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Applicability of a Toroidal Hull Structure for Floating Wind

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ABSTRACT

The growing maturity of offshore wind energy and in particular floating foundation to support Wind Turbine Generators (WTG) has allowed novel concepts to be examined which some years ago were considered unfeasible. One such concept is the use of a toroidal shaped hull structure as a pontoon for its enhanced motion keeping properties. This paper examines the current sub-structures in use and proposed for floating wind turbines. A hydrodynamic analysis is performed on a toroidal pontoon using numerical methods and the results presented. The merits of using a vertical axis WTG are also considered.

KEY WORDS: offshore, wind, renewable, energy, floating, toroidal

NOMENCLATURE

A_w : water plane area
A_s : sectional area
b_{33} : dampening coefficient for a torus in heave
B_{33} : high frequency approximation for the dampening coefficient for a torus in heave
g : acceleration due to gravity
F_{wave} : vertical force due to wave
H_w : wave height
K : wave number
g : acceleration due to gravity
k_y : hydrostatic stiffness in heave
R : radius of the torus
ζ_0 : wave amplitude
M : mass of structure
M_{AVM} : added mass of the complete structure
$M_{AVM,hull}$: added mass of the hull structure
$M_{AVM,col}$: added mass of the column structure
ρ : density of water
ω_n : natural frequency
ω_d : damped frequency
λ : Froude's scaling factor
Θ : incremental angle

INTRODUCTION

The floating hull concept is well proven in the oil and gas industry, namely the semi-submersible design. The main structure of the vessel or structure is located below the ocean surface giving a number of advantages over traditional structures, which usually have the hull form close to the water surface. For instance, the advantages of having the main structure below the water surface includes: reduced wave loads, (since the wave kinematics decay exponentially with depth) and longer natural periods of motion, hence a reduced response motion. Also the deep submergence of the pontoons combined with a structure made up of pontoons, columns and bracing yields the above characterization of low motion response to waves. However, the floating wind farm application requires considerably larger semi-submersible structures with deeper drafts and larger displacements. Given these characteristics it was found that waves contribute to the majority of the rigid-body motion-inducing dynamic loads of a large floating structure (Henderson, 1997). Technical challenges are related to minimizing the wave-induced motion and understanding the coupling between support structure and the wind turbine and achieving static and dynamic stability. The idea for the Multiple Unit Floating Offshore Wind Farm (MUFOW) concept was originally developed at UCL (University of College London) in the early 1990s (Musial, 2003). While it was concluded that such a support structure was not cost-effective to support wind farms, cost analysis was never done for the support structure of a 'hybrid' incorporating both wind and wave energy, which would most certainly increase the cost-effectiveness. Nonetheless, recent advancements in the Vertical Axis Wind Turbines (VAWT) and continued research in particular the benefits of combing the VAWT with a floater produces an improved case.

BACKGROUND

The main challenges for the design of floating foundations are limiting the accelerations of the rotor nacelle assembly (and thereby the motions of the platform), the mooring system design, dynamic electrical inter-array/export cable and higher sea states further offshore.

Floaters

The main floater technologies used in floating wind are outlined in the following subsections.

Semi-submersible
A number of large columns linked by connecting bracings / submerged pontoons. The columns provide the hydrostatic stability, and pontoons provide additional buoyancy. The foundation is kept in position by catenary or taut spread mooring lines and drag anchors. Principle Power (WindFloat) and Fukushima FORWARD had early Demonstrators in Portugal in 2011 (2 MW) and in Japan in 2013 (2 MW) and 2015 (7 MW). Ideol (Floatgen) planned demonstration in France in 2017 (2 MW) and has now been installed and connected to the electricity grid. Hexicon planned demonstrator in UK in 2018 using multi turbine (two ~5 MW turbines) on a single platform called Dounreay Tri however this project was cancelled. Other companies using the semi-submersible solution include Aerodyn, DCNS/GE, DeepCwind, Floating Power Plant, GustoMSC, NAUTILUS Floating Solutions, Nenuphar/ EDF, TetraFloat.

Spar-buoy
A cylinder with low water plane area, uses ballast to keep the centre of gravity below the centre of buoyancy. The foundation is kept in position by catenary or taut spread mooring lines with drag or suction anchors. Equinor, formerly Statoil (Hywind) successful demo in Norway and recent commissioning of the first commercial floating wind farm in world (5 x 6 MW array) has given this floater type an advantage in terms of maturity of technology Japan Marine United demonstrated in Japan in 2013 (advanced spar used to support floating substation) and in 2016 (5 MW). Other similar solutions include DeepWind, SeaTwirl, Windcrete.

Tension leg
Highly buoyant, with central column and arms connected to tensioned tendons which secure the foundation to the suction / piled anchors. The Germany company GICON demonstrated in Germany in 2016 using a 2.3 MW turbine. Glosten Associates also introduced the PelaStar and are currently seeking site for 6 MW demonstrator. Additional Blue H Group, DBD Systems, Iberdrola, Nautica Windpower each uses the TLP technology.

Toroidal Hull
The toroidal hull concept was first used by a German consortium (ERNO Raumfahrttechnik and Partners) in the new design of semi-submersible called the RS 35. The symmetrical arrangement was said to give good motion characteristics and eliminated the need for cross bracing. The project to design this new semi-submersible type was funded by the German Ministry of Science and Technology (Clauss, 1978) To give some idea of the scale of the structure, the ring-hull has an overall diameter of about 100 meters (m), each tubular section has a diameter of about 10m and the vertical supporting columns have a diameter of about 12m. In its operational mode the ring-hull is submerged to a depth of about 20 m (The Naval Architect, 1980).

HYDRODYNAMICS OF A TOROIDAL HULL

Added Mass
In the presence of plane progressive incident waves of small amplitude, the torus will perform small oscillatory motions with the same frequency in the horizontal direction (surge), in the vertical direction (heave) and about an axis perpendicular to these two directions (pitch). In the linear theory this motion can be decomposed into three radiation problems of forced oscillations in each mode, in otherwise calm water, plus a diffraction problem of waves that are incident upon the stationary body. Combining the hydro-dynamic pressure forces from each of these problems yields linearized equations of motion for the oscillations of the body in waves. The three radiation problems are solved by simple application of the method of matched asymptotic expansions. The damping and added- mass components for a torus are given as:

In heave

$$b_{33} \cong \pi c B_{33}$$

$$m_{33} \cong 2\pi c M_{33} \cong \left[\frac{(1-4)}{(3\pi K a) m} \right] \tag{1}$$

Where $m = \pi^2 \rho a^2 c$ is the mass of the fluid displaced by the torus and the high frequency approximation for B_{33} and M_{33} are derived by (Ursell, 1953). The added mass can be deduced from a simple strip theory, as the product of the two dimensional added mass and the circumference of the torus.

In surge:

$$m_{11} \cong \tfrac{1}{2} \pi^2 \rho a^2 c = \tfrac{1}{2} m \tag{2}$$

In the high frequency limit, similar results can be derived using the approximations:

$$B_{11} \cong 4 \rho \omega / K^2$$

$$M_{11} \cong \frac{2}{\pi} \rho a^2 \left[1 - \frac{4}{Ka} \left(\frac{\pi}{9} + \frac{1}{3\pi} \right) \right] \tag{3}$$

Equation (3) above defines the two-dimensional damping and added mass (Newman, J.,1978).

Forces
The forces due to waves are assumed to develop due to the variation in pressure due to the passage of the wave – the Froude-Krylov force and the inertia forces due to the effects of the acceleration of the particles within the wave on the added virtual mass of the body. It should be noted that the forces due to particle velocity effects are assumed to be negligible in comparison with the above forces. For each case the exciting force will be derived for a wave with a crest at the centre of circular hull. This will tend to produce a pure heaving force and viscous effects will be small. The equation to the surface wave will be:

$$y = \zeta_0 \cos(kx - \omega t), \quad \zeta_0 = 0.5 H_w \tag{4}$$

Now, heave response equation:

$$(M + M_{AVM,y}) \ddot{y} + c \dot{y} + ky = F_{WAVE} \tag{5}$$

The solution is given as:

$$y = \frac{(F_{wave} / K_{,y}) \cos(\omega t + \phi)}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n} \right)^2 \right]^2 + \left(\frac{2\omega}{\omega_n} \omega_d \right)^2}} \tag{6}$$

Where, K_y = Heave stiffness = $\rho g A_w$ = ρg x # of columns x $\frac{\pi d^2}{4}$

$$\omega_n = \sqrt{\frac{K_y}{M + M_{AVM}}} \tag{7}$$

M: submerged mass = density of water X (volume of hull + volume of columns)

$$M_{AVM} = M_{AVM,hull} - M_{AVM,col}$$

For circular hull, we get:

$$M^{heave}_{AVM,hull} = \int_0^{2\pi} m_{AVM,\theta} \times R \times d\theta \qquad (8)$$

$$F_{WAVE} = F_{HULL} - F_{COLUMN} \qquad (9)$$

$$F_{hull} = \int_{-L}^{L} \rho A_s a_y dx + \int_{-L}^{+L} M_{AVM,y} a_y dx \qquad (10)$$

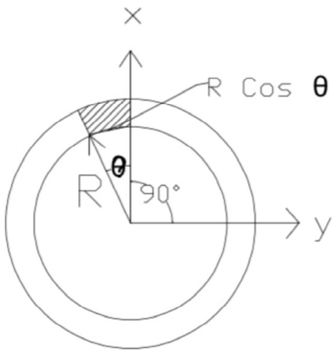

Figure 1 Schematic of a toroidal hull (plan view)

But dx = R.dΘ and x = R cosΘ

Equation (10) becomes:

$$F_{hull} = -2\pi R.\omega^2.\zeta_0.e^{-kz}.[\cos(KR.\cos.\theta)] \qquad (11)$$

From the formulae:

$$J_0(Z) = \frac{1}{2\pi}\int_0^{2\pi} \cos(Z\cos\theta)d\theta \qquad (12)$$

$$J_1(Z) = \frac{1}{2\pi}\int_0^{2\pi} \cos\theta \sin(Z\cos\theta)d\theta$$

Where $Z = K.R$, $K = \frac{\omega^2}{g}$ (13)

Note: The Bessel function of the first kind, equation (12) was used to evaluate terms in the equation (11). The order of the Bessel function, $J_0(Z)$ and $J_1(Z)$ were used to evaluate surge and pitch forces.

$$F_{COLUMN} = F_{PRESSURE} + F_{ACCELERATION} \qquad (14)$$

$$F_{PRESSURE} = \left(P_d \middle| A_s\right)^{x}_{-y} + \left(P_d \middle| A_s\right)^{-x}_{-y}$$

$$F_{ACCELERATION} = \left(M_{AVM,Y} \times a_y \Big|^{x}_{-y} + M_{AVM,Y} \times a_Y \Big|^{-x}_{-y} \right) \times 2$$

Now, since the pontoon is symmetrical about the z and x axis, heave forces in beam sea condition = heave forces in head sea condition.

Experimental Setup and Results

Reference is made to the experiment which was performed at Newcastle University, School of Marine Science and Technology Experimental Facilities. The combined wind, wave and current tank facility was used because of its ability to simulate the three environmental loadings thereby simulating a realistic environment.

The toroidal hull was applied to a combined wind and wave energy novel concept as depicted below in Figure 4, Figure 3 and Figure 4.

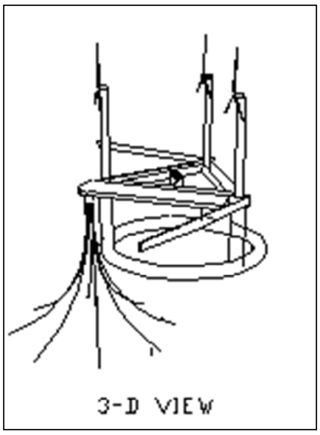

Figure 2: Sketch of the toroidal hull applied to a novel multi turbine-floating concept

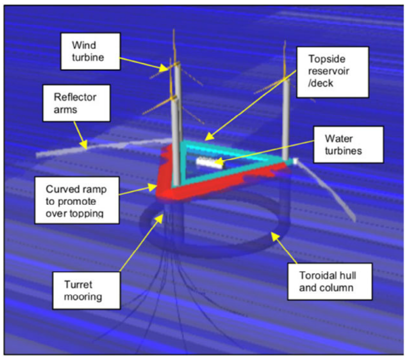

Figure 3: Rendered 3D view with the torus hull submerged.

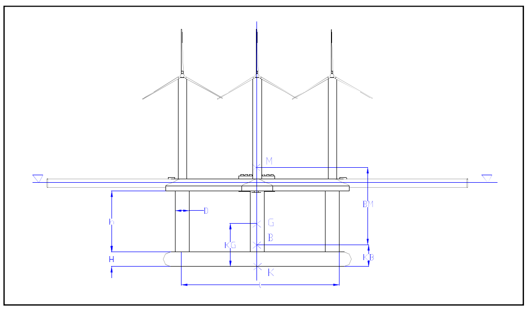

Figure 4: Front view of the novel concept showing stability geometry.

The experimental model was a 1/200 scaled rigid model which was made of PVC plastic, insulation foam, and polystyrene. For the convenience of fabrication, the open-deck triangular structure was modified to one solid triangular shape. Figure 5 below shows a setup of measuring apparatus. Experiment was conducted for an incident wave angle of 90 degrees. Mooring system was modeled as a three linear horizontal spring in which the spring constant of the spring arrangement was determined by experimental methods. Water depth of the wave tank was set at 1m, the incident waves with wave heights of 2cm, 4cm, and 8cm. Wave periods ranged from 0.8 to 4.0 seconds with 0.2 seconds interval.

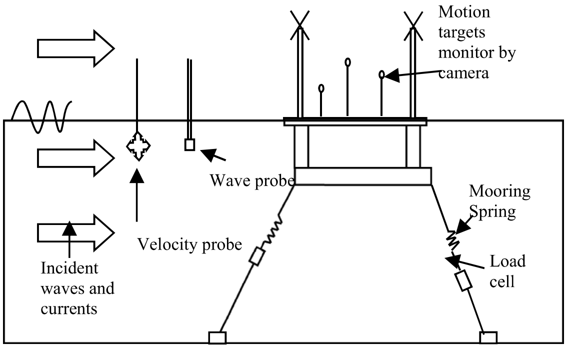

Figure 5: Typical experimental set up

The load cells were calibrated to read up to 25 N based on previous calculations. It was given an electronic input from the load cell and sent to a computer; the same computer also analyzed the motion tracking, while another computer operated the wave paddles of the tank. The Qualisys motion tracking system was used which is a 3-D positioning system for analyzing ship stability and hydrodynamics. Motions of the experimental model are determined as transforming motion of four reflective markers on the deck of the model, which were monitored using two infrared cameras, into displacements in six degrees of freedom by means of motion picture analysis. The water particle velocity was measured using a ventrino+ velocity probe, a capacity probe for measuring the wave motion and three load cells for measuring the load in the forward, port and starboard of the mooring lines. The unprocessed data is shown graphically in Figure 6.

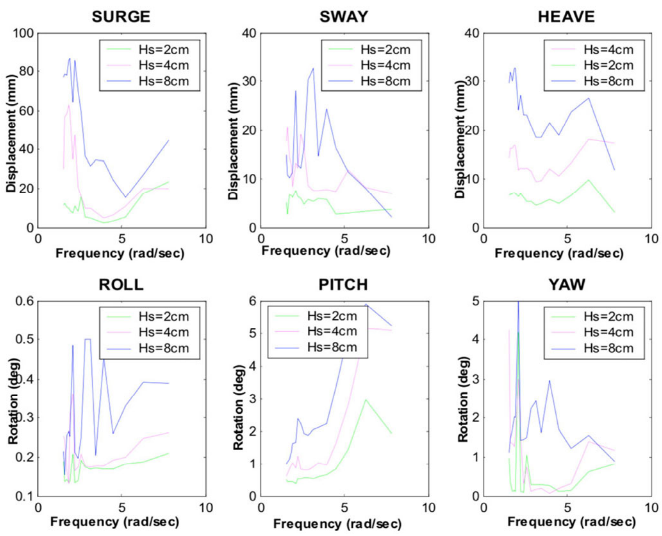

Figure 6: Motion response for various wave heights.

Figure 7 and Figure 8 shows a comparison between the theoretically calculated motion and the experimentally measured response amplitude operator plotted against frequency for heave and pitch respectively. Similar results were obtained for the surge motion; however, in the interest of clarity the surge motion has been omitted.

At low frequencies the Froude-Krylov effect governs the loading and the hydrostatic stiffness determines the response, so the structure follows the water surface, i.e. the RAO (amplitude of structure motion/amplitude of water motion) tends to the value one. As the frequency increases the wave shortens and as the wavelength reduces to that of the structure, the overall buoyancy force cancels out over the length of the structure so the RAO reduces to close to zero.

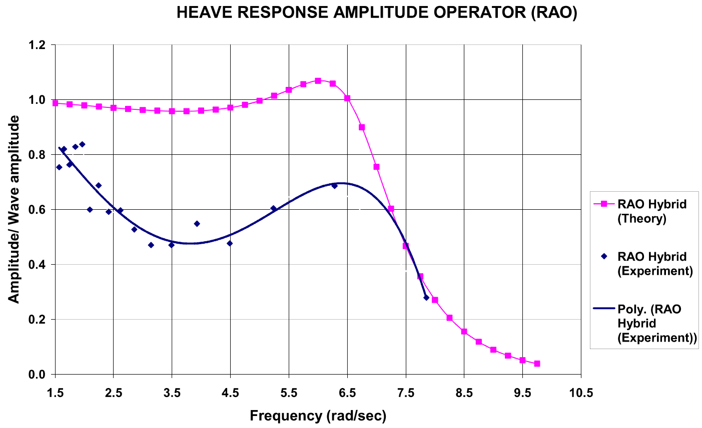

Figure 7: Heave RAO

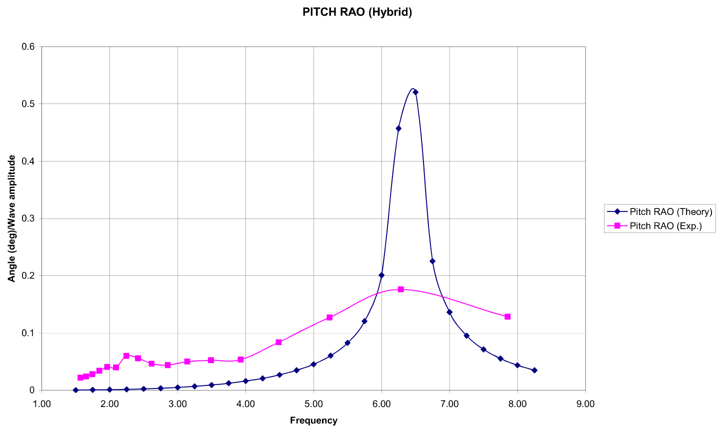

Figure 8: Pitch RAO

The torus is unique in several respects. The axisymmetric geometry is a major simplification, which enables a short wave asymptotic theory to be derived from a simple matching argument. The results depend principally upon the two-dimensional damping and added mass coefficients of the circular section. The absence of body ends and the curved form of the axis are significant, particularly in the scattering problem.

The theoretical model used in the calculation showed a correlation to the experimental response depicted in Figure 7 and Figure 8. The viscous damping was ignored but has a significant contribution in obtaining accurate results. Due to model construction limitations, the torus hull in the experimental model had a rather rough surface finish, this would likely give an increased viscous damping which could justify the difference in magnitude seen in the heave RAO.

CONCLUSIONS

During the evolution of the floater concepts for offshore wind, a number of solutions have emerged and each one has its own merits for the given environment and application. It can however be inferred that the trend for some concepts seems to converge into a triangular or square shaped hull. Considering only the hydrodynamic and not the application, the toroidal shape due to the asymmetrical reduces the forces imposed. The calculated heave response showed a magnification factor of slightly less than one, increasing slightly at the frequency 6.0 rad/sec. The low motion response of the toroidal hull offers a unique solution to platform design when applied to floating wind and in particular to the VAWT. The challenge with the torus hull will be the economical fabrication in an industry has become very competitive; nonetheless, awareness of the possible solution is a first step.

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