Evaluation of a low-fidelity hydrodynamic modelling approach for a floating wind turbine mounted on an enhanced spar

Guido Grassi González¹, Albert Meseguer Urbán¹, Sergio González Horcas¹, Laura Vollá I Roqueta¹ and Sergio Hernández Blanco²

¹ Department of Wind Energy, Technical University of Denmark (DTU), Frederiksborgvej 399, 4000 Roskilde, Denmark
² BlueNewables, C/ El Vizconde 7, chio 38670 (Adeje), Santa Cruz De Tenerife

E-mail: sgho@dtu.dk

Abstract. The present work revolves around the numerical simulation of floating offshore wind turbines, a promising technology for energy harnessing in deep water conditions. A reference 10MW wind turbine is studied, mounted on a commercial enhanced spar buoy, referred to as the WIND-bos platform. The focus is put on the hydrodynamic modeling. In particular, the Morison equation is used, accounting for a fixed set of coefficients. Those coefficients are initially estimated based on the literature, and subsequently calibrated through the comparison with experimental results on a scaled geometry. This allows to assess the modeling capabilities of the Morison approach, together with its challenges and limitations. Larger discrepancies between the numerical model and the experiments were assumed to be related to the geometrical particularities of the floating platform. In particular, the studied structure accounts for both large and slender members, potentially limiting the applicability of the method. It is shown that the deviations could be attributed to the frequency-variation of parameters such as the added mass, the radiation damping and viscous damping. Therefore, it is concluded that the agreement between the numerical model and the experiments could be improved by re-calibrating the coefficients for each of the studied sea states.

1. Introduction
The need for the deployment of a sustainable grid for energy production has become evident for most of the concerned organizations. For instance, the European Union has committed to climate-neutrality by 2050 [1] and has identified offshore wind energy as a pillar technology. In order to achieve these goals, the installed capacity needs to grow from the current 20GW to 450GW [2]. Exploiting deep waters becomes necessary for such a challenge, motivating the industry and academia to investigate innovative ways for harnessing energy from the wind. Of particular relevance to the present work is the ensemble of technologies relying on wind turbines mounted on floating substructures, which are often referred to as simply floating wind.

Scaled experiments are often used to dimension the floating substructure, taking also into account the presence of the wind turbine [3, 4]. The main objective of such tests is the hydrodynamic characterization of the system. This is usually achieved through the computation of the response amplitude operators (RAOs) of the assembly, a series of the so-called decay tests, or both [5, 6]. It is also possible to model the aerodynamic coupling by scaling the wind...
turbine along with a wind-generator device [7], or by using an actuator that can reproduce the aerodynamic loading [8, 9]. The results of the scaled tests allow the designers to have a first estimation of the load envelope expected during operation, and of the overall dynamics of the system. However, this data set is far from being comprehensive in terms of design constraints. Indeed, the scaled experiments do not take into consideration the whole range of expected operational configurations of the wind turbine, or the wide variety of expected atmospheric conditions and sea states. That is why the experimental results are subsequently used for the validation and the calibration of full-scale hydrodynamic numerical models.

During the design stage several hydrodynamic models are often used, ranging in fidelity and computational cost. The most commonly used approaches can be grouped as: computational fluid dynamics (CFD) methods, potential flow theory and the Morison equation [10, 11]. We refer as CFD to the ensemble of techniques that rely on the numerical solution of the Navier Stokes equations in a control volume. Through the careful choice of the appropriate modeling (i.e. with regards to the turbulence model), CFD methods allow to properly account for diffraction-radiation effects, inertial effects and viscous effects. However, the significant computational resources required for those approaches often limit their application to a few selected critical cases. On the other side of the computational cost spectrum, there are the techniques relying on potential flow theory. Those methods are capable of capturing the frequency-variation of hydrodynamic coefficients as the added mass, radiation damping, and exciting wave loading. However, the use potential flow theory also implies a certain degree of simplification, since the hydrodynamics of the system are often linearized around a particular position, and contributions such as the quadratic drag are disregarded. Additionally, the use of potential flow theory software packages often lies outside of the expertise of wind turbine designers, posing a practical problem for their introduction in an industrial design process. All the above justifies the popularity of the so-called Morison approach to represent the hydrodynamic response of offshore support structures [12]. This approach can be seen as a good compromise between easiness of implementation and modeling capabilities. Indeed, the Morison equation is a simple formulation that explicitly takes into account the quadratic drag contribution and the motion of the floating platform. This allows to study a broad range of foundation designs and environmental conditions at a reduced computational cost.

Loads prediction using Morison’s theory depends on a set of force coefficients, namely the added mass and drag coefficients. A set of fixed force coefficients with regards to the environmental conditions is often used. Alternatively, look-up tables depending of the considered sea state (such as the ones included in [13]) can be employed. Despite its capabilities, it should be noted that several important assumptions are made when deriving the Morison equation. In particular, the underlying strip theory is only completely valid to predict inline forces on slender members in oscillating flows, where wave diffraction and radiation effects are negligible. This implies a considerable degree of uncertainty in the modeling of three-dimensional effects and large structures. Nonetheless, previous work validated the application of the Morison equation to a semi-submersible wind turbine, and revealed the significance of adding the correct axial contributions (i.e. in terms of pressure force an drag) [14]. This statement should, in principle, be re-assessed when changing the floating platform geometry and configuration.

This is indeed the cornerstone of the present work. A relatively complex spar-type offshore substructure, referred to as the WIND-bos platform, is used. Both experiments and Morison-based hydrodynamic simulations are performed, taking into account the presence of the wind turbine. That allows to calibrate and validate the model, and to analyze the potential limitations of the numerical approach along with the sources of uncertainty of the comparison.
2. Platform Description
The WIND-bos platform is a modified spar comprising a hull bound to a counterweight through a set of three deployable legs. Its installation or dismantling does not require specialized heavy lifting vessels. The turbine is assembled in harbour and it is a self-installing platform. Furthermore, it can be wet towed, avoiding the need of the corresponding equipment transportation as well.

The WIND-bos platform can be separated into three main parts: the counterweight, the legs, and the hull (see Fig. 1). The lowermost part is referred to as the counterweight. This part is deployed when filled with seawater to add most of the weight to the system. The parts connecting the counterweight and the hull are the legs. Unlike other concepts that use flexible tendons to fix the counterweight to the rest of the structure, this design uses a set of rigid legs. Those elements absorb the dynamic forces due to waves and currents, avoiding excessive wear. Additionally, the rigid legs prevent the slacking of cables and tendons, while increasing the fatigue life. The hull is designed to add the required buoyancy and withstand the design wave and current loading. The lower part of the hull is conformed by three equally spaced arms made of a set of vertical flaps and heave plates. The aim of those components is to add damping to the platform. The upper part of the hull is the central stiffened column, that supports the wind turbine tower. It is made of a set of decks, which divide the column into separate compartments. The main structural properties of the WIND-bos platform are summarized in Table 1.

The WIND-bos platform is designed to mount the reference wind turbine DTU 10MW RWT [15]. This turbine is an upwind machine, variable-speed and collective pitch controlled with a rotor diameter of 178.3m.

Further detail of the floating platform properties and dimensions cannot be made publicly available as it represents part of BlueNewables trade secret. However, the authors believe that the information provided suffices the purpose of understanding the technical background and conclusions of the study.

3. Methodology
A numerical model of the WIND-bos platform, mounting the DTU 10MW RWT wind turbine, was built in a commercial aero-servo-hydro-elastic code. The model was subsequently validated
Table 1. Main structural properties of the floating platform, where CoG stands for the center of gravity

| WIND-bos Floating Platform Structural Properties                  |       |
|------------------------------------------------------------------|-------|
| Total External Volume [m$^3$]                                    | 27911 |
| Total Draft [m]                                                  | 81.0  |
| Distance from SWL to Platform Top [m]                            | 18.0  |
| CoG Location in Z-axis [m]                                       | -72.69|
| Platform Roll Inertia about CoG [kg m$^2$]                       | 9.735E+09|
| Platform Pitch Inertia about CoG [kg m$^2$]                      | 9.788E+09|
| Platform Yaw Inertia about CoG [kg m$^2$]                        | 9.067E+09|

based on a series of wave basin experiments. Those included decay tests (in a free-floating and moored setting), and RAOS with a significant wave height of 2m and periods ranging from 2 to 21 seconds.

3.1. Experimental Model
A scaled model of the WIND-bos platform and the reference wind turbine DTU 10MW RWT was extensively tested by Oceanide at the BGO First facility in La Seyne-sur-mer, France. The floating wind turbine model is a 1/40$^{th}$ scale Froude model of the DTU 10MW RWT machine. Its thrust is modelled by the means of a ducted fan, fixed atop the WIND-bos platform (see Fig. 2). The mooring system consists of three horizontal mooring lines and a set of springs to adjust the stiffness. The cable runs horizontally from the model fairlead to a pulley fixed on the side, from where it runs vertically up to the springs. This setting is used to avoid any catenary effect or wave loads on the springs. Platform motion was captured by the triangulation of fixed-position cameras and markers, installed in the tower mock-up.

3.2. Numerical model
The complete system, including floater and wind turbine, were modeled in HAWC2 [16]. This tool is intended to calculate the floating wind turbine response in the time domain, accounting for the coupled interaction of the external effects, such as the aerodynamic and hydrodynamic loads and the action of the control system. The multi-body formulation of HAWC2 enables the structural definition of independent bodies, increasing the modelling accuracy. In this study, all the bodies in the floating platform were considered to be infinitely stiff, consistently to the experiment.

The hydrodynamic modelling was performed by assigning added mass and drag coefficients to the defined bodies’ nodes. These coefficients model the added mass, viscous damping, and wave exciting forces on the structure. Initially, the force coefficients were estimated based on the literature. This was done by assuming cylindrical sections and planar flow. For the particular case of the non-cylindrical members of the hull, equivalent coefficients were estimated (based on the similarity of the cross-sectional area), in such a way that an equivalence between the loads predicted by HAWC2 and the ones expected from the design geometries was achieved. It should be noted that this simplification may introduce a significant uncertainty, especially with regards to the drag coefficient. Analogously to the cross-sectional hydrodynamic modeling, additional axial loads were taken into account through a set of force coefficients. This consideration is important to model the dynamics in the heave direction, for example, for all support structures with large diameter columns as spar buoys or some semi-submersibles. Table 2, presents the set of estimated force coefficients which were the starting point for the validation process. The force
coefficients are either radial or axial coefficients, i.e. associated with hydrodynamic forces acting parallel or perpendicular to the body cross-section plane respectively. Hence, the cross-sectional and axial forces are decomposed in the different DOFs based on the orientation of the bodies at each time step.

Table 2. Added mass and drag coefficients (base case)

|                      | Added Mass ($C_A$) | Drag ($C_D$) |
|----------------------|--------------------|--------------|
|                      | Radial  | Axial  | Radial  | Axial  |
| Counterweight        | 1.1683  | -      | 2.1635  | -      |
| Legs                 | 1.0     | 0.65   | 0.6     | 1.1    |
| Hull arms            | 0.9848  | 0.6024 | 2.1930  | 1.1410 |
| Hull column          | 1.0     | -      | 0.6     | -      |

Mooring lines were modeled through a dynamic approach, based on a general cable formulation. As the effect of mooring lines on the still water level was negligible, first simulations were performed in a free-floating context.

4. Results

4.1. Decay Test

The natural frequencies were computed based on the free decay tests, performed in the numerical model by an impulse load induced motion. The induced displacement for each DOF coincides with the initial displacement observed in the experimental decay tests, allowing not only a comparison of natural frequencies but also damping levels. Figure 3 presents the decay test time series for the pitch and heave DOF in a free-floating setting. The comparison of the natural frequencies computed by HAWC2 and the experimental values revealed relative differences of -1.56%, 3.66% and 3.3% for the DoFs of heave, pitch and roll respectively.

Based on the first comparison between the experimental results and the original Morison’s, the model was updated for a better correlation. The shift in the frequency was adjusted by first increasing the added mass coefficients in the hull arms by roughly 20% to match the heave natural period, and then reducing the legs and hull column coefficients to match the pitch and roll natural periods. Note that this last adjustment was made on vertically oriented members that do not have section ends exposed to water, therefore having a low impact on the dynamics in heave.

The damping in the system is slightly higher in the experimental model for all free-floating DOFs considered. In order to adjust these levels, the drag coefficients of the hull arms and counterweight were increased 10% in the first place, correcting most of the observed deviation. For the roll and pitch DOFs, an increase of 100% in the legs and hull column radial $C_D$ from 0.60 to 1.20 was required. This adjustment has a minor impact on the overall damping level and has a negligible effect on heave motion due to the members vertical orientation. For the heave DOF, an extra linear drag coefficient dependent on the diameter of the hull column was added. The value was adjusted based on observation of the time series and finally equal to $C_{D,\text{linear}} = 4.6$. Table 3, presents the added mass and drag coefficients adjustment on floating platform’s members.

The apparently substantial change in the legs’ and hull column’s added mass coefficients, led to small a small change in the natural frequencies in pitch and roll, as most of the contribution is governed by the counterweight and hull arms. The decay tests were repeated for the numerical model with the coefficients adjustments. Table 4 quantifies the natural periods, evidencing how the difference became negligible after the adjustments.
Figure 3. Numerical and experimental heave and pitch free decay tests comparison without mooring lines. Left column: baseline case. Right column: results with adjustments

Table 3. Added mass and drag coefficients before and after the adjustment

| Radial Force Coefficients | Added Mass (C_A) | Drag (C_D) |
|---------------------------|-----------------|------------|
|                           | Base            | Adj.       | Base       | Adj.       |
| Counterweight             | 1.1683          | 1.1683     | 2.1635     | 2.40       |
| Legs                      | 1.0             | 0.0        | 0.6        | 1.20       |
| Hull arms                 | 0.9848          | 1.1777     | 2.1930     | 2.40       |
| Hull columns              | 1.0             | 0.0        | 0.6        | 1.20       |

Table 4. Natural periods comparison between the HAWC2 model and the experimental decay tests for the heave, pitch, and roll degrees of freedom without mooring lines after calibration

| Natural Periods | HAWC2 Results | Exp. Results | Error |
|-----------------|---------------|--------------|-------|
| Heave           | 44.9 s        | 45.0 s       | -0.22%|
| Pitch           | 27.4 s        | 27.3 s       | +0.37%|
| Roll            | 27.4 s        | 27.3 s       | +0.37%|

Following the comparison in a free-floating setting, the natural periods were compared in a moored condition. The pitch, heave, and roll DOFs presented a minor shift in the natural frequencies. Table 5 quantifies the observed difference and confirms the small deviation in the pitch and heave DOFs. The damping level is still in fine agreement.
Table 5. Natural periods comparison between HAWC2 and the experimental decay tests for the surge, heave, pitch, and roll degrees of freedom with mooring lines

| Natural Periods | HAWC2 | Exp. Results | Error |
|-----------------|-------|--------------|-------|
| Heave           | 44.4 s| 44.1 s       | +0.68%|
| Pitch           | 26.9 s| 25.9 s       | +3.86%|
| Roll            | 26.9 s| 25.9 s       | +3.86%|

4.2. Response Amplitude Operators

The RAOs were computed for the entire range of regular wave cases considered in the experimental tests. These quantify the wave frequency-dependent displacements due to waves with unit amplitude, evidencing how the dynamic response is affected by the wave loading.

![Figure 4. Motion RAOs for the heave (upper figure) and pitch (lower figure) DOFs](image)

Notwithstanding the deviations brought by the uncertainty present in the scaled model and the measuring devices or the mismatch in the mooring lines stiffness, there is a fine agreement for a broad range of sea states with the exception of the most extreme ones for the heave DOF.

There is no clear indication that the absence of diffraction effects led to discrepancies with the experiments that could be directly attributed to this cause. Furthermore, it assumed that the deviations observed are most likely due to not accounting for the frequency-variation of parameters as the added mass, radiation damping, and viscous damping or the wave exciting forces when using a set of fixed force coefficients for all sea states. As suggested in Robertson et al. [17] and Jonkman [18], Morison’s applicability to these types of floating foundations is reasonable when diffraction effects can be neglected, radiation damping is negligible, and flow separation may occur.
The influence of flow separation effects can be evaluated by calculating the diffraction ratio, defined as the diameter or the size of the side facing the flow divided by the wave length \( D/\lambda \). The boundary where these effects become non-negligible is usually recognized as 0.2 [19]. Fig. 7, shows the diffraction ratio for each member of the platform for a set of sea states ranging from mild to severe, proposed in [18] (see Table 6). It can be observed that only the mildest sea states have large diffraction ratios, while for most sea states its value is below 0.2. Hence, wave scattering effects not accounted for in the current model could be the source of discrepancies, although further study would be required to confirm the need for a different treatment when subjected to these sea states.

| Table 6. Sea states ranging from mild (sea state 1) to severe (sea state 8) |
|---------------------------------------------------------------|
| Sea States | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| \( T_p \) [s] | 2.0 | 4.8 | 6.5 | 8.1 | 9.7 | 11.3 | 13.6 | 17.0 |
| \( H_s \) [m] | 0.09 | 0.67 | 1.40 | 2.44 | 3.66 | 5.49 | 9.14 | 15.24 |

Table 7. Diffraction ratio of the counterweight (‘CW’), legs, hull arms (‘H\(_{arms}\)’) and hull column (‘H\(_{column}\)’) at different sea states. The cases where the diffraction ratio is over 0.2 are highlighted in bold.

| Table 7. Diffraction ratio of the counterweight (‘CW’), legs, hull arms (‘H\(_{arms}\)’) and hull column (‘H\(_{column}\)’) at different sea states. The cases where the diffraction ratio is over 0.2 are highlighted in bold. |
|---------------------------------------------------------------|
| Sea States | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| CW | \textbf{1.92} | \textbf{0.33} | 0.18 | 0.12 | 0.08 | 0.06 | 0.04 | 0.03 |
| Legs | \textbf{0.64} | 0.11 | 0.06 | 0.04 | 0.03 | 0.02 | 0.01 | 0.01 |
| \( H_{arms} \) | \textbf{1.92} | \textbf{0.33} | 0.18 | 0.12 | 0.08 | 0.06 | 0.04 | 0.03 |
| \( H_{column} \) | \textbf{1.92} | \textbf{0.33} | 0.18 | 0.12 | 0.08 | 0.06 | 0.04 | 0.03 |

Evaluating the influence of radiation damping is not as straightforward, though it is expected that the larger the deviation from the long wave-length approximation is, the larger the influence will be.

The Keulegan–Carpenter number, KC, is a parameter associated with the dominance of inertial or viscous loads, with large KC numbers indicating the presence of flow separation effects. For cylinders, as the KC number approaches 0, these effects become negligible, being 2 the limit for which they become important [20]. On the contrary, for rectangular cylinders, small vortices form at low KC numbers, yielding large viscous effects. Therefore, for circular cylinders in the structure such as the legs and hull column, viscous effects are expected to be important for only the most extreme sea states where the KC number is over 2 (see Fig. 5), whereas for the rectangular cylinders, these effects would be important among all sea states.

Another parameter that can affect the drag coefficients is the Reynolds number (Re). Drag coefficients of rectangular sections with sharp edges, as present in this platform, are nearly independent of the Re number. For circular cross-sections, its value is nearly constant for supercritical regimes. Fig. 6 shows that the cylindrical members in the structure are subjected to critical and super-critical regimes in most cases.

Overall, this assessment let us determine that diffraction effects will be important for the mildest sea states, where the Morison’s main assumptions are compromised. For moderate and severe sea states, the low KC numbers indicate that the loading will be mostly inertia dominated.
Figure 5. KC number of the floating platform’s members at different sea states (‘SS’). The KC number was computed using the amplitude of the wave particle velocity, which is a function of the depth, wave number, water depth (with a reference depth of 100 m) and the given sea state, i.e. wave height and period. The reference length scale is the diameter for circular cross-sections and the length of the side facing the flow in the case of rectangular cross-sections.

Figure 6. Re number of the cylindrical floating platform’s members at different sea states (SS). The Re number was computed using the amplitude of wave particle velocity, which is a function of the depth, number, water depth (with a reference depth of 100 m) and the given sea state, i.e. wave height and period. The reference length scale is the diameter for circular cross-sections and the length of the side facing the flow in the case of rectangular cross-sections.

Non-diffraction dominated flows are suited for a Morison representation, especially in the case of viscous loads, not accounted for in other models as potential flow models.

If the agreement is not sufficient, the force coefficients can be calibrated for more than one sea state by either comparing with experimental results or using look-up tables found in the literature. Furthermore, evaluating the influence of inertial or viscous loads allows focusing the calibration efforts. Adjusting added mass coefficients is less cumbersome than adjusting drag
coefficients, which are affected by many more parameters. Using the WIND-bos platform as an example, it is expected that tuning the inertial coefficient for more than one sea state would produce a beneficial impact on the agreement with experimental results.

5. Conclusions
In this work, the applicability of Morison’s theory to a floating platform that violates, to some degree, its core assumptions, was studied. It was observed that it is possible to assess the conditions for which the approach is expected to be compromised based on a set of sea states.

Overall, reaching a high level of agreement in terms of natural frequencies and damping levels was proven to be possible, requiring only minor adjustment efforts if the initial modelling is done appropriately. This included studying the members’ geometry and representative force coefficients for the expected sea states.

Regarding the dynamic response under wave loading, although the calculation of forces and moments with Morison’s equation using the relative fluid velocity accounts for the radiation-induced added mass and flow separation induced drag, neither diffraction nor radiation damping forces are included in the model. A fine agreement was achieved for a broad range of sea states, which was expected for maritime conditions yielding low diffraction ratios. For wave regimes yielding large diffraction ratios, the deviation from the experiments could not be directly attributed to wave scattering effects; although it should be the subject of further study.

Hence, this approach allows a fair representation of the dynamic response at a low computational level. Additionally, for those cases where the level of agreement is not sufficient, a further validation step is proposed, consisting on adjusting these coefficients for more than one sea state.

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