This is a peer-reviewed, accepted author manuscript of the following research conference paper: Akandu, E., Incecik, A., & Barltrop, N. (2015). How to determine the principal dimensions of FPSO vessel. In *Proceedings of the International Conference on Offshore Mechanics and Arctic Engineering - OMAE* (Vol. 1). [OMAE2015-42274] American Society of Mechanical Engineers (ASME). https://doi.org/10.1115/OMAE2015-42274

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

Ezebuchi Akandu,[a,*] Atilla Incecik,[a] and Nigel Barltrop,[a]

[a)] *Department of Naval Architecture, Ocean and Marine Engineering, University of Strathclyde, Glasgow, United Kingdom*

*Corresponding author: ezebuchi.akandu@strath.ac.uk

Paper History
Received:
Received in revised form:
Accepted:

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_{min}	Minimum separation of heave- and-pitch zeros
Γ	Gamma function

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 note-worthy 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 in comparism 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 underdeck 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^{o}; linear motions, 0 to 0.25g; periods, 3 to 15s [5].

Table 1: Wave Periods and Wavelengths for a Number of Sea Areas

Periods and Wavelengths Exceeded 2% of Time					
		Pierson-Moskowitz		JONSWAP	
Area	Tz	Tp [s]	[m]	Tp [s]	[m]
Central North Sea	8.7	12.3	236	11.2	196
Northern North Sea	10.9	15.4	370	14.1	310
West of Shetland	11.3	15.9	395	14.6	333
Brazil	10	14.1	324	12.9	260

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

Operator	Fields	Reserves (mmbbls)
Shell	Bonga	600
	Forcados-Yokri	1235
	Nembe Creek	950
Mobil	Erha	1200
	Ubit	945
Chevron Texaco	Agbami	1000
	Meren	1100

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 significant wave height and the wave period (which are in common use in wave description) for specified values of p and q (For Pierson-Moskowitz spectrum, p=5 and q=4). The n^{th} moment of the spectrum which is very useful in obtaining the wave characteristics is expressed as:

$$m_n = \int_0^\infty \omega^n S(\omega) d\omega = \frac{A}{q}\left[\frac{\Gamma[(p-n-1)/q]}{B^{[(p-n-1)/q]}}\right] \qquad (7)$$

The zeroth moment (n=0, $m_n = m_0$) or the variance of the wave elevation is defined as the area under the Spectral curve. The mean wave frequency $\bar{\omega}$ is the ratio of the first moment to the zeroth moment. The zero-crossing frequency ω_z is the square root of the ratio of the second moment to the zeroth moment. The spectral peak frequency can be obtained by differentiating $S(\omega)$ with respect to the wave frequency, ω and equating the result to zero. By substituting the expressions for A and B, the modified version of the wave spectrum is therefore obtained as:

$$S(\omega) = 124\frac{{H_s}^2}{{T_z}^4}\omega^{-5}\exp[-496.1(\omega T_z)^{-4}] \qquad (8)$$

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:

$$\left(M_{jk} + A_{jk}\right)\ddot{\eta}_k + d_{jk}\dot{\eta}_k + C_{jk}\eta_k = F_j \qquad (9)$$

Where: M_{jk} are the elements of the generalized mass matrix for the structure; A_{jk} are the elements of the added mass matrix; d_{jk} 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; η_k represents responses; $\dot{\eta}_k$ and $\ddot{\eta}_k$ are the velocity and acceleration terms; and ω 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:

$$C_{11} = w\left[cosh^{-1}\left(1+\frac{h}{a}\right) - 2\left(1+\frac{2a}{h}\right)^{-\frac{1}{2}}\right]^{-1} \qquad (10)$$

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 sinkage respectively.

$$C_{33} = \rho g BL \qquad (11)$$

$$C_{55} = Mg \times GM_L = \rho g LBT \times \frac{L^2}{12T} = \rho g B\frac{L^3}{12} \qquad (12)$$

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-breadth (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:

$$C_n = \mathrm{LBD} = \frac{L^3}{{x_b}^2 \times y_d} = \frac{B^3}{[y_d/x_b]} = \frac{\mathrm{D}^3}{[x_b \times {y_d}^2]^{-1}} = \frac{\nabla}{(T/\mathrm{D})} = \left(\frac{S_c}{C_f \times E_s}\right) \qquad (13)$$

From eqn. (1), it follows that:

$$\text{The Length, } \mathrm{L} = f_1\left(\frac{S_c}{C_f \times E_s}\right)^{1/3} \qquad (14)$$

$$\text{Breadth, B} = f_2\left(\frac{S_c}{C_f \times E_s}\right)^{1/3} \quad (15)$$

$$\text{Depth, D} = (f_1 f_2)^{-1}\left(\frac{S_c}{C_f \times E_s}\right)^{1/3} \quad (16)$$

$$\text{Draught, T} = z_m \text{D} \quad (17)$$

∇ is the displacement; and the new dimensionless factors are: $f_1 = [x_b{}^2 \times y_d]^{1/3}$; $f_2 = [y_d/x_b]^{1/3}$; $z_m = \nabla/C_n$

S_c: Required oil storage capacity in barrel (bbl); E_s: Oil storage Efficiency; and Conversion factor, $C_f = 6.28981077$; $6.28981077 bbl = 1 m^3$.

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_1, is usually taken to be approximately equal to the amplitude of the Froude-Krilov (pressure force), F_{FK1} as given in Eq. (18).

$$\left.\begin{aligned} \frac{F_1}{\zeta_a} &\approx 2\left(\frac{\rho g \text{B}}{\text{k}}\right)\left(1 - \text{e}^{-\text{kT}}\right)\sin\left(\frac{kL}{2}\right) \\ F_1 &\approx \rho g \zeta_a \left(\frac{\text{B}\lambda}{\pi}\right)\left(1 - \text{e}^{-2\pi\text{T}/\lambda}\right)\sin\left(\frac{\pi L}{\lambda}\right) \end{aligned}\right\} \quad (18)$$

Therefore, the Surge Response Amplitude Operator, RAO_1, is:

$$RAO_1 = \frac{F_1 Q_1}{C_{11}\zeta_a} = \frac{2Q_1}{C_{11}}\left(\frac{\rho g \text{B}}{\text{k}}\right)\left(1 - \text{e}^{-\text{kT}}\right)\sin\left(\frac{kL}{2}\right) \quad (19)$$

Q_1 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:

$$\phi = g\frac{\zeta_a}{\omega}e^{kz}\cos(\omega t + kx) \quad (20)$$

The vessel is divided into strips of equal sizes and the force acting on each strip (dF_3) 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.

$$dF_3 = pBdx + A_{33}^{(2D)} a_3 dx = \left(-\rho\frac{\partial\phi}{\partial t}\right)Bdx + A_{33}^{(2D)}\left(\frac{\partial^2\phi}{\partial z \partial t}\right)dx$$

$$= \zeta_a\left(\rho g B - A_{33}^{(2D)} k g\right)e^{-kT}\sin(\omega t + kx)\,dx$$

$$F_3 = \zeta_a\left(\rho g B - A_{33}^{(2D)} k g\right)e^{-kT}\int_{-\frac{L}{2}}^{\frac{L}{2}}\sin(\omega t + kx)\,dx$$

$$= 2\zeta_a\left(\frac{\rho g B}{k} - A_{33}^{(2D)} g\right)e^{-kT}\sin\left(\frac{kL}{2}\right)\sin(\omega t)$$

Where $A_{33}^{(2D)}$ is the 2-D added mass in heave, while the amplitude of the heave force is given by:

$$F_{3\text{a}} = 2\zeta_a\left[\frac{\rho g B}{k} - A_{33}^{(2D)}\text{g}\right]\left(\text{e}^{-\text{kT}}\right)sin\left(\frac{kL}{2}\right)$$

$$= \rho g \zeta_a\left[\left(\frac{\text{B}\lambda}{\pi}\right) - c_v\pi\left(\frac{B}{2}\right)^2\right]\left(\text{e}^{-\text{kT}}\right)\sin\left(\frac{\text{k}L}{2}\right) \quad (21)$$

Therefore, the Heave Response Amplitude Operator, RAO_3, defined as the heave amplitude per wave amplitude, is:

$$RAO_3 = \frac{F_{3\text{a}}Q_3}{C_{33}\zeta_a} = \frac{\rho g Q_3}{C_{33}}\left[\left(\frac{\text{B}\lambda}{\pi}\right) - c_v\pi\left(\frac{B}{2}\right)^2\right]\left(\text{e}^{-\text{kT}}\right)\sin\left(\frac{\text{k}L}{2}\right) \quad \ldots (22)$$

Q_3: Dynamic magnification factor in heave; λ: wavelength; c_v: virtual added mass coefficient in heave; ζ_a: wave amplitude; and wave number, $k = 2\pi/\lambda$.

Both heave force, $F_{3\text{a}}$ and response, RAO_3 will be equal to zero when $\left[\left(\frac{\text{B}\lambda}{\pi}\right) - c_v\pi\left(\frac{B}{2}\right)^2\right]$ or $sin\left(\frac{kL}{2}\right)$ is equal to zero. These happen at wavelengths of $\frac{c_v\pi^2 B}{4}$, 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:

$$F_{5\text{a}} = \rho g \zeta_a\left[\left(\frac{\text{B}\lambda}{\pi}\right) - c_v\pi\left(\frac{B}{2}\right)^2\right]\frac{\left(\text{e}^{-\text{kT}}\right)}{k}\left[\frac{kL}{2}\cos\left(\frac{kL}{2}\right) - sin\left(\frac{kL}{2}\right)\right] \quad \ldots (23)$$

So, the Pitch Response Amplitude Operator, RAO_5, defined as the pitch response amplitude per wave amplitude, is:

$$RAO_5 = \frac{F_{5\text{a}}Q_5}{C_{55}\zeta_a}$$

$$= \frac{\rho g Q_5}{C_{55}}\left[\left(\frac{\text{B}\lambda}{\pi}\right) - c_v\pi\left(\frac{B}{2}\right)^2\right]\frac{1}{k}\left[\frac{kL}{2}\cos\left(\frac{kL}{2}\right) - sin\left(\frac{kL}{2}\right)\right] \quad (24)$$

Q_5 is the dynamic magnification factor in pitch motion.

The pitch moment, $F_{5\text{a}}$ and its corresponding response, RAO_5 will be equal to zero if $\left[\left(\frac{\text{B}\lambda}{\pi}\right) - c_v\pi\left(\frac{B}{2}\right)^2\right]$ or $\left[\frac{kL}{2}\cos\left(\frac{kL}{2}\right) - sin\left(\frac{kL}{2}\right)\right]$ is equal to zero. These happen at wavelengths of $\frac{c_v\pi^2 B}{4}$, L/1.43, L/2.45, L/3.47, 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:

$$S_{min} = min\left(L - \frac{c_v\pi^2 B}{4}, \quad \frac{L}{1.43} - \frac{c_v\pi^2 B}{4}\right) \quad (25)$$

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, $\in$, at various wavelengths.

$$\in = \left(\frac{F_{5\text{a}}}{F_{3\text{a}}}\right)\left(\frac{RAO_5}{RAO_3}\right) \quad (26)$$

The wavelengths at which these energy coefficients, $\in$, tend to infinity are called critical wavelengths, λ_{cr}. It is important to note that these phenomena occur at heave zeros. That is, $\text{L}/\lambda_{cr} = N$, 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: $dQ_X = dF_E - dF_I - dF_R - dF_D$

$$dF_E = \zeta_a \left(\rho g B - A_{33}^{(2D)} kg\right) e^{-kT} \sin(\omega t + kx)\, dx$$
$$dF_I = \left[\left(\rho BT + A_{33}^{(2D)}\right) dx(\ddot{\eta}_3 - x\ddot{\eta}_5)\right]$$
$$dF_R = [(\rho g B dx)(\eta_3 - x\eta_5)]$$
$$dF_D = B_{33}^{(2D)} dx(\dot{\eta}_3 - x\dot{\eta}_5)$$

$$Q_X = \int_0^X (dF_E - dF_I - dF_R - dF_D)$$

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.

$$M_X = -\int_0^X x\, dQ_X$$

$$M_X = \zeta_a \left(\frac{\rho g B}{k} - A_{33}^{(2D)} g\right) \frac{e^{-kT}}{k} \left\{ \begin{array}{c} [kX\cos(kX) - \sin(kX)]\cos(\omega t) \\ +[1 - kX\sin(kX) - \cos(kX)]\sin(\omega t) \end{array} \right\} + \frac{1}{2}X^2 \left\{ \begin{array}{c} \left(\rho BT + A_{33}^{(2D)}\right)\left(\ddot{\eta}_3 - \frac{2}{3}X\ddot{\eta}_5\right) + B_{33}^{(2D)}\left(\dot{\eta}_3 - \frac{2}{3}X\dot{\eta}_5\right) \\ +\rho g B\left(\eta_3 - \frac{2}{3}X\eta_5\right) \end{array} \right\} \quad (27)$$

Let $\frac{M_X}{\zeta_a} = I_1 \sin(\omega t) + I_2 \cos(\omega t)$

$$I_1 = \left\{ \begin{array}{c} \left(\frac{\rho g B}{k} - A_{33}^{(2D)} g\right) \frac{e^{-kT}}{k} [1 - kX\sin(kX) - \cos(kX)] \\ -\frac{X^2}{2}\left(\rho BT + A_{33}^{(2D)}\right)\omega^2 RAO_3 \\ +\frac{X^3}{3} B_{33}^{(2D)} \omega\, RAO_5 \\ +\frac{X^2}{2} \rho g B\, RAO_3 \end{array} \right\} \quad (28)$$

$$I_2 = \left\{ \begin{array}{c} \left(\frac{\rho g B}{k} - A_{33}^{(2D)} g\right) \frac{e^{-kT}}{k} [kX\cos(kX) - \sin(kX)] \\ +\frac{X^3}{3}\left(\rho BT + A_{33}^{(2D)}\right)\omega^2 RAO_5 \\ +\frac{X^2}{2} B_{33}^{(2D)} \omega\, RAO_3 \\ -\frac{X^3}{3} \rho g B\, RAO_5 \end{array} \right\} \quad (29)$$

Where the damping factor is:

$$\xi_3 = \frac{B_{33}^{(2D)}}{\left(\rho BT + A_{33}^{(2D)}\right)\omega_{n3}}$$

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

$$\left(\frac{M_X}{\zeta_a}\right)_{amplitude} = \left({I_1}^2 + {I_2}^2\right)^{1/2} \quad (30)$$

5.0 RESULTS AND DISCUSSIONS

From the design 1 (see Table 3) analyses (which are executed in the PD programme with other subroutines and theoretically 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/s^2 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).

Table 3: Analysis of Rectangular Block Design for Oil Storage Capacity of 2 million barrels

Design 1: ($Note$: $S_{min} \approx 10$)						
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
Heave and Pitch Zeros	Heave Zeros			Pitch Zeros		
$\frac{c_v \pi^2 B}{4}$	L	$\frac{L}{2}$	$\frac{L}{3}$	$\frac{L}{1.43}$	$\frac{L}{2.45}$	$\frac{L}{3.47}$
204.5	306.4	153	102	214	125	88.3
Metacentric Heights and Natural Periods						
GM_L	KM_L	$T_{n3,5}$	λ_{n3}	GM_T	KM_T	T_{n4}
355	371	14.8	343	6.43	23.2	17.2
Required Oil Storage Capacity and Vessel Size						
Sc [Mbll]	Es [%]	M [t]	∇ [m^3]	Hull Area	Top Area	Trans Area
2	58	393356	383762	57679	17390	1789

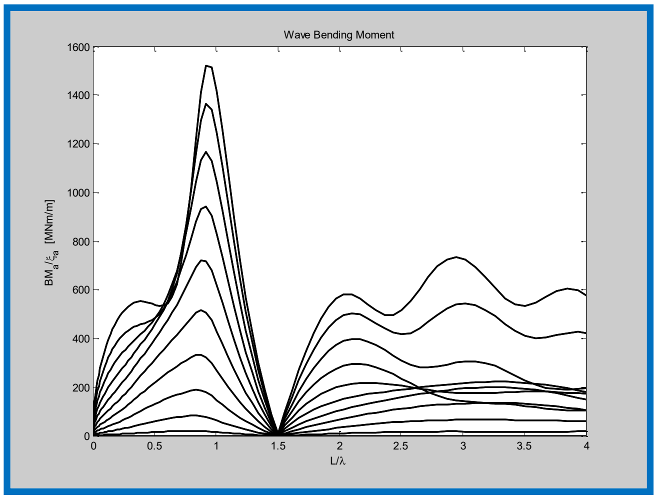

Figure 1: The Bending Moment per unit wave amplitude at various sections of the vessel from aft to amidships

Table 4: Comparism of predicted Wave Bending Moments [in MNm] with those of ABS and DNV (Vessel: Design 1)

	BM (Predicted)	**BM (ABS)**	**BM (DNV)**
Sagging	10105	10701	10256
Hogging	10812	10872	11282

Design 2 (See Table 5) shows a vessel of equivalent storage capacity as that of design 1 but with an improved minimum separation of heave and pitch zeros. Consequently, the dynamic performance is equally improved. This improvement is also notable in the reduction of the design wave bending moment at amidships as shown in Table 6.

Table 5: Design 2 showing improvement in S_{min}

Design 2: ($Note$: $S_{min} \approx 23$)						
L	B	T	L/B	B/D	B/T	c_v
256.9	54.7	25.4	4.7	1.4	2.2	1.5

Table 6: Comparism of predicted Wave Bending Moments [in MNm] with those of ABS and DNV (Vessel: Design 2)

	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.

ACKNOWLEDGEMENTS

The work reported here has been supported by the Federal Government of Nigeria through the **Petroleum Technology Development Fund**. The authors are grateful for this support.

REFERENCE

[1] Henery, D., and Inglis, R. B. (1995). "Prospects and challenges for the FPSO." *Offshore Technology Conference, OTC 7695, Houston, May*.

[2] Bensimon, L. F., and Delvin, P. V. (2001). "Technology gaps and preferred architectures for deepwater FPSOs." *Offshore Technology Conference, OTC 13088, Houston, May*.

[3] Hollister, H. D., and Spoke, J. J. (2004). "The Agbami project: A world class deepwater development." *Offshore Technology Conference, OTC 16987, Houston, May*

[4] D'Souza, R., Delepine, Y., and Cordy, A. "An Approach to the Design and Selection of a Cost-effective Floating Production Storage and Offloading System." *Proc., Offshore Technology Conference*.

[5] Miller, N. S. (1992). "Review of Reasons for Selecting a Floating Production System." *Advanced Design for Ships and Offshore Floating Systems*.

[6] Adebola, D. F., Okoro, J., and Nwasike, O. (2006). "Building local capability: a case study of Agbami project." *A paper presented at the 30th SPE Annual International Conference and Exhibition (NAICE 2006), Abuja, Nigeria 31st July–2nd August*.