Validation of drift motions for a semi-submersible floating wind turbine and associated challenges

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Abstract. In the EU H2020 project LIFES50+, a 1:36 scaled model test campaign was carried out for the NAUTILUS-DTU10 semi-submersible floating offshore wind turbine with active ballast. This paper concentrates on the modelling capabilities of a state-of-the-art time domain simulation model FAST8 for floating offshore wind turbines and specific challenges associated with the validation process. For the modelling, the platform is considered as a rigid body, and the frequency dependent radiation damping, added mass, and wave excitation are evaluated with a panel code using potential theory. The results from the scaled model are compared to the simulations, looking into the effects on the platform when the first order radiation-diffraction hydrodynamics through the Cummins equation, the mean drift coefficients from the nearfield solution for Newman's approximation of the second-order difference-frequency wave forces, and full quadratic transfer function (QTF) are taken into account. Moreover, viscous forces on the floating platform are modelled through Morison elements with coefficients of drag. Frequency domain analysis of the motions showed good agreement after modifications of the coefficients of drag of the Morison elements for tests performed with a pink wave spectrum. On the other hand, the extreme wave test showed large discrepancies between results and simulations, which could not be overcome by the inclusion of the QTFs or tuning of the coefficients of drag which model the viscous forces. This is followed by a discussion of the challenges in the modelling approach, and other validation techniques are proposed for future research. The main goal is to define the Morison elements for the floater, and tune the drag coefficients to enable the numerical tool to capture the floater’s motions. The effect of the change of the coefficients on the simulation outputs are shown.

1. Introduction

In the field of wind energy, research efforts have focused on the development of engineering simulation tools capable of modeling the floating wind turbine, as well as the verification and validation of these tools. Works, such as those from [1] have contributed to the development of tools such as FAST while projects such as OC4 [2] have compared different tools used for simulations of floating wind turbines. The work presented in the OC5 project, [3] has shown however that there are discrepancies between the tools and the physical experiments for the semi-submersible platform tested. The motions and forces created by second order effects were not reproduced appropriately. The second order forces are more important for low natural frequency degrees of freedom like surge. The effect of including the second order on the simulation results will be discussed in the paper.
In this paper, a comparison is made of the scaled model test of a semi-submersible floating wind turbine platform with a numerical model build in FAST v8.16.00a-bjj [4]. The numerical model is first calibrated with the use of the decay tests performed on the scaled model and then comparisons with irregular wave tests are carried out. The experimental data is from a 1:36 NAUTILUS-DTU10 semi-submersible floating wind turbine [5] carried out for the EU H2020 project, LIFES50+ at the SINTEF Ocean facilities [6].

For this research, special focus has been given to the modelling of the hydrodynamics. The numerical model includes first order wave forces, viscous effects in the form of Morison elements with associated coefficients of drag, as well as second order wave forces, either with Newman’s approximation or with full quadratic transfer functions (QTF). After calibration of the coefficients of drag to match the damping of the decay tests, the numerical model was compared to the tests for the cases of irregular waves. The validation exercise shows that not all motions created in the frequencies related to the second order region are well represented by the decay-calibrated numerical model. Further tuning of the drag coefficients shows the load case dependency of the coefficients of drag used. The challenges in the representation of the viscous effects in the engineering simulations models are then discussed. Finally, other calibration techniques are proposed for future research such as depth dependent drag, and studying the load case dependency of the viscous forces with both aerodynamics and hydrodynamics loads taken into consideration.

2. Model test description
The scaled model test campaign referred to here is that of the NAUTILUS-DTU10 presented in [6]. The Froude scaling law was used based on the length scale factor of 1:36. In this article though, all values are given in full scale size unless noted. It has a 10MW turbine mounted on a floating platform. The model is composed of a 4 column (10.5m diameter) semi-submersible floating platform connected by a square pontoon. The platform has a 26m height and 65.25m width and breadth. It has 4 catenary mooring lines, one connected to each of the columns. The rotor-nacelle assembly of the 10MW turbine has been replaced by a Real-time hybrid model referred to as ReaTHM. This is a robotic system which simulates the wind loads through a series of tension lines. The constructed model as well as the reference coordinate system is shown in Figure 1. The mass of the ReaTHM is similar to the mass of the Rotor Nacelle Assembly (RNA).

![Figure 1](image_url)

The NAUTILUS-DTU10 has 4 catenary chain mooring lines. The mooring system of the scaled model was built to match the linear stiffness and the weight characteristics in water of the lines. This means that the scaled model was built of 2 chains ballasted with lead wires, and a spring element inserted at around 100m from the fairlead to adjust for the Young’s modulus. The lines are anchored 130m below the mean water level.
The scaled model was tested inside a wave tank at SINTEF Ocean. The tests in the wave tank consisted of a variety of decay, pull-out, regular wave, pink wave spectrum, extreme irregular wave, (with and without simulated wind) tests. The scaled model also possesses extensive measurement equipment; this includes measurement of the displacements of the platform.

2.1. Experiments
A variety of experiments were performed at the wave basin and the platform displacements in surge, sway and heave, and the rotations in the roll, pitch and yaw were measured. This included: decay tests without mooring for the pitch and heave DOFs, decay tests with mooring for all DOFs, pullout tests to identify the static tension on the mooring lines, pink noise wave spectrum and extreme irregular wave tests. The tests used for this research were performed without simulated wind loads. For the waves’ tests, a -15° incoming wave direction has been used (see coordinates in Figure 1). This direction was used because the floater’s prototype was installed in the water tank test with a 15° offset from the water tank reference frame. This was done during testing to be able to produce waves both in line with the mooring lines, and exactly between them.

Table 1: Subset of experimental conditions selected for this study. No wind was applied for these tests.

| Test Number | Case                                      | H_s [m] | T_p [s] | Test duration [s] |
|-------------|-------------------------------------------|---------|---------|-------------------|
| 2021        | Pullout surge direction                   | -       | -       | -                 |
| 3011        | Free decay heave (initial offset = 1.49m) | -       | -       | -                 |
| 3020        | Free decay pitch (initial offset = 2.12deg) | -       | -       | -                 |
| 3130        | Moored decay heave (initial offset = 0.7m) | -       | -       | -                 |
| 3110        | Moored decay surge (initial offset = 6.8m) | -       | -       | -                 |
| 3151        | Moored decay pitch (initial offset = 5.25deg) | -       | -       | -                 |
| 3160        | Moored decay yaw (initial offset = 11.8 deg) | -       | -       | -                 |
| 4210        | Pink noise                                | 2.0     | 4.5-18.2 | 11880            |
| 6241        | Extreme wave: Pierson-Moskowitz spectrum | 10.9    | 15      | 11880            |

3. Numerical model
In the LIFES50+ Deliverable 4.5 [7], a FAST8 model has been outlined to describe the DTU 10MW Reference Wind Turbine mounted on the NAUTILUS-10 floating substructure. The model used for this work is based on this FAST8 model but a series of changes had to be made to match the geometry of the model tested in the wave basin. Masses and dimensions have been thus adjusted accordingly. The changes of the wave basin scaled model with ReaTHM robotic actuator to that of the designed prototype include:

- Overhang of the rotor-nacelle assembly (RNA) is neglected, since the RealTHM actuator is centred on the tower.
- Numerical part of the ReaTHM hardware in the loop model includes two DOFs in addition to the platform DOFs; the generator and the first flapwise mode of the blade.
- Platform mass and inertias are slightly different for the scaled model test than those of the original design NAUTILUS-DTU10.
- Active ballast system of NAUTILUS-DTU10 is not entirely modelled in the experiment, static distributions of the ballast mass are tested.
Table 2: Summary of masses of the FAST8 model

| Component                                      | Modified model | FAST8 | SINTEF wave basin test (ballast filled for 0 m/s wind speed) |
|------------------------------------------------|----------------|-------|-------------------------------------------------------------|
| Blade mass (kg)                                | 41,692         | -     | -                                                           |
| Nacelle mass (kg)                              | 446,036        | -     | -                                                           |
| Hub mass (kg)                                  | 10,5520        | -     | -                                                           |
| Tower (kg)                                     | 879,380        | -     | -                                                           |
| Tower-top (3 blades + hub + nacelle) (kg)      | 676,634        | -     | -                                                           |
| RNA + tower (kg)                               | 1,556,015      | 1,557,000 |                                                            |
| Platform mass (kg)                             | 7,328,000      | 7,328,000 |                                                            |
| Ixx platform (kg.m²)                           | 4,635,000,000  | 4,635,000,000 |                                                   |
| Iyy platform (kg.m²)                           | 4,634,000,000  | 4,634,000,000 |                                                   |
| Izz platform (kg.m²)                           | 7,981,000,000  | 7,981,000,000 |                                                   |

3.1. Mooring Model

In the FAST8 model, the 3 segments per mooring line were not modelled and only one long segment was used. Thus to better match the model tested in the wave tank, the value of the specific weight of the mooring line which was then used was averaged to be 180.6 (kg/m). This results from the equation relating the specific weight of each segment and its length to an average value:

\[ S_{w_0} = \frac{S_{w_1}L_1 + S_{w_2}L_2 + S_{w_3}L_3}{L_0} \] (1)

where \( S_{w_0} \) is the specific weight (kg/m) of the mooring line, with 3 lines sections per mooring line used in the tested model with different lengths \( L \).

3.1.1. Mooring line damping. The software used for modelling the mooring lines is called MAP++. The software creates a static model. The mean forces in the mooring lines are due to elasticity, weight of mooring line in the fluid, and geometric nonlinearities are taken into account. MAP++ ignores the inertial forces and fluid drag loads. This means that any damping created by the mooring lines in the wave tank tests can only be taken into account in the simulations by changing the numerical model’s platform damping. In this research, I tried at the beginning to use Moordyn for the simulations, but it adds the viscous drag effect to the mooring lines. This adds more uncertainty to drag coefficients included in the model. Therefore, I am using MAP++ to decrease the sources of uncertainty in my simulations.

3.2. Hydrodynamic model

The hydrodynamics of the FAST8 model are based on first order radiation-diffraction hydrodynamics through the Cummins equation. The potential damping, potential added mass and the force-RAO are calculated in WAMIT [9]. These are applied at the centre of flotation in the FAST model. This however neglects the relative velocity between the floating substructure and the wave field. In FAST, there are two methods to account for the viscous effects. First, FAST includes a Morison-formulation for the viscous drag on the body, a strip-theory-based approach. Additionally, FAST uses linear and quadratic lumped damping matrices to capture viscous effects, which we are not going to use in this work.

The panel code calculation was run with a draft of 17.36 m, corresponding to the ballast mass for zero wind speed. The mean drift coefficients from the nearfield solution are used for Newman’s approximation of the second-order difference-frequency wave forces. The wave direction is of -15 degrees, according to the main wave heading angle of the tests [10].
3.2.1. Damping properties. The platform drag discretization was created with four different coefficients of drag to represent the horizontal and vertical components of drag on the columns and pontoon of the platform. To model the square pontoons through circular drag elements, the projected area from the bird’s eye view of the pontoon has been discretized into circles of 10.5m in diameter. The circular area of the column and the circular areas of the pontoon have been assigned two different vertical coefficients of drag. The areas can be seen in Figure 2.

The thickness of the pontoon is taken as the diameter for a cylinder (2m) in the simulation model, which connects the columns and has a horizontal coefficient of drag. This would be the green line in Figure 2. The vertically oriented columns, which are also modeled as cylinders (10.5m in diameter), have a different horizontal coefficient of drag. At the beginning, the values of the drag coefficients were first based on work done in [1] but were then changed to better match the decay tests of the platform.

Before starting to estimate the horizontal or vertical drag coefficients, we can predict that the values of the vertical drag coefficient will be higher values, due to the difference between the real geometry of the floater and the model implemented in Hydrodyn. This is because Hydrodyn is only capable of modelling cylindrical elements, and can not model rectangular shapes like the Nautilus pontoons. We expect the horizontal drag coefficients to be higher because if we looked at the drag force ($F_D$).

$$F_D = \frac{1}{2} \rho v^2 C_D A$$

The drag force depends the square of the velocity of the motion ($v^2$), the drag coefficient ($C_D$), and the surface area ($A$). In the horizontal direction, the area of the floater is totally modelled by the green (thickness of pontoons) and yellow cylinders (vertical cross section of the yellow cylinders) in Figure 2. However, for the vertical direction as it is clear in Figure 2, there are areas which are not covered in the model by neither the red nor the yellow cylinders (white areas in Figure 2). This means that in order to get the total value of the drag forces the drag coefficients value must increase.

4. Tuning of numerical model to experimental data
The numerical model was further adjusted to match certain data from the experiments. This includes the verification of the tension on the mooring lines by comparing simulations with the pull-out test as well as comparison of the frequency and damping of the decay tests. This was an iterative process since the properties of the mooring lines had to be changed to best match the decay frequencies and then verifying that the mooring tensions were also well modeled. As a result, the mooring lines’ mass in the simulation model is changed to 157.172 (kg/m) which is a 12.9% change.

4.1. Decay tests
Decay tests are used to find out the natural periods and damping of the system. The change of the numerical model is the carried out to best match the decay frequencies and damping of the experiment. While this approach of using the decay tests is well suited for the drag-induced damping, it does not model the drag-induced wave forces.

First, the decay with no mooring was tested and simulated, and the comparison of the tests in heave and in pitch was carried out as in Figure 3. Having no mooring helps isolate the influence of the masses and inertias of the platform on the decay frequencies, as well as the platform damping values used in the simulation. It was found that the frequencies of decay match well. The vertical damping coefficients were then used to try to best match the damping of the experiments.
Further, the decay tests with the mooring lines are compared to simulations. During the process of matching the frequencies, it was found that the surge frequency in the decay test did not match well the experiment. Since the surge response is very important in the analysis of the system, modifications to the mooring numerical model were carried out to obtain a better match. In this case the specific weight of the mooring lines was lowered (alternatively one could also change the anchor point distances of the mooring model). Even though this improved the match of the frequency in the surge response, it came with the side effect of also changing the yaw frequency in decay.

For the numerical model, the vertical coefficients of drag have an influence on the heave and pitch oscillations amplitudes, while the horizontal coefficients of drag of the platform have an effect on the surge and yaw oscillation amplitudes as shown in Figure 4.

Summary of the tuned drag coefficients used in decay simulations is shown in Table 3.

| Cd_{ver col} | Cd_{ver pon} | Cd_{hor col} | Cd_{hor pon} | Specific weight mooring line (kg/m) |
|-------------|-------------|-------------|-------------|----------------------------------|
| FAST8 decay tuned | 78.05 | 12.95 | 0.3575 | 1.025 | 157.172 |

The experimental behaviour of the damping, as shown in Figure 5 is hard to fit into a simple model, and there are also dependencies of the damping on the Reynolds number and the KC number. In the left side of Figure 5 one can see a best fit estimate of an exponential function to the decay data from the wave basin tests for the moored heave decay. On the right side, one sees the damping ratio as a function of the mean amplitude of the decay motion for each period of oscillation. Here, a constant damping ratio (blue line) would represent a system with only linear damping. However, the scattering of the data shows the non-linear behaviour of the damping of the system. The scattering of the damping ratio D also shows the difficulty to fit experimental data to a numerical model for the damping.
Figure 4: Decay tuned FAST8 model simulations compared to the decay tests

Figure 5: Plot of damping ratio (D) based on maximums of the experimental data for the heave decay test

A comparison between the estimated frequencies from the simulations, and the test results is summarised in Table 4.

Table 4: Comparison of simulated and measured excitation frequencies

|                  | Surge Moored | Heave Moored | Pitch Moored | Yaw Moored |
|------------------|--------------|--------------|--------------|------------|
| Test (Hz)        | 0.0079       | 0.0527       | 0.0314       | 0.0110     |
| FAST8 decay tuned (Hz) | 0.0082     | 0.0533       | 0.0322       | 0.0100     |

4.2. Pull out test

The static tension of the mooring lines is check with the pull-out test of the platform in the surge direction. The static solution at a certain surge displacement for the FAST8 model is compared to the estimated tension from the experiments in Figure 6. The experimental data had an initial offset in the surge direction which is removed from the signal measurement for comparison purposes.

Figure 6: Pull out test 2021 mooring line tension
The underestimation of the tension in FAST8 could be caused by the adjustment of the specific weight of the mooring line model. However, these adjustment were needed to match the decay frequency response in surge for simulation and experiment as mentioned earlier.

5. Second order wave excitation forces
Second order nonlinear drift forces can be represented through Newman’s approximation [11]. The approximation assumes low rigid-body natural frequencies of the FOWT, such that the difference frequency \( \delta_{ij} = \omega_i - \omega_j \) is small and \( \omega_i \) and \( \omega_j \) are close to the diagonal of the QTF. The assumption of small natural frequencies holds for the surge degree of freedom (DOF) but potentially not anymore for the pitch-DOF (at 0.032Hz for the tested system). Even though the design of the FOWT will demand its natural frequencies outside the range of the wave frequencies, the natural frequencies often coincide with the frequencies of the drift forces, leading to high excitations, especially of the mooring system [12], [13].

Newman’s approximation for the second order forces provides the advantage of not being as computationally expensive as determining the full quadratic transfer functions for the slow and mean drift forces. The implementation of the second order forces and the difference full quadratic transfer functions in FAST8 is outlined in [14]. The following sections will thus show a comparison of scaled model test results with a numerical model (with Morison damping elements tuned to fit the decay tests) for validation purposes.

5.1. Pink wave test
The platform motion response in the frequency domain is compared for a pink wave Test 4210 with a significant wave height of 2m and periods \( (T_p) \) ranging from 4.5-18.2s. A comparison is shown for the numerical model (with Morison element damping drag coefficients tuned to decay tests) and the scaled model tests.

Figure 7 shows the numerical models can reproduce the motions of the platform in the region of the wave frequencies, which are in this case above 0.06Hz. For the region below the wave excitation frequencies, the model without any second order wave forces cannot properly excite the low natural frequencies of the platform. When including the second order forces in the form of the Newman approximation or the full QTFs, the response of the surge and sway showed good agreement. For heave, roll, pitch and yaw, the peak response for frequencies below the wave frequencies (corresponding to the natural frequencies of the platform) was under-predicted. This indicates that the platform is over damped and the decay tuned decay coefficients are an over estimation for this load case.

For the second order forcing, the response with full QTF and Newman approximation is the same for surge and sway DOFs. This is expected since the natural frequencies of sway and surge are very low (0.007 Hz). The difference between QTF and Newman approximation can be seen in the roll, and pitch DOFs response. This is due to the fact the higher pitch natural frequency, compared to surge, which can only be captured using the full QTFs.
Figure 7: Comparisons of response of the platform in the frequency domain of the simulation with MAP++ mooring model for Test4210 with pink wave spectrum.

5.2. Extreme irregular wave test
Similarly the comparison is carried out for the irregular extreme wave Test6241 with a significant wave height of 2m and peak periods 18.2s. The results shown in Figure 8 of the decay tuned numerical model show that in this case, as opposed to the pink noise test, the response in the surge and sway direction are underestimated, both by the simulation with Newman’s approximation and with full QTFs. Furthermore, the peak response for the roll, pitch and yaw for the frequency region below the wave excitation frequencies is under-predicted similar to the pink wave test. The region of the wave frequencies (above 0.05Hz) for the roll and pitch show a large excitation, corresponding to the extreme wave spectrum used. However, this excitation is over-predicting the response of the platform at these frequencies. Only the heave response shows good agreement. For heave DOF, models with no second order wave forces, Newman’s approximation, and full QTFs show good results.

The underestimated response in surge and sway comes from the viscous effects which have a high effect on the model response in the low frequency region. The system is currently over damped at low frequency, which shows that the decay tuning for the drag coefficients over damps the system. A load case specific tuning is needed.

For the second order forcing, the same comment can be seen as in pink wave case. The response with full QTF and Newman approximation is the same for surge and sway DOFs. This is expected since the natural frequencies of sway and surge are low. The difference between QTF and Newman approximation can be seen in the pitch DOF response. This is due to the fact the higher pitch natural frequency, compared to surge, which can only be captured using the full QTFs. After observing the difference between using full QTF and Newman approximation on the accuracy of our simulation compared to the experimental data, we decided to use the full QTFs for the rest of the simulations and during the tuning process.
Figure 8: Comparisons of response of the platform in the frequency domain of the simulation with MAP++ mooring model for extreme wave test.

6. Load case specific drag coefficients

The effect of changing the coefficients of drag of the Morison elements in the vertical and horizontal directions, as well as the case-dependency of the coefficients used will be shown in the following sections. The coefficients of drag used for the following section are shown in Table 6. Through this section the pink wave test, and the extreme wave test will be used to tune the Morison drag coefficients, and the comparison between the new simulations and the wave tank tests result will be presented.

Table 5: Coefficients of drag for the Morison elements used in FAST8 model

| Model                    | Test                      | $C_d_{ver\,col}$ | $C_d_{ver\,pon}$ | $C_d_{hor\,col}$ | $C_d_{hor\,pon}$ |
|--------------------------|---------------------------|------------------|------------------|------------------|------------------|
| Decay tuned              | Combination of different decay tests | 78.05            | 12.95            | 0.715            | 2.05             |
| Pink noise tuned Cds     | 4210                      | 23.415           | 3.885            | 0.715            | 2.05             |
| Extreme irregular wave tuned Cds | 6421              | 31.22            | 5.18             | 0.5125           | 0.1787           |

6.1. Pink wave test

The first case examined is that of the pink noise spectrum. As results shown in Figure 7, the decay tunes model has not been able to reproduce the heave, roll, pitch and yaw motions in the low frequency range. The drag coefficients in the vertical direction are now modified. The vertical drag coefficient was picked as it influences the heave and the pitch degrees of freedom as discussed in section 4. The results of the changes are shown in Figure 9.

The results of the heave, pitch and roll in Figure 9 show the excitation of the respective natural frequencies. The highest values of damping are those used in the decay tuned model. As the damping in the vertical direction gets smaller, the increase in the excitation due to the second order forces around the natural frequency is evident. It can be seen that by changing the coefficients of drag in the vertical direction the simulated response can better reflect that of the experiments than by the use of the decay tuned values. The pitch and heave DOFs simulation now shows the peaks at lower
frequency with a good agreement to the experimental results. The roll DOF response shows an overestimation while the yaw DOF under predicts the motion responses at lower frequencies.

![Graphs comparing simulation and experimental data](image)

**Figure 9:** Comparisons of response in the frequency domain of the simulation with MAP++ mooring model for pink noise Test4210 for changing coefficients of drag in the vertical direction.

### 6.2. Extreme irregular wave test

Similarly, for the case of the irregular wave Test6421, the effect of the coefficients of drag on the response is now analysed. In this case, both the vertical and the horizontal coefficients altered to try to best match the response. Both coefficients are changed as none of the DOFs motion responses (except for heave) earlier in Figure 8 showed agreement to the experiments, which means both drag coefficients need to be decreased to capture the low frequency drift forces effects.

Changing of the vertical coefficients of drag changes the response in roll, pitch and heave, while changing the horizontal coefficients changes the response in surge, sway and yaw as shown in Figure 10. For the surge, sway and yaw, it can be seen that the low frequency response in the extreme wave is dominant over that in the wave frequency range. Here the tuned horizontal coefficients cannot match all three of the low frequency responses simultaneously and the yaw is under-predicted and the sway over-predicted.

For the heave, the decrease in the vertical coefficients do not seem to have much effect on the overall response, yet for the pitch and roll, an evident change is seen in the region for the wave excitation and for the low frequency region. Here, the damping and forcing effect of the drag Morison model is seen.

An increase in the response for the low frequency region and a decrease in the forcing response at the wave frequency region are seen due to a decrease in the damping coefficients. The tuned simulations cannot, however, properly reproduce the experiments, overestimating the low frequency pitch response, as well as some of the yaw response at the frequencies slightly above the natural frequency.
Challenges in the validation of drift motions for a semi-submersible platform

For this particular exercise, the validation of the experiments was not achieved. Other modelling approaches of the viscous drag could be used to better represent the motions of similar systems. Berthelsen et al. in [15] and [16] have used depth dependant regions for Cd values. Additionally, to better match the simulation to the results and take into account the duel effect of damping and forcing by the Morison Model, they have increased the drag coefficient in the splash zone only and use a lower value for the rest of the platform. Global damping matrices have been also used in FAST8 instead of Cds as shown in [17] and [2]. These procedures still have the disadvantage of needing tuning with experimental data as well as having a load case dependency of the tuning process. Even if the load case dependency is necessary, the lack of a mathematical model for the tuning of the viscous drag, and thus the dependency on the experimental data, leaves open questions as far as how to standardize a process of validation.

The complexity of the second order forces, the viscous forces and the system response to them is thus of prime importance for the validation process of the engineering modeling tools such as FAST8. A load case dependant workflow for calculation of first order and second order wave forces is already standard, by applying software packages with potential flow theory such as ANSYS-AQWA or WAMIT. This can take into account incoming wave direction, wave height and frequencies of excitation. Yet, the load case dependency of the viscous forces, encompasses the dependency on the KC number and the Reynolds number, and no best practice has been outlined so far for the numerical modeling.

Forced oscillation numerical simulations or basin tests could help better identify the damping needed for the engineering models. Work within deliverable D4.8 of the LIFES50+ project [17] showed that for computational fluid dynamic (CFD) simulations (in ANSYS CFX using Reynolds-averaged Navier-Stokes equations) of forced oscillation in the heave direction, the radiation damping estimated was dependent on the amplitude of the oscillation (and thus the KC number) and at the same
time the linear potential flow theory values were much lower than those calculated using CFD. Also the work of [18] investigated the hydrodynamic coefficients of the heave plate during experiments by using forced oscillation. This included changes to variables such as the KC number, frequency, plate depth, thickness-to-width ratio, among others. Developing a methodology for including such research and results into the validation of the engineering models is work that could contribute to better simulation results.

To summarize, the shown challenges in the reproduction of the drift loading on the floating wind turbine motivates further investigation into the quantification and reduction of uncertainties in the model predictions.

8. Conclusion and outlook
From the analysis presented, the following findings were obtained regarding the use of Morison based elements for damping and second order wave forces for modelling the motions of the NAUTILUS-DTU10 FOWT when compared to wave tank tests:

- The use of the full difference-frequency QTF increased the response of the platform for the low-frequency region (below the wave excitation region), mostly for pitch and roll, when compared to Newman’s approximation for the second order excitation forces. However, neither of the models that used Morison elements with decay-tuned coefficients of drag were able to reproduce appropriately the platform motions during the pink noise spectrum or extreme irregular wave spectrum test.
- Load case specific vertical coefficients of drag were necessary for the model in order to approximate the test response well for all degrees of freedom except the yaw under the pink noise spectrum. Large excitation in the low frequency range of the test results of the yaw degree of freedom could be due to the -15 wave direction. The reason why the model cannot capture this excitation is not clear.
- The sensitivity analysis of the vertical and horizontal drag coefficients showed the increase in response in the low frequency region for decreasing values, as would be expected. However, the changes to the drag coefficients were not sufficient for complete replication of all the platform’s DoF for the irregular wave test cases and the pink noise test. For the extreme irregular wave case, it was also seen that as the vertical drag coefficients decreased, the response in the pitch, and roll decreased at wave frequencies. This would be due to the fact that the Morison equation creates both damping and forcing.

Furthermore, future work should look at:

- The use of load case dependant coefficients of drag were necessary for the tests, yet changing the coefficients for different sea states as well as dependency of the coefficients of drag on the Reynolds number, KC number, possible marine growth, and incoming wave direction necessitate more comprehensive studies. Scaling effects of the platform response and loads will also be of interest for the future development of the platform concept.
- The analysis of test with the turbine in operation is also necessary. The aeroelasticity of the turbine will contribute to the forces and moments as well as provide damping to the system.
- The addition of a dynamic mooring model is also of interest.

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