A STUDY ON INFLUENCE OF HEAVE PLATE ON DYNAMIC RESPONSE OF FLOATING OFFSHORE WIND TURBINE SYSTEM

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ABSTRACT: Offshore wind farms offer a sustainable source of renewable energy. In deep sea these wind turbines need to placed on floating systems to improve economy. Modeling principles for these floaters are usually borrowed from offshore oil and gas industry. However, knowledge of coupled floater and wind turbine behavior needs to be investigated. In this study, a water tank experiment is performed on a semisubmersible floating single wind turbine model to investigate influence of heave plate on dynamic response. It is found that use of heave plates increases natural period of heave, resulting in reduced heave response at rated and extreme sea states. Further, a FEM code is used to predict dynamic response for the experimental model. It has been observed that for estimation of restoring force, considering simple hydrostatic model fails to model heave response of floating system, whereas non-hydrostatic model is able to provide good agreement. The performance of non-hydrostatic model has been checked for large floating systems and provides appropriate results. Finally, the significance of modeling mooring system has been emphasized considering experiment setup in the current research.

KEYWORDS: Floating wind turbine system, Heave plate, Non-hydrostatic force, Morison’s equation, FEM.

1. INTRODUCTION

Wind energy is one of the most renowned sources of renewable energy. Onshore wind resources are limited and little land is available for large wind farms. Offshore wind energy offers obvious advantages of no land usage and a more reliable wind resource. Offshore wind energy development is thus a great prospect for power generation. In shallow waters, bottom-mounted support is economically feasible; however floating support systems are essential in deep waters around the Tokyo (Kanto) area of Japan (Figure 1). Technology to develop offshore platform for wind turbines has been borrowed from existing oil and gas (O&G) industry. Compared with O&G structures, floating wind turbines are much lighter and dynamic behaviour of floating system would be different because of heavy nacelle and blade at tower top.
Further, the floater itself has to be lighter to improve economy. Therefore, a new technology is still needed to make floating offshore wind energy economically competitive.

Several concepts ([1], [2], [3], [4], [5], [6]) of floating wind turbine systems have been proposed based on technologies in O&G industry such as semi-submersible, spar and tension leg platform. All these studies are numerical in nature and lack the validation through experiment. Recently, Ishihara and Phuc ([7], [8]) have validated their results through experiment for semi-submersible type floater system supporting three wind turbines. They ([7], [8]) investigated the influence of aerodynamic and hydrodynamic damping and effect of elastic deformations owing to a lighter floater. Their prototype, however impose a strong wake influence on the wind turbines. Therefore, using a single wind turbine floater is essential and heave plates can be used to obtain results comparable with larger floaters.

The present study considers a semi-submersible type floater with single wind turbine to investigate the influence of heave plates on dynamic response of floaters. A sophisticated FEM code that considers a non-hydrostatic model for restoring force is used in this study and its efficiency is validated through comparison with hydrostatic model and experiments. Finally, significance of considering full coupling between the floater and mooring system is investigated. Section 2 provides information about model detail and experiment setup used in this study, section 3 explains the numerical model and the finite element code. Section 4 includes results and discussion and finally conclusion are made in section 5.

2. WATER TANK EXPERIMENT

2.1. Model Detail

<table>
<thead>
<tr>
<th>Description</th>
<th>Prototype (m)</th>
<th>Model (cm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Span</td>
<td>60.0</td>
<td>60.0</td>
</tr>
<tr>
<td>Floater base Diameter</td>
<td>8.0</td>
<td>8.0</td>
</tr>
<tr>
<td>Floater Top Diameter</td>
<td>6.0</td>
<td>6.0</td>
</tr>
<tr>
<td>Heave plate diameter</td>
<td>12.0, 16.0</td>
<td>12.0, 16.0</td>
</tr>
<tr>
<td>Tower Height</td>
<td>70.0</td>
<td>70.0</td>
</tr>
</tbody>
</table>

Figure 1 Offshore Wind Resource near Tokyo [9]

Figure 2 Floater Model

Considering Froude’s Number similarity and limitations of testing facility [10], 1:100 scale has been selected for experiment model. Submerged depth of model is 20 m and total water depth of the tank is 150 m. The floating wind turbine model is shown in Figure 2. Table 1 gives the floater details and the heave plate sizes considered. Six set of experiments are performed considering three floater plates of 8m,
12m and 16 m diameter (FP8, FP12 and FP16) and Wave heights of 4 m and 12 m (H4 and H12) corresponding to rated and extreme sea condition respectively. The case FP8 case corresponds to no heave plate while cases FP12 and FP16 correspond to cases with heave plates. Linear springs are used to model mooring stiffness of the floater. As this is a unidirectional wave experiment and the mooring stiffness is heave direction is very small, only mooring stiffness in surge direction is considered. Wind turbine blades are therefore not considered and weight of rotor is included in the nacelle.

2.2. Model Layout

The experiment has been performed in pulsating wind tunnel with water tank facility of the National Marine Research Institute, Tokyo, Japan [10]. Figure 3 shows the model layout in water tank. The model is placed at mid span of the tank. It is connected to elastic bands through Kevlar thread which is connected to tension meter. An initial tension of 2.95 N is setup in elastic bands to obtain the required mooring stiffness of 45.0 N/m in surge direction. Displacements are measured from a four-legged LED target fixed to the wind turbine tower using CCD camera. Wind is not considered in this experiment, since the influence of aerodynamic damping is already discussed in previous studies ([7], [8]).

3. NUMERICAL MODEL

3.1. Equation of Motion

The general formulation of equation of motion for the floater system can be written as

\[
[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{F\} \quad \text{where} \quad \{F\} = \{F_e\} + \{F_H\} + \{F_G\} + \{F_W\}
\]

Where \{X\}, \{\dot{X}\} and \{\ddot{X}\} are unknown displacements in the six DOF and their time derivatives. [M] is mass matrix, [C] is damping matrix and [K] is stiffness matrix of the structure. \{F\} is total external force vector changing with time, where \{F_e\} is the wave exciting force acting on the floater, \{F_H\} is hydrostatic restoring force, \{F_G\} is mooring force and \{F_W\} is aerodynamic force acting on wind turbine. As the experiment is performed without wind, modeling of aerodynamic force shall not be discussed hereafter.

3.2. External Forces

a). Hydrodynamic Force Model

Morison’s equation [11] is well known for estimation of wave exciting force on slender bottom mounted cylinders. The equation assumes disturbing force to be composed of inertia and drag force linearly added together and is usually applicable when structure is small compared to wavelength. For a floating
structure, free to oscillate in the presence of waves and current, Morison equation is modified by Sarkaya et al. [12] as follows:

\[ \{F_e\} = \{F_{EM}\} + \{F_{EW}\} + \{F_{ED}\} \]

\[ \{F_{EM}\} = -[M_a]\{X\}, \quad \{F_{EW}\} = \rho_w C_a V \{\ddot{u}\}, \quad M_a = \rho_w (C_m - 1)V \]

\[ \{F_{ED}\} = 0.50 \rho_w C_D A\left|u - \dot{X}\right| \left|u - \dot{X}\right| \Rightarrow \{F_{ED}\} = 0.5 \rho_w C_D a \Delta L (u - \dot{X}) \quad C_C = \frac{8}{3\pi} \frac{C_D}{C_O} \left|u - \dot{X}\right|_{\text{max}} \]

In equation (2) first term ‘\(F_{EM}\)’ is added inertia force, second term ‘\(F_{EW}\)’ is in proportion to wave particle acceleration and ‘\(F_{ED}\)’ is drag force. In equation (3) and (4), ‘\(\rho_w\)’ is density of water, ‘\(M_a\)’ is added mass. ‘\(A\)’ and ‘\(V\)’ are characteristic area and volume for buoyancy. ‘\(X\)’ is velocity of moving element, ‘\(\Delta L\)’ is submerged length of element, ‘\(u\)’ and ‘\(\ddot{u}\)’ are wave particle velocity and acceleration. ‘\(C_D\)’ and ‘\(C_M\)’ are hydrodynamic drag and inertia coefficients respectively. For detail regarding estimation these coefficients reference is made to Phuc and Ishihara [8]. Since relative wave particle velocity as well as first term of wave exciting force contains velocity of moving element, hydrodynamic damping is automatically taken into account.

As the Morison’s equation cannot predict the vertical force on base of submerged cylinder, an added inertial force proposed by Haslum [13] as shown in equation (5) is included for numerical simulation. Where ‘\(D\)’ is diameter of the cylinder base. In this study, a linear damping ratio of 15% is used following Srinivasan [14]. The damping force as effect from the bottom of vertical cylinder is defined by equation (6), where \(\{X\}\) is the vector containing velocity of moving element in all direction and \([C_{ED}]\) is a damping matrix. This damping force can be modeled by a pseudo structural damping of the system and equation (4) can be written as equation (6). \([C_{ED}]\) the hydrodynamic damping matrix is estimated using Raleigh’s damping model.

\[ M_a = \rho_w (C_M - 1) \frac{2\pi}{3} \left(D/2\right)^3, \quad C_M = 2.0 \]

\[ \{F_{ED}\} = -[C_{ED}]\{X\} \]

**b) Restoring Force Model**

In order to estimate the restoring force due to vertical displacement of the floater, a non-hydrostatic model [NHM] is proposed and its efficiency w.r.t simple hydrostatic model [HM] has been verified through experiment. The restoring force discussed in equation (1) can be estimated using these two models as fellows:

\[ \{F_R\}_{HM} = -[K_R]\{X\} \quad \{F_R\}_{NHM} = -[K_R]\{X\} - \{\eta\} \]

\[ [K_R] = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -\rho_w g A_w & 0 & 0 \\ 0 & 0 & -W \times G M_X & 0 & 0 \\ 0 & 0 & 0 & -W \times G M_Y & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \]

Where \([K_R]\) is first order hydrostatic restoring force coefficient [15], \(\{X\}\) is displacement vector and \(\{\eta\}\) is wave elevation vector, which only has non-zero element in vertical direction. \(g\) is acceleration due to gravity, \(A_w\) is the cross-sectional area, \(W\) is the weight of the model, \(G M_X\) and \(G M_Y\) are meta-centric height in X- and Y-directions respectively.
c). **Mooring Force Model**

The mooring force has been estimated using a linear stiffness; the force is estimated as follows:

\[
F_G = [K_G] \{X\}
\]

Where \([K_G]\) is mooring stiffness, in the experiment 4 elastic bands are used that have a combined stiffness of \(45.0 \text{ N/m}\).

### 3.3. Numerical Scheme

For numerical solution rewriting equation (1) as:

\[
([M] + [M_E])\{\ddot{X}\} + ([C] + [C_{ED}])\{\dot{X}\} + ([K] + [K_G])\{X\} = \{F_E\}
\]

A FEM code based on equation (10) has been developed in time domain to predict dynamic response of floating offshore wind turbine system. Numerical integration is carried out using Newmark-\(\beta\) method \((\beta = 1/4)\). For damping, the Rayleigh's damping model is used represented as fellows:

\[
[C] = \alpha[M] + \beta[K] ; \alpha = 2\omega_1\omega_2 \left(\frac{\omega_1\zeta_2 - \omega_2\zeta_1}{\omega_1^2 - \omega_2^2}\right), \quad \beta = \left(\frac{\omega_1\zeta_1 - \omega_2\zeta_2}{\omega_1^2 - \omega_2^2}\right)
\]

Where \(\omega_1, \omega_2\) and \(\zeta_1, \zeta_2\) are natural frequency and damping for the 1\textsuperscript{st} and 2\textsuperscript{nd} mode. In this study, a structural damping of 0.50% is considered. Further details regarding the FEM code are presented in Table 2.

<table>
<thead>
<tr>
<th>Dynamic Analysis</th>
<th>Direct Numerical Integration (Newmark-(\beta))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Eigen-value Analysis</td>
<td>Subspace Iteration Procedure</td>
</tr>
<tr>
<td>Element Type</td>
<td>Beam (12-DOF), Truss (4-DOF), Spring(1-DOF)</td>
</tr>
<tr>
<td>Formulation</td>
<td>Total Lagrangian formulation</td>
</tr>
<tr>
<td>Damping Estimation</td>
<td>Rayleigh Damping</td>
</tr>
</tbody>
</table>

### 3.4. Finite element model of Experiment Setup

The finite element model for the scaled model of floating offshore wind turbine has been developed. Referring to Figure 3, the floater is modeled using beam elements having 122 elements and 114 nodes. The elastic bands are modeled as linear springs and the Kevlar thread is modeled using truss elements. The whole system has 142 nodes and 150 elements. The mass of the model is considered to be concentrated on nodes, which also incorporates difference in mass due to different floater plates (FP8, FP12 and FP16). The effect of different heave plates on buoyancy is considered by using equation (5) to estimate the added mass. In the dynamic analysis, \(C_D\) and \(C_M\) the hydrodynamic drag and inertia coefficients have been considered as functions of Kevlegan-Carpenter number \(\textit{KC}\) as recommended by Offshore Standard DNV-OS-J101 [16]. \(\textit{KC}\) has been simplified for incident wave height \(\textit{H}\) and diameter of column \(\textit{D}\) by the equation (12). Therefore, 6 FE-models have been prepared considering three plate sizes (FP8, FP12 and FP16) and two wave heights (H4 and H12). Another set of three models have been developed for each floater plate size, which are to be used in free vibration test. In these models, \(C_D\) and \(C_M\) are estimated based on element velocity \(\textit{U}\) and period \(\textit{T}\) from the free vibration experiment in surge, heave and pitch.
\[ K_C = \frac{U T}{D} = \pi \frac{H}{D} \]

The wave particle velocity and accelerations are generated through linear ‘Airy’ wave theory, simulations were also carried out for nonlinear wave using Stream function theory, as the results were identical therefore the ones from linear theory will be discussed.

4. RESULTS AND DISCUSSION

4.1. Free Vibration

Free vibration tests have been carried out during experiment to estimate natural periods and system damping of the floating wind turbine system in Surge and Heave degree of freedom. In a similar fashion free vibration test was simulated to check the accordance of FE-model with experiment model. Figure 3 shows comparison of observed and predicted natural period w.r.t prototype scale. The error bars represent variation in experimental results over a set of 20 observations. The maximum difference in predicted and observed is about 5%. The FE-models are therefore, in acceptable agreement with the actual experiment models.

![Comparison of observed and predicted natural periods](image)

Figure 4 Comparison of observed and predicted natural periods

4.2. Response of Floating Offshore Wind Turbine

In view of limitation of the testing facility [10], the unidirectional wave experiment has been carried out over a wave period range 0.60 ~ 3.00 sec (prototype scale 6.0 ~ 30.0 sec) at an interval of 0.20 sec for the three floater models and two wave heights. This range of wave period comfortably covers the rated condition \((T = 7.4 \text{ sec})\) and extreme condition \((T = 13.4 \text{ sec})\). The same range of wave period has been considered in the simulation. Surge and heave amplitudes are normalized w.r.t. wave amplitude ‘H/2’ whereas the pitch is normalized with respect to wave amplitude ‘H/2’ and wave length ‘L’ w.r.t rated condition \((H=4.0 \text{ m}, T=7.40 \text{ sec})\) and extreme condition \((H=12.0 \text{ m}, T=14.0 \text{ sec})\) for wave height of 4.0 m and 12.0 m respectively. The time period has been reverted to prototype scale to compare results with three wind turbine floater system of Ishihara [7] and Phuc et al. [8]. Wave height of 4.0 m and 12.0 m correspond to 2.667 and 8.0 cm using 1:150 scale to simulate response of the 3WT-floater for discussion in sub-section b).

a). Validation of Non-Hydrostatic Model

Reduction in floater size shall cause the non-hydrostatic effect discussed in equation (7) on restoring force to become more and more dominant. As the model discussed in equation (7) only influence the restoring force in vertical direction. Therefore, response predicted by the two models in surge mode is almost same
and only heave response prediction shall be discussed in this section. Figure 5 (a) & (b) show the comparison of the hydrostatic [HM] and non-hydrostatic [NHM] model for the original model FP8 (without heave plates) for 4.0 m and 12.0 m wave heights respectively. The results indicate that hydrostatic model underestimate heave response after natural period (12.50 sec), whereas the non-hydrostatic model gives essentially good agreement with experiment. The difference is owing to underestimation of restoring force by hydrostatic model as represented in Figure 6 for both wave heights. The Non-hydrostatic model is therefore clearly the better choice for response prediction in the current study.

![Figure 5 Comparison of Hydrostatic [HM] and Non-Hydrostatic [NHM] Model for Floater Model FP8](image)

(a) Wave Height = 4.0 m  (b) Wave Height = 12.0 m

Figure 5 Comparison of Hydrostatic [HM] and Non-Hydrostatic [NHM] Model for Floater Model FP8

![Figure 6 Restoring Force prediction in HM and NHM models for Floater Model FP8](image)

(a) Wave Height = 4.0 m  (b) Wave Height = 12.0 m

Figure 6 Restoring Force prediction in HM and NHM models for Floater Model FP8

**b). Influence of Heave plates**

As size of the floater is reduced, its response under wave force is increased. In order to reduce this response, influence of heave plates have been investigated. The purpose is to investigate if the response for single wind turbine floater be reduced to similar magnitude as the multi-turbine floater ([7],[8]) Two heave plate sizes, with 1.5 and 2.0 times diameter ([FP12] and [FP16] respectively) of original floater [FP8] have been considered. Figure 7 and Figure 8 show surge and heave response of the three models for wave height of 4.0 and 12.0 m respectively. Experiment data is plotted with marks and simulation results using non-hydrostatic model [NHM] are plotted with solid lines. FP8, FP12 and FP16 are represented by red, blue and green colours respectively. Along with these, results for three wind turbine semi-submersible floater [3WT] ([8]) have been plotted to compare response for the two floaters systems.

![Figure 7 Surge Response for Floater Models](image)

(a) Wave Height = 4.0 m  (b) Wave Height = 12.0 m

Figure 7 Surge Response for Floater Models

![Figure 8 Heave Response for Floater Models](image)

(a) Wave Height = 4.0 m  (b) Wave Height = 12.0 m

Figure 8 Heave Response for Floater Models
The surge response prediction show similar trends for the three plate sizes (FP8, FP12 and FP16), but indicate different response peak corresponding to natural frequency. This effect is not captured in the experiment results which can be because of reflected wave and wave disturbances at higher periods during experiment. The heave response prediction also shows good agreement with experiment and captures the shift in resonance peak with increase in plate diameter.

As can be seen in case of both wave heights, heave plates does not any significant influence on the surge response of the floater however they do effect heave response. The heave plates increase natural period of floater and resonance peak shifts towards higher periods. This shift does not have a significant influence on rated state (Figure 7) but for extreme state (Figure 8), the heave response is reduced to less than 50% for FP16 and is almost similar to the response predicted for much larger floater in Ishihara [7] and Phuc et al. [8]. This indicates that the presented floating offshore wind turbine prototype with heave plates can provide similar stability has is obtained in larger multi-wind turbine floater systems.

c). Influence of Mooring System

To investigate the influence of mooring system on floater response while being able to validate with experiment, a ‘Simplified’ FE-model of the experimental set up has been developed considering mooring stiffness ‘K’ only as a boundary condition instead of the actual elastic bands and the Kevlar thread
arrangement. This simplification ignores the wave loading on the mooring system (thread and bands) and the effect of its motion on mooring stiffness. The line tension for this model is estimated using the surge displacement ‘X’ of floater and the spring stiffness ‘k’ as ‘k.X’. The results for this ‘Simplified’ model and the ‘Complete’ FE-model used in sections a) and a). The two types of FE-models have identical results in surge and heave mode that are not show here. This indicates that the external force on the mooring system does not have any significant influence on floater response. Figure 9(a) shows the pitch response for FP8 and FP12 for the two types of FE-models at wave height of 4.0 m. The response has been normalized with respect to wave amplitude ‘H/2’ and wave length (rated stated) ‘L’. As can be observed the ‘Complete’ model provides better comparison with experiment than ‘Simplified’ model. This is because the ‘Simplified’ model ignores any change in mooring stiffness with floater motion, whereas in reality stiffness is changed as the line inclination change due to floater motion. Figure 9(b) shows the comparison of line tension for the same cases. The tension has been normalized with respect to wave height as it shows slight variations in experiment. The ‘Complete’ model again shows good comparison whereas the ‘Simplified’ model overestimates the line tension. The line tension from experiment at longer periods (>25 sec) show a different trend from simulation as the wave height distribution in the tank becomes non-uniform because of wave reflection from tank walls. A better comparison with ‘Complete’ model indicates that even such simple mooring arrangement can have significant influence on floater response and therefore for accurate prediction of floater response, mooring systems needs to be considered in detail.

5. CONCLUSIONS

A sophisticated FEM code that considers coupled dynamic response of floater, wind turbine and mooring system using Morison’s equation and Srinivasan’s model and non-hydrostatic model for restoring is developed. This code is used to investigate the influence of heave plates on the dynamic response of semi-submersible type single wind turbine floater. Two heave plate sizes have also been considered under wave height for rated and extreme state. To validate the predicted results, series of water tank experiment are performed on 1:100 scale models. It is observed that heave plates causes resonance peak to move towards longer periods, thus decreasing the response at rated and extreme states. Single wind turbine floater with heave plates for can provide similar stability as observed in multi-turbine floaters.

Numerical prediction of response shows that non-hydrostatic model shows good agreement whereas hydrostatic model fails to show agreement for heave. The performance of the non-hydrostatic model is also found to be adequate for large floaters. Finally, the influence of considering actual mooring system instead of simplified mooring stiffness boundary condition is discussed and it is found that the simplification reduces the prediction accuracy and should not be considered.
REFERENCE


