia and the damping matrix of the body in the motion computation were increased as well and thus, affecting the RAO by the end of the transfer function.

In addition, the wave-heading angle applied in the study also affect the responses shown by the semi-submersible. Different wave-heading angles will generate different responses. Pedersen (2012) claimed that the highest response for heave, pitch and roll are generated when the wave headings at 90 ° (beam sea) and 180° (head sea). In this study, the wave-heading angle was at 180° (head sea) whereas the wave-heading angle used in the study performed by Chen, Mei and Mills (2007) was not stated clearly and thus, in an ambiguous state. Furthermore, the neglection of the current and wind effects in this study may also be one of the cause of discrepancies in the result.

It can be concluded that the result of this study is acceptable as the difference is within the tolerable ranges. With this, the numerical data has been validated and thus, the next analysis on the effects of heave plates on the heave responses of conventional semi-submersible can be conducted and discussed.

4.3 Effects of Heave Plates in terms of Elevation

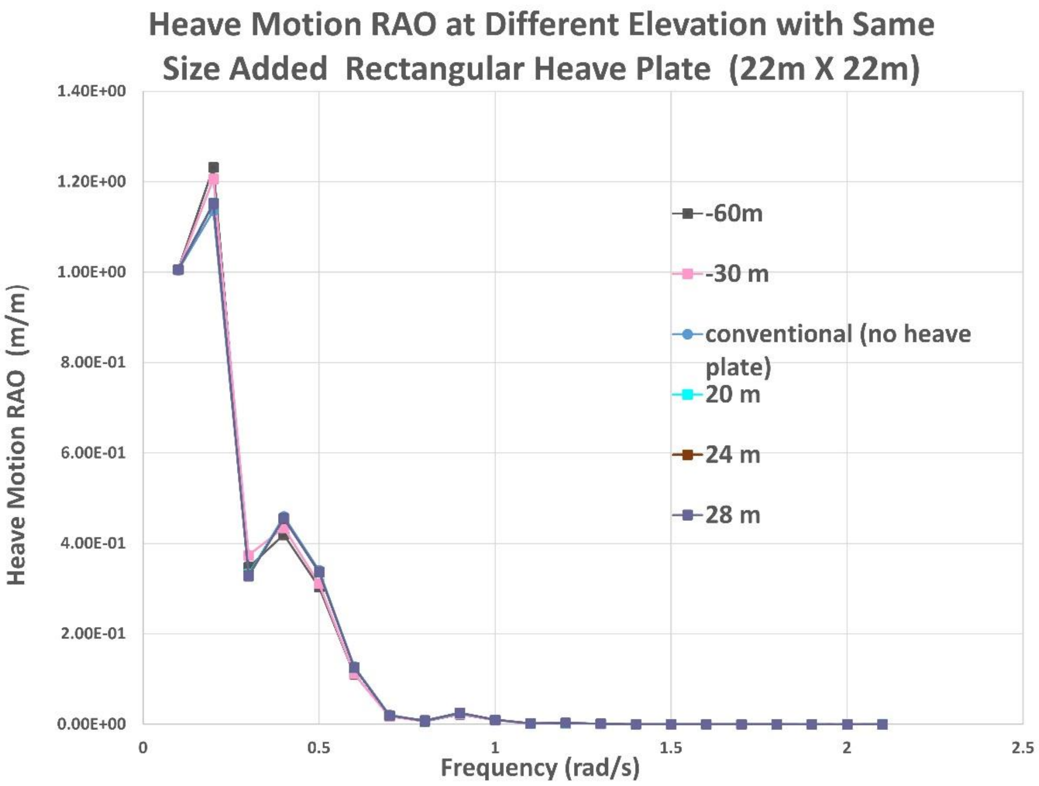


Figure 13: Heave Motion RAO at Different Elevation with 22m X 22m Size Rectangular Heave Plate

Figure 13 shows the results of heave motion RAO at different elevation for the semi-submersible with added rectangular heave plates with a size of 22 m x 22 m. The mesh of the heave plates were generated on the each column of the semi-submersible. The range of the elevation in this study is from -60 m, -30 m, 20 m, 24 m and 28 m. These elevations were measured from the keel of the semi-submersible.

As shown in from Figure 13, the maximum heave motion RAO of the semi-submersible is at 0.2 rad/s. However, since the maximum wave period in the environmental data of this study is 14.2 s which is approximately 0.44 rad/s, only the heave motion RAOs from 0.4 rad/s until 2.1 rad/s was taken account for and discussed throughout this study.

From 0.4 rad/s onwards, it is shown that the higher the elevation of the heave plate from the keel, the higher is the heave motion RAO. For example, in Figure 12, the highest heave motion RAO at 28 m is 0.455 m/m whereas the highest heave motion RAO at -60 m 0.419 m/m. Both of these motion RAO occurred at the wave frequency at 0.4 rad/s. The highest heave motion RAO value of the conventional semi-submersible without any heave plate at the similar wave frequency is 0.46 m/m. This value is used as a benchmark to judge the heave performance improvement of the heave plates.

Lower heave motion RAO means better heave performance of the semi-submersible. From the result, it implies that the semi-submersible will have a better heave performance if the heave plates are placed at the lower elevation; -60 m. The results shown here is in agreement with the results obtained by Chen, Mei & Mills (2007) in which they claimed that the heave plate will be more effective in lowering the heave motion of the semi-sub if it is placed about 60 m or more below the semi- keel.

The reason is that the wave load increases as the elevation goes from the sea-bottom up to the sea surface. Hence, by adding the heave plates at higher elevation, it will only increase the cross section area exposed to the wave load. Eventually, the wave loads on the heave plate outweighed the propitious added mass and damping introduced by the heave plate and caused the semi-submersible to have higher motion in heave.

On the other hand, at lower elevation, the wave load is lower and will not exert much load onto the heave plates. Hence, as shown in the results, adding heave plates at lower elevation (-60 m) caused the heave motion RAO to decrease as heave plates provide added mass and damping to the structure and thus, resulted in a larger wave exciting forces.

4.4 Effects of Heave Plates in terms of Size

As the previous section shows that the elevation plays an important role in determining the effectiveness of the heave plate. Heave plate perform better at lower elevation. Therefore, in this section, the elevation was fixed at -60 m before the variation of the sizes of heave plates for the rectangular, hexagon and cylinder. Taking into consideration of the maximum limit of the total structure mass, the maximum size for rectangle plate is 29 x 29, cylinder plate is 29 ($\emptyset$) and hexagon plate is 31($\emptyset_1$) and 35.8 ($\emptyset_2$).

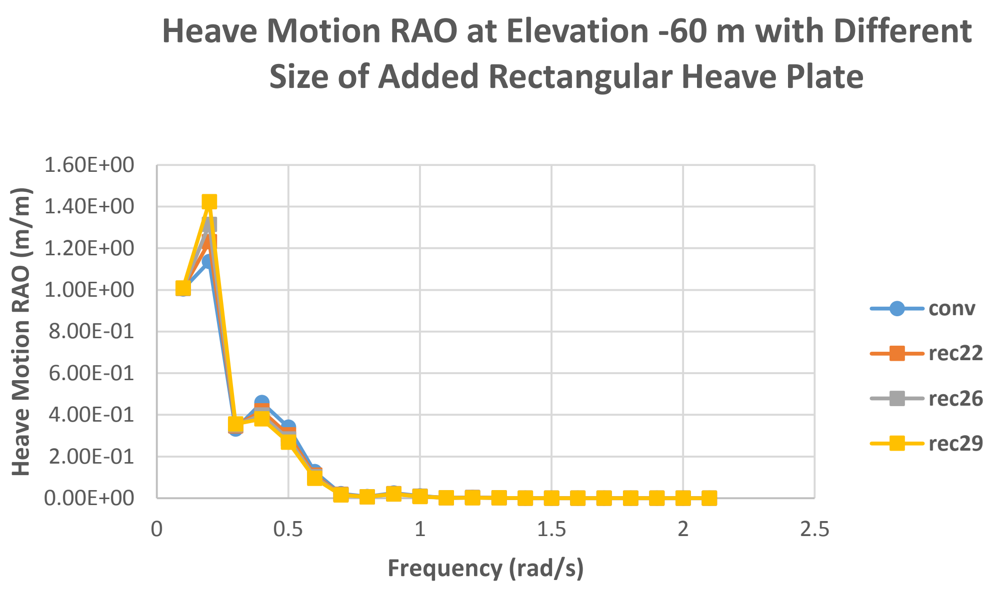


Figure 14: Heave Motion RAO at Elevation -60 m Different Size of Added Rectangular Heave Plate

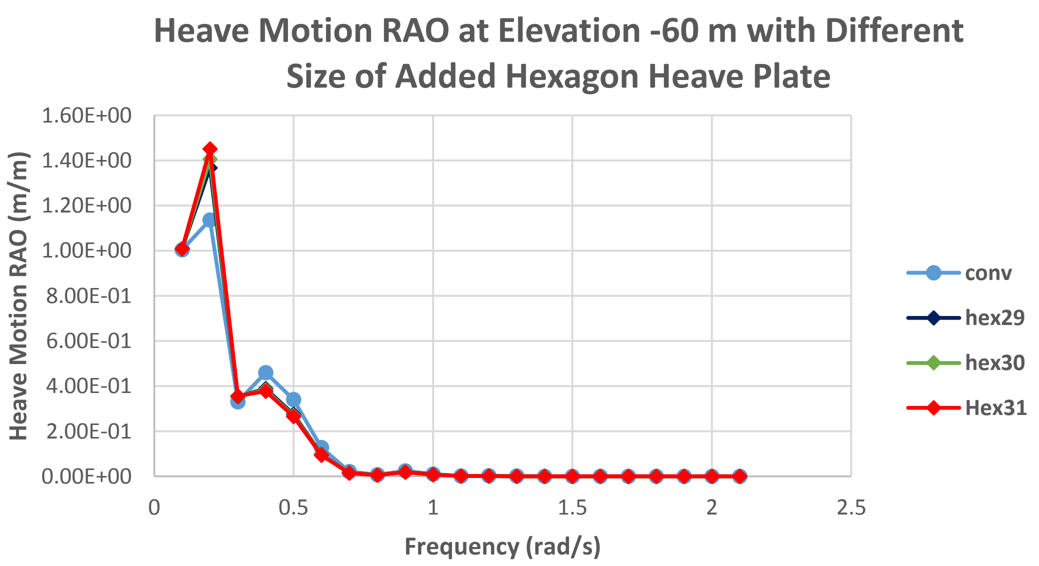


Figure 15: Heave Motion RAO at Elevation -60 m Different Size of Added Hexagon Heave Plate

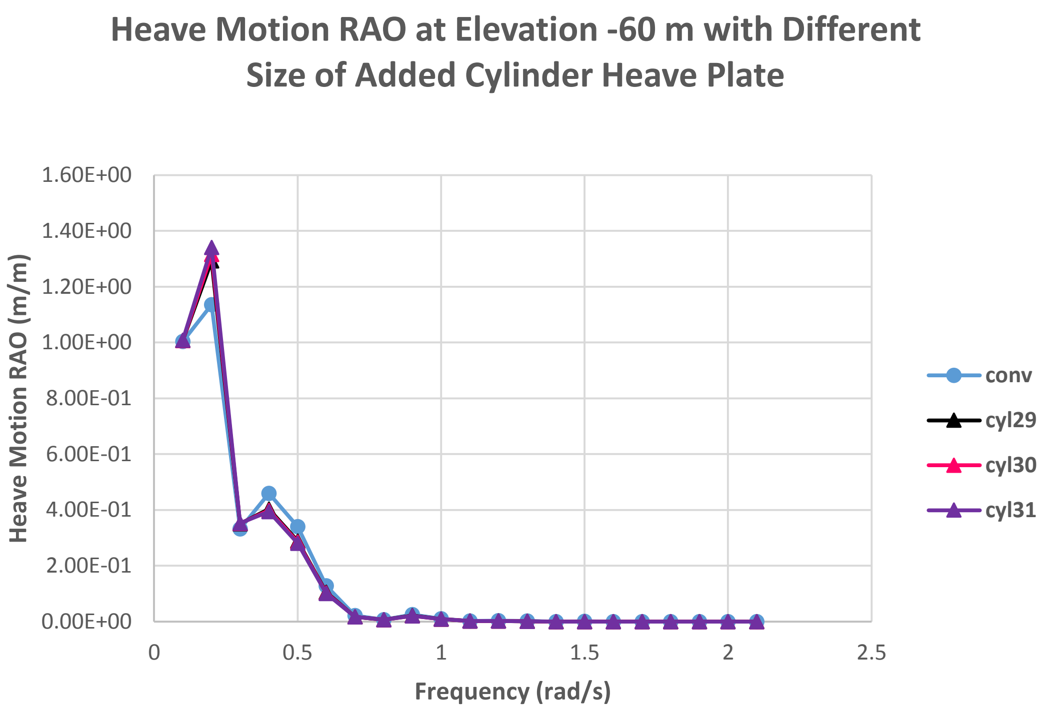


Figure 16: Heave Motion RAO at Elevation -60 m Different Size of Added Cylinder Heave Plate

Based on Figure 14, the semi-submersible with the largest size of added heave plates has the lowest heave motion RAO among all. Semi-submersible with the added rectangular heave plate of size 29 x 29 has the lowest heave motion RAO; 0.381 m/m. On the other hand, rectangular heave plate of 22 x 22 and 24 x 24 has higher values which are 0.419 rad/s and 0.399 rad/s.

Similarly, in Figure 15 and Figure 16, the lowest heave motion RAO also belongs to the semi-submersible with the largest size of added heave plates. This indicates that the semi-submersible with a larger size of heave plates will always have a lower heave motion RAO compare to the smaller size in their own respective shapes.

An additional fluid inertia forces will be included in the motion computation of the structure when there is an acceleration imposed on the fluid flow due to the accelerating or decelerating body. These fluid inertia forces is known as added mass. The relationship between the added mass of an object and its volume is directly proportional. Larger size of heave plates have a larger surface and volume. Hence, an increase on the size of the heave plates caused an increase on the inertia effect of the heave plates and subsequently, decreased the heave motion RAO of the semi-submersible.

4.5 Effects of Heave Plates in terms of Shape

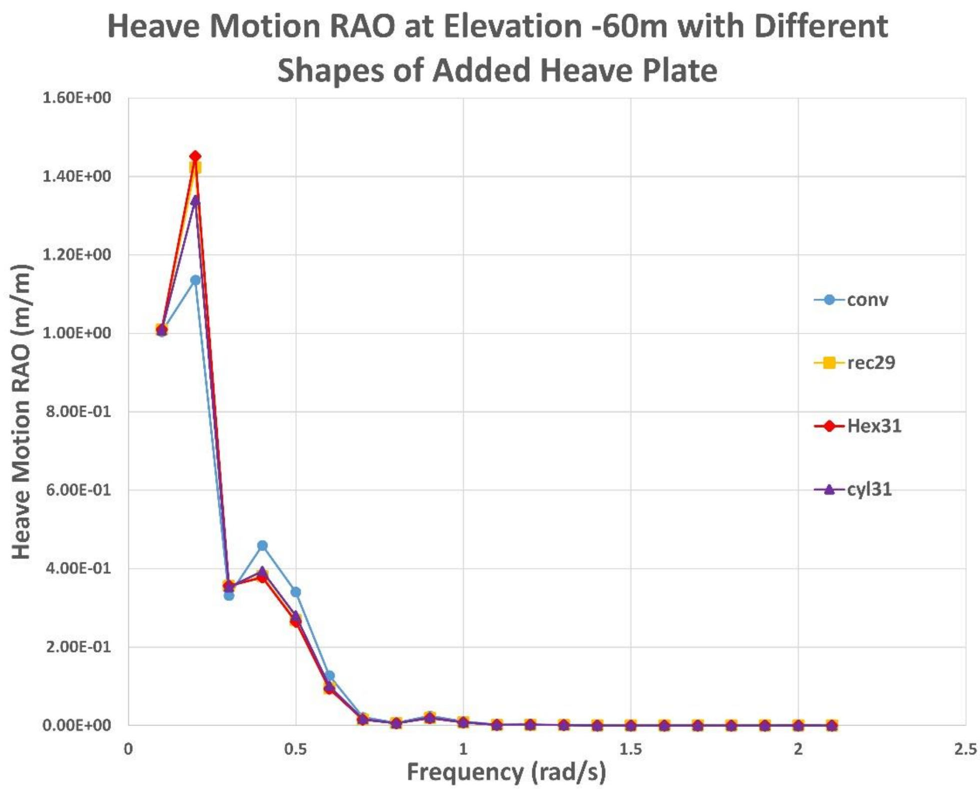


Figure 17: Heave Motion RAO at Elevation -60 m Different Shapes of Added Heave Plates

Figure 17 shows the comparison of heave motion RAO between the submersible with different shape of heave plates such as rectangle, hexagon and cylinder. Based on the result, semi-submersible with the cylinder heave plates has the highest heave motion RAO among all with a value of 0.394 m/m. The second highest belongs to the semi-submersible with the rectangle heave plates; 0.381 m/m followed by the semi-submersible with the hexagon heave plates; 0.378 m/m. This means that hexagon heave plate has the best heave reducing effectiveness compare to other two shapes.

Due to the shape differences, their surface area and also volume also differed from one another. Rectangle heave plate (29 m x 29 m) has the largest volume; 1009.2 m^3 while hexagon heave plate (d1 =31 m, d2 = 35.8 m) has a volume of 998.7 m^3. There is only 1.04 % of difference between rectangle and hexagon heave plates in term of their volume. On the other hand, cylinder heave plate has the smallest volume; 905.72 m^3.

As proven earlier, heave plate with lower volume will have a lower added mass. This could explain why the cylinder heave plate has the worst heave reducing effect. On the contrary, the hexagon heave plate still manage to have the best heave reducing effect even though its added mass is lower compare to the rectangle heave plate. The reason might be due to the fact that the added damping introduced by the hexagon heave plate is so great that it overcame the favourable added mass and damping introduced by the rectangle heave plates.

The high value of added damping of the hexagon heave plate may resulted from its large diameter size; diameter 1 = 31 m and diameter 2 = 35.8 m. Conversely, rectangle heave plate only has a diameter of 29 m. This finding is actually coherent with the study conducted by Cai and Tao (2004) regarding on the heave motion suppression of a spar with a heave plate. In the study, they claimed that the vortex shedding modes as well as the hydrodynamic properties of the heave plates are significantly influenced by the geometry configurations of the spar cylinder and disk (heave plate) , such as the aspect ratio of the disk; t_d/D_d and diameter ratio; D_d/D_c.

D_d and t_d referred to the diameter and thickness of the disk whereas D_c referred to the diameter of the cylinder. However, in this study, D_c will be referred as the diameter of the column of the semi-submersible. Since the thickness of the heave plate was maintained at 1.2 m throughout this study, the effect of the aspect ratio of the heave plate is not applicable in this case. Hence, only the effect of the diameter ratio; D_d/D_c will be considered and discussed in this study. An increase in the diameter ratio increases the drag forces imposed by the flowing water surrounding the heave plate and when the drag forces increases, the added damping increases as well. Damping is the occurrence of a drag force which is non-conservative; the mechanical energy is removed gradually. As a result, the amplitude of an oscillation or vibration will be reduced due to the energy being dissipated.

Nonetheless, according to Cai and Tao (2004), any further increase in a certain diameter would not result in considerable increases in drag. The optimum diameter ratio is normally within the range of 1 to 2. In this study, the diameter ratio of the hexagon, cylinder and rectangle heave plates are 1.79, 1.55 and 1.45 respectively. Cai and Tao (2004) also stated that larger heave plate diameter will result in a stronger vortex shedding processes which lead to an effective mean of energy dissipation. With this, the damping force increases as well as the exciting force in the motion equation. Consequently, the heave motion of the system reduces.

From this study, it can be deduced that hexagon heave plate has a better hydrodynamic properties than the rectangle and cylinder heave plates. For that reason, it acts as a better mechanical damping devices and eventually lead to a better heave performance.

4.6 Heave Performance Improvement of the Semi-sub with Added Heave Plates

Through the data collected from the results, the maximum heave response of the conventional semi-submersible as well as the semi-submersible with added heave plates can be calculated and compared in order to determine the heave performance improvement. The heave response can be calculated through the formula:

$$H(w) = RAO \times \text{Wave Amplitude} \quad (30)$$

Where,

H (w) = heave response

Wave Amplitude = Wave height/2

Thus, the heave response of the conventional semi-submersible is 3.174 m given that the heave motion RAO is 0.46 m/m and the wave amplitude is 6.9 m. As discussed earlier, the ideal system in this study is the semi-submersible with the added hexagon heave plate with a size of diameter 1 = 31 m and diameter 2 = 35.8m. The heave plates were located at 60 m below the semi keel. For that reason, the calculation of the heave response of the semi-submersible with added heave plates is based on it. From the calculation, the heave response is 2.61 m given that the heave motion RAO is 0.378 m/m and the wave amplitude remained the same, 6.9 m.

With this, the heave performance improvement can be obtained. If the calculation shows a positive value, it signifies that there is an improvement in the heave motion and vice versa. It can be calculated by using the formula as below:

$$\text{Heave Performance Improvement} = \frac{A-B}{A} \times 100\% \quad (31)$$

Where,

A = Highest heave response of the conventional semi-submersible (3.174 m)

B = Highest heave response of the semi-submersible with added heave plate (2.61m)

Based on the calculation, the heave performance of the semi-submersible improved drastically by 17.8 %.

This heave performance improvement is significant and it may allow the semi-submersible to be considered as a viable vessel for dry-tree risers. Nevertheless, since the heave plates have to be attached at the elevation of 60 m below the semi keel, the design of the connections between the columns and the heave plates has to be taken into consideration in order for this conceptual design to be practical and feasible.

CHAPTER 5
CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The study on the effect of the heave plates on the heave performance of a conventional semi-submersible is presented. From the study, it can be concluded that the performance of the semi-submersible behaviour on heave motion can be enhanced through the addition of the heave plates provided that the heave plates are attached at the lower elevation below the semi keel. The study also reveals that the effectiveness of heave plates to act as an additional mechanical damping devices is also influenced by their sizes and shapes. Nonetheless, practical design of the connections between the columns and heave plates has to be taken account into as well for the feasibility of this study.

5.2 Recommendation

Based on the current study, there are few recommendations that can be done in order to further improve the results of future study. These recommendations are as follow:

1. Experimental work is required in order to validate the numerical results.
2. New parameter study on the heave plates can be included such as the porosity of the heave plates and the form edges of the heave plates.
3. The viscous damping introduced by the heave plates should be studied and included in the dynamic analysis in order to obtain a more accurate result.
4. Wind and current load should be included in the numerical and experimental modelling.
5. Conduct the hydrodynamic analysis of this study in time domain analysis as it includes the presence of all system nonlinearities and thus, a more accurate result can be obtained.

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APPENDIX

Appendix A

List of Operating Semi-Submersibles

No.	Field	Vessel	Hull	Oil Company	Location	Water Depth (m)	First Oil	Last Oil	Field Reserves Oil (MMbbl)	Field Reserves Gas (Bm^3)	EPS
1	Argyll	North Sea Pioneer	Transworld 58	Amerada (Hamilton)	UK	79	1975	1984	73		
2	Casablanca		Aker H3	Repsol	Spain	135	1977	1978	12		EWT
3	Enchova		Sedco 135D	Petrobras	Brazil	122	1977	1980	9		Yes
4	Casablanca	Afortunada		Repsol	Spain	135	1978	1982	12		Yes
5	Dorada		Sedco I	Repsol	Spain	93	1978	1985	16		
6	N Garoupa	P22	Sedco 135F	Petrobras	Brazil	124	1979	1982	4		Yes
7	E Enchova	P24	Penrod 72	Petrobras	Brazil	111	1979	1983	2		Yes
8	Pampo		Secdo 135D	Petrobras	Brazil	117	1980	1981	2		Yes
9	Pampo/Linguado	P I	Transworld 61	Petrobras	Brazil	115	1980	1984			Yes
10	Buchan	Buchan A	Pentagone 83	Talisman (BP)	UK	118	1981		13		
11	Bicudo		Sedco 135D	Petrobras	Brazil	135	1982	1988	72	2.2	Yes
12	Bonito		Penrod 71	Petrobras	Brazil	190	1982	1994			
13	Garoupinha	P22	Sedco 135F	Petrobras	Brazil	113	1982	1986			Yes
14	S Pampo	P21	Sedco Staflo	Petrobras	Brazil	112	1983	1988			Yes
15	Corvina	P9	Aker H3	Petrobras	Brazil	230	1983				
16	Pirauna	P15	MD 503	Petrobras	Brazil	243	1983		10		
17	Anequim		Pentagone 82	Petrobras	Brazil	107	1984	Yes			
18	Argyll/Duncan/Innes	Deepsea Pioneer	Aker H3 AC	Hamilton	UK	79	1984	1992			
19	Trilha/Badejo	P I	Transworld 61	Petrobras	Brazil	115	1984	1989			
20	Linguado	P12	Aker H3	Petrobras	Brazil	103	1984				
21	Parati/Anequim		Pentagone 7	Petrobras	Brazil	105	1985	1989			
22	Viola	Atlantic Zephyr	Ocean Zephyr	Petrobras	Brazil	126	1985				
23	Innes	North Sea Pioneer	Transworld 58	Hamilton	UK	72	1985	1986			
24	Balmoral	Balmoral	GVA 5000	Agip (Sun)	UK	143	1986		100		
25	Moreia	P22	Sedco 135F	Petrobras	Brazil	114	1986		26	15.4	
26	Bicudo	P7	Aker H3	Petrobras	Brazil	209	1987		72	2.2	
27	Green Canyon 29	Green Canyon 29	Penrod 72	Placid Oil	USA	469	1988	1990	1		
28	Birch	Benvrackie		Occidental	UK	128	1988	1989			
29	Enchova		Sedco 135D	Petrobras	Brazil	122	1989	Yes		2.2	
30	Badejo/Trilha	P21	Sedco Staflo	Petrobras	Brazil	112	1989	1998			
31	Ivanhoe/Rob Roy	AH001	Sedco 700	Amerada Hess	UK	140	1989				
32	Veslefrikk	Veslefrikk B		Statoil	Norway	175	1989		330	2.3	
33			Sedco K	ONGC	India		1989	1993			
34	Crawford	North Sea Pioneer	Transworld 58	Hamilton	UK	117	1989	1991	4		
35	Marlim	P13	THP-2800	Petrobras	Brazil	620	1991	1992		2.4	Yes
36	Coral		Ocean Century	Petrobras	Brazil	150	1991	Yes	280	15.0	
37	Emerald	Emerald Producer	Aker H3	MSR	UK	150	1992	1996	17		
38	Marlim	P20	GVA 4000	Petrobras	Brazil	625	1992		1500		Yes
39	Pirauna/Marimba	P8	Aker H3	Petrobras	Brazil	423	1993		85		
40	Bijapura/Salema	P13	THP-2800	Petrobras	Brazil	625	1993		155	2.4	
41	Caravela	P14	THP-2800	Petrobras	Brazil	195	1993				
42	Dai Hung	Dai Hung 1 (Deepsea Pioneer)	Aker H3 AC	Vietsov (BHP)	Vietnam	120	1994		100	56.6	
43	Albacora	P24	Penrod 72	Petrobras	Brazil	252	1994		660	9.0	Yes
44	Marlim	P18	GVA 4500	Petrobras	Brazil	910	1994				
45	Machar		Sedco 707	BP	UK	100	1994	1996			
46	Troll West	Troll B		Norsk Hydro	Norway	320	1996		470		
47	Banff		Sedco 707	Conoco	UK	100	1996	1997	100	1.1	EWT
48	Albacora	P25	Zapata SS4000	Petrobras	Brazil	515	1996		660		
49	Cooper	Enserch Garden Banks	Ocean Victory	Enserch	USA	668	1996	1999	45		
50	Liuhua 11-1	Nan Hai Tiao Zhan	Sedco 700	Amoco	China	332	1996				
51	Njord	Njord A	Aker P-45	Norsk Hydro	Norway	330	1997		200	7.2	
52	Oribi	Orca	Sedco I	Soekor	S Africa	118	1997		15		
53	Marlim	P19	Penrod 78	Petrobras	Brazil	770	1997		1500	34.0	
54	PY-3	Tahara	Sedco K	British Borneo	India	82	1997		20		Yes
55	Galley	Northern (Emerald) Producer	Aker H3	Texaco	UK	140	1998		67		
56	Marlim	P26	Pacesetter	Petrobras	Brazil	990	1998				
57	Visund	Visund FPS	GVA 8000	Norsk Hydro	Norway	335	1998		305	60.0	
58	Voador	P27	Penrod 71	Petrobras	Brazil	530	1998				
59	Marimba	P21	Sedco Staflo	Petrobras	Brazil	700	1998				
60	Janice	Janice A	Aker H3.2	Phillips	UK	80	1999		70		
61	Troll West	Troll C	GVA 8000	Norsk Hydro	Norway	340	1999		380	46.0	
62	Roncador	P36	Trendsetter	Petrobras	Brazil	1360	2000				
63	S Marlim	P40		Petrobras	Brazil	1080					
64	Asgard	Asgard B	GVA 70	Statoil	Norway	300	2000		770		
65	Snorre	Snorre B		Norsk Hydro	Norway	350	2001		365	6.9	

Appendix B

Input and Output of Modules in HydroSTAR

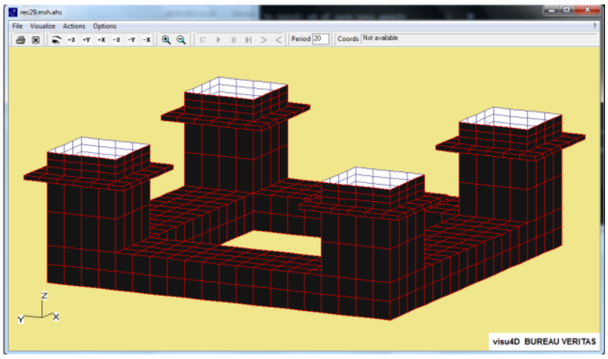
Module	Keyword	Input	Output
Mesh Generation	hsmsh	Main dimensions of the body	Input file for hslec
Reading the Mesh	hslec	Body's geometry (coordinates, panel connectivity and condition of symmetry)	Hydrostatic properties of the body (volume, inertia, centre of buoyancy, waterplane area and wetted surface)
Verification of the Mesh	hschk	Output of hslec	Check of the mesh (inconsistency, normal orientation)
Visualization of the Mesh	hvisu	Output of hschk	View of the mesh
Information about the Mesh and its Mechanical Components	hsinf	Output of hslec	Information regarding the Mesh (mean length of panels) or mechanical components (frequencies, headings)
Hydrostatic Properties Verification and Inertia Matrices Computation	hstat	Weight distribution	Hydrostat properties and input data for hsmcn and hswld

Radiation and Diffraction Computation	hsrdf	Wave conditions (wave frequencies, water depth)	Elementary solutions including added mass, radiation damping and wave excitation loads
Wave Visualization	hswav	Free surface mesh and wave components to visualize	Input data for simulation of vessel's motions and waves
Motion Computation	hsmcn	Mechanical Properties (mass distribution, additional stiffness and additional damping matrices)	Motions of floating bodies
Frequencies/ Resonance Period Computation	hsrsn	Output of hsmcn	Resonance period/ Frequencies
Transfer of Hydrodynamic Pressure Loads to FEM	hsfem	Whole ship finite element model and wave conditions (heading and frequency)	Real and imaginary parts of hydrodynamic pressure loads
Construction of the Transfer Function	hsrao	Selection of the transfer function and the name of the storage file	Transfer function of motions, velocities, acceleration and second order loads
Pressure Computation	hsprs	Coordinates of points to compute pressure	Pressure at the given points

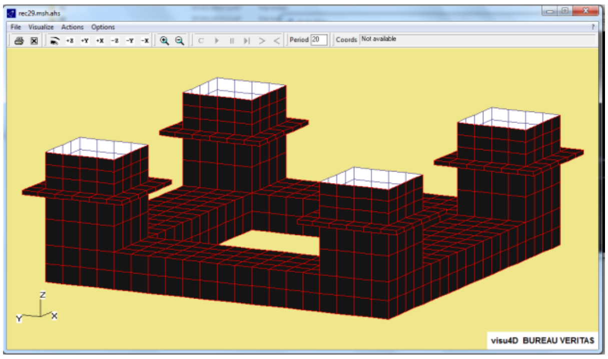
Computation of Global Wave Loads	hswld	Mass distribution along the vessel	Efforts per defined station
Second-order Drift Computation in Uni-Directional Waves	hsdft	Choice of formulation field type (near-field, middle field or far field)	Second-order drift loads in uni-directional waves
Second-order Drift Computation in Bi-Directional Waves	hsmdf	Choice of formulation field type (near-field or middle field)	Second-order drift loads in bi-directional waves
Second-order Low Frequency Computation in Uni and Bi-Directional Waves	hsqtf	Choice of formulation field type (near-field or middle field); different frequencies and wave frequencies for the computation	Second order low frequency loads in uni and bi-directional waves
Plotting of RAOs	hsplt	Output data from hsrao	Graphic view of RAOs
Spectral Analysis of Short and Long term	hspec	Wave data	Spectral results

Appendix C

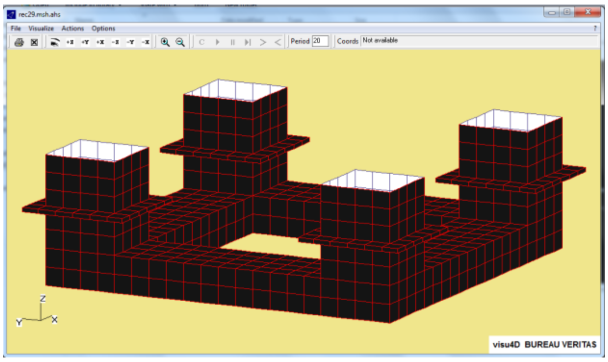
Images of the Mesh Generation



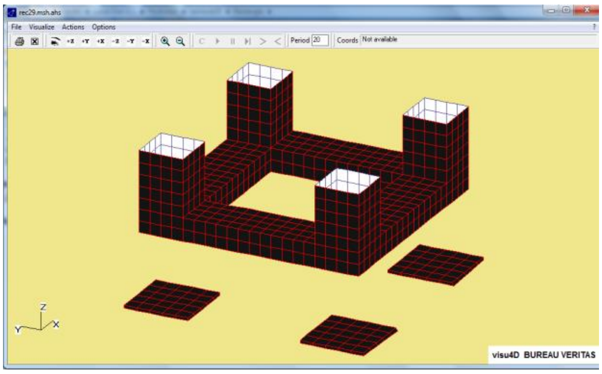
Rec at Elev 28 m



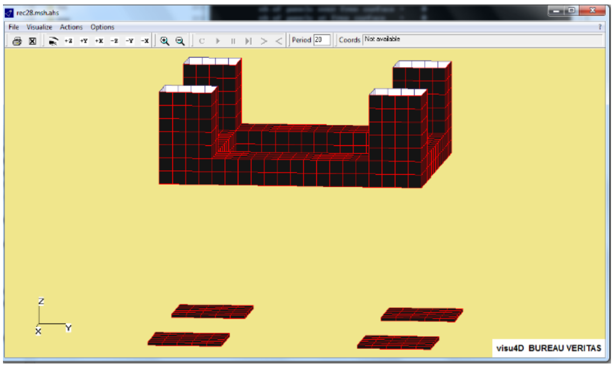
Rec at Elev 24 m



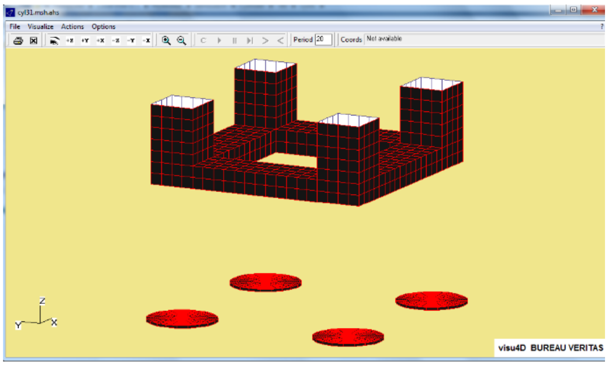
Rec at Elev 20 m



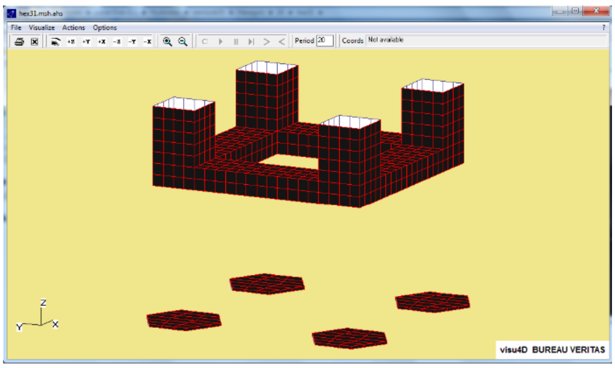
Rec at Elev -30 m



Rec at Elev - 60 m



Cyl at Elev - 60 m



Hex at Elev - 60 m

Appendix D

Raw Data of Heave Motion RAO of Conventional Semi-submersible and Semi-submersible with Added Rectangle Heave Plates (22 m x 22 m).

Wave Frequency (rad/s)	Heave RAO (m/m)					
	Elevation -60 m	Elevation -30 m	Conventional (no heave plate)	Elevation 20 m	Elevation 24 m	Elevation 28 m
0.1	1.01E+00	1.01E+00	1.00E+00	1.01E+00	1.01E+00	1.01E+00
0.2	1.23E+00	1.21E+00	1.14E+00	1.15E+00	1.15E+00	1.15E+00
0.3	3.47E-01	3.74E-01	3.31E-01	3.32E-01	3.29E-01	3.28E-01
0.4	4.19E-01	4.35E-01	4.60E-01	4.55E-01	4.55E-01	4.55E-01
0.5	3.04E-01	3.12E-01	3.40E-01	3.36E-01	3.37E-01	3.37E-01
0.6	1.11E-01	1.12E-01	1.27E-01	1.25E-01	1.25E-01	1.26E-01
0.7	1.83E-02	1.72E-02	2.11E-02	1.98E-02	1.96E-02	1.94E-02
0.8	6.36E-03	7.03E-03	6.95E-03	7.79E-03	8.01E-03	8.33E-03
0.9	2.20E-02	2.25E-02	2.45E-02	2.51E-02	2.53E-02	2.55E-02
1	9.16E-03	9.24E-03	1.01E-02	1.02E-02	1.01E-02	9.86E-03
1.1	1.75E-03	1.78E-03	2.06E-03	2.00E-03	2.09E-03	2.20E-03
1.2	2.49E-03	2.51E-03	2.78E-03	2.95E-03	3.11E-03	3.32E-03
1.3	1.13E-03	1.13E-03	1.32E-03	1.26E-03	1.32E-03	1.34E-03
1.4	1.09E-04	1.08E-04	7.84E-05	1.19E-04	1.14E-04	1.85E-04
1.5	1.18E-04	1.17E-04	1.66E-04	1.23E-04	1.75E-04	3.22E-04
1.6	7.00E-05	7.07E-05	1.54E-05	6.49E-05	3.37E-05	1.09E-04
1.7	6.54E-05	6.52E-05	3.02E-05	3.87E-05	5.59E-05	2.12E-04
1.8	6.43E-05	6.52E-05	1.48E-05	5.85E-05	3.43E-05	9.87E-05
1.9	9.45E-05	9.57E-05	2.79E-05	7.65E-05	8.10E-06	1.22E-04
2.0	2.80E-05	2.85E-05	7.84E-06	1.60E-05	2.46E-05	1.16E-04
2.1	9.72E-05	9.96E-05	2.00E-05	9.10E-05	2.72E-05	7.19E-05

Appendix E

Raw Data of Heave Motion RAO of Semi-submersible with Added Rectangle Heave Plates with Different Size at Elevation -60 m

Wave Frequency (rad/s)	Heave Motion at Elevation – 60 m			
	Conventional	(22 m x 22 m)	(26 m x 26 m)	(29 m x 29m)
0.1	1.00E+00	1.01E+00	1.01E+00	1.01E+00
0.2	1.14E+00	1.23E+00	1.31E+00	1.42E+00
0.3	3.31E-01	3.47E-01	3.52E-01	3.55E-01
0.4	4.60E-01	4.19E-01	3.99E-01	3.81E-01
0.5	3.40E-01	3.04E-01	2.85E-01	2.68E-01
0.6	1.27E-01	1.11E-01	1.03E-01	9.55E-02
0.7	2.11E-02	1.83E-02	1.68E-02	1.55E-02
0.8	6.95E-03	6.36E-03	6.03E-03	5.74E-03
0.9	2.45E-02	2.20E-02	2.06E-02	1.94E-02
1	1.01E-02	9.16E-03	8.57E-03	8.06E-03
1.1	2.06E-03	1.75E-03	1.64E-03	1.55E-03
1.2	2.78E-03	2.49E-03	2.34E-03	2.20E-03
1.3	1.32E-03	1.13E-03	1.05E-03	9.93E-04
1.4	7.84E-05	1.09E-04	1.01E-04	9.52E-05
1.5	1.66E-04	1.18E-04	1.10E-04	1.02E-04
1.6	1.54E-05	7.00E-05	6.61E-05	6.28E-05
1.7	3.02E-05	6.54E-05	6.12E-05	5.76E-05
1.8	1.48E-05	6.43E-05	6.08E-05	5.78E-05
1.9	2.79E-05	9.45E-05	8.93E-05	8.49E-05
2.0	7.84E-06	2.80E-05	2.65E-05	2.52E-05
2.1	2.00E-05	9.72E-05	9.26E-05	8.87E-05

Appendix F

Raw Data of Heave Motion RAO of Semi-submersible with Added Cylinder Heave Plates with Different Size at Elevation -60 m

Wave Frequency (rad/s)	Heave Motion at Elevation – 60 m			
	Conventional	(29 m)	(30 m)	(31 m)
0.1	1.00E+00	1.01E+00	1.01E+00	1.01E+00
0.2	1.14E+00	1.29E+00	1.32E+00	1.34E+00
0.3	3.31E-01	3.51E-01	3.52E-01	3.53E-01
0.4	4.60E-01	4.03E-01	3.98E-01	3.94E-01
0.5	3.40E-01	2.89E-01	2.85E-01	2.80E-01
0.6	1.27E-01	1.05E-01	1.03E-01	1.01E-01
0.7	2.11E-02	1.71E-02	1.68E-02	1.64E-02
0.8	6.95E-03	6.10E-03	6.03E-03	5.95E-03
0.9	2.45E-02	2.09E-02	2.06E-02	2.03E-02
1	1.01E-02	8.70E-03	8.57E-03	8.43E-03
1.1	2.06E-03	1.66E-03	1.64E-03	1.61E-03
1.2	2.78E-03	2.37E-03	2.34E-03	2.30E-03
1.3	1.32E-03	1.07E-03	1.05E-03	1.04E-03
1.4	7.84E-05	1.03E-04	1.01E-04	9.97E-05
1.5	1.66E-04	1.12E-04	1.10E-04	1.08E-04
1.6	1.54E-05	6.70E-05	6.61E-05	6.52E-05
1.7	3.02E-05	6.22E-05	6.12E-05	6.02E-05
1.8	1.48E-05	6.16E-05	6.08E-05	6.00E-05
1.9	2.79E-05	9.05E-05	8.93E-05	8.81E-05
2.0	7.84E-06	2.69E-05	2.65E-05	2.62E-05
2.1	2.00E-05	9.37E-05	9.26E-05	9.15E-05

Appendix G

Raw Data of Heave Motion RAO of Semi-submersible with Added Hexagon Heave Plates with Different Size at Elevation -60 m

Wave Frequency (rad/s)	Heave Motion at Elevation – 60 m			
	Conventional	(29 m)	(30 m)	(31 m)
0.1	1.00E+00	1.01E+00	1.01E+00	1.01E+00
0.2	1.14E+00	1.37E+00	1.41E+00	1.45E+00
0.3	3.31E-01	3.54E-01	3.55E-01	3.56E-01
0.4	4.60E-01	3.89E-01	3.84E-01	3.78E-01
0.5	3.40E-01	2.76E-01	2.70E-01	2.65E-01
0.6	1.27E-01	9.88E-02	9.65E-02	9.41E-02
0.7	2.11E-02	1.60E-02	1.56E-02	1.52E-02
0.8	6.95E-03	5.87E-03	5.78E-03	5.68E-03
0.9	2.45E-02	2.00E-02	1.96E-02	1.92E-02
1	1.01E-02	8.29E-03	8.12E-03	7.95E-03
1.1	2.06E-03	1.59E-03	1.56E-03	1.53E-03
1.2	2.78E-03	2.26E-03	2.22E-03	2.17E-03
1.3	1.32E-03	1.02E-03	1.00E-03	9.81E-04
1.4	7.84E-05	9.81E-05	9.60E-05	9.39E-05
1.5	1.66E-04	1.06E-04	1.03E-04	1.01E-04
1.6	1.54E-05	6.43E-05	6.32E-05	6.21E-05
1.7	3.02E-05	5.93E-05	5.81E-05	5.68E-05
1.8	1.48E-05	5.92E-05	5.82E-05	5.71E-05
1.9	2.79E-05	8.69E-05	8.55E-05	8.39E-05
2.0	7.84E-06	2.58E-05	2.54E-05	2.50E-05
2.1	2.00E-05	9.05E-05	8.92E-05	8.79E-05