Computational and Experimental Modelling of Mooring Lines Dynamics for Offshore Floating Wind Turbines

Ph.D. Thesis

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I hereby declare that except where specific reference is made to the work of others, the contents of this dissertation are original and have not been submitted in whole or in part for consideration for any other degree or qualification in this, or any other University. This dissertation is the result of my own work and includes nothing which is the outcome of work done in collaboration, except where specifically indicated in the text. This dissertation contains fewer than 65,000 words including appendices, bibliography, footnotes, tables and equations and has fewer than 150 figures.

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Abstract

The load calculation of floating offshore wind turbine requires time-domain simulation tools taking into account all the phenomena that affect the system such as aerodynamics, structural dynamics, hydrodynamics, control actions and the mooring lines dynamics. These effects present couplings and are mutually influenced. The results provided by integrated simulation tools are used to compute the fatigue and ultimate loads needed for the structural design of the different components of the wind turbine. For this reason, their accuracy has an important influence on the optimization of the components and the final cost of the floating wind turbine.

In particular, the mooring system greatly affects the global dynamics of the floater. Many integrated codes for the simulation of floating wind turbines use simplified approaches that do not consider the mooring line dynamics. An accurate simulation of the mooring system within the integrated codes can be fundamental to obtain reliable results of the system dynamics and the loads. The impact of taking into account the mooring line dynamics in the integrated simulation still has not been thoroughly quantified.

The main objective of this research consists on the development of an accurate dynamic model for the simulation of mooring lines, validate it against wave tank tests and then integrate it in a simulation code for floating wind turbines. This experimentally validated tool is finally used to quantify the impact that dynamic mooring models have on the computation of fatigue and ultimate loads of floating wind turbines in comparison with quasi-static tools. This information will be very useful for future designers to decide which mooring model is adequate depending on the platform type and the expected results.

The dynamic mooring lines code developed in this research is based in the Finite Element Method and is oriented to the achievement of a computationally efficient code, selecting a Lumped Mass approach. The experimental tests performed for the validation of the code were carried out at the École Centrale de Nantes (ECN) wave tank in France, consisting of a chain submerged into a water basin, anchored at the bottom of the basin, where the suspension point of the chain was excited with harmonic motions of different periods. The code showed its ability to predict the tension and the motions at several positions along the length of the line with high accuracy. The results demonstrated the importance of capturing the evolution of the mooring dynamics for the prediction of the line tension, especially for the high frequency motions.

Finally, the code was used for an extensive assessment of the effect of mooring dynamics on
the computation of fatigue and ultimate loads for different floating wind turbines. The loads were computed for three platforms topologies (semisubmersible, spar-buoy and tension leg platform) and compared with the loads provided using a quasi-static mooring model. More than 20,000 load cases were launched and postprocessed following the IEC 61400-3 guideline and fulfilling the conditions that a certification entity would require to an offshore wind turbine designer. The results showed that the impact of mooring dynamics in both fatigue and ultimate loads increases as elements located closer to the platform are evaluated; the blade and the shaft loads are only slightly modified by the mooring dynamics in all the platform designs, the tower base loads can be significantly affected depending on the platform concept and the mooring lines tension strongly depends on the lines dynamics both in fatigue and extreme loads in all the platform concepts evaluated.
Resumen

El cálculo de cargas de aerogeneradores flotantes requiere herramientas de simulación en el dominio del tiempo que consideren todos los fenómenos que afectan al sistema, como la aerodinámica, la dinámica estructural, la hidrodinámica, las estrategias de control y la dinámica de las líneas de fondeo. Todos estos efectos están acoplados entre sí y se influyen mutuamente. Las herramientas integradas se utilizan para calcular las cargas extremas y de fatiga que son empleadas para dimensionar estructuralmente los diferentes componentes del aerogenerador. Por esta razón, un cálculo preciso de las cargas influye de manera importante en la optimización de los componentes y en el coste final del aerogenerador flotante.

En particular, el sistema de fondeo tiene gran impacto en la dinámica global del sistema. Muchos códigos integrados para la simulación de aerogeneradores flotantes utilizan modelos simplificados que no consideran los efectos dinámicos de las líneas de fondeo. Una simulación precisa de las líneas de fondeo dentro de los modelos integrados puede resultar fundamental para obtener resultados fiables de la dinámica del sistema y de los niveles de cargas en los diferentes componentes. Sin embargo, el impacto que incluir la dinámica de los fondeos tiene en la simulación integrada y en las cargas todavía no ha sido cuantificada rigurosamente.

El objetivo principal de esta investigación es el desarrollo de un modelo dinámico para la simulación de líneas de fondeo con precisión, validarlo con medidas en un tanque de ensayos e integrarlo en un código de simulación para aerogeneradores flotantes. Finalmente, esta herramienta, experimentalmente validada, es utilizada para cuantificar el impacto que un modelos dinámicos de líneas de fondeo tienen en la computación de las cargas de fatiga y extremas de aerogeneradores flotantes en comparación con un modelo cuasi-estático. Esta es una información muy útil para los futuros diseñadores a la hora de decidir qué modelo de líneas de fondeo es el adecuado, dependiendo del tipo de plataforma y de los resultados esperados.

El código dinámico de líneas de fondeo desarrollado en esta investigación se basa en el método de los Elementos Finitos, utilizando en concreto un modelo "Lumped Mass" para aumentar su eficiencia de computación. Los experimentos realizados para la validación del código se realizaron en el tanque del École Centrale de Nantes (ECN), en Francia, y consistieron en sumergir una cadena con uno de sus extremos anclados en el fondo del tanque y excitar el extremo suspendido con movimientos armónicos de diferentes periodos. El código demostró su capacidad para predecir la tensión y los movimientos en diferentes posiciones a lo largo de la longitud de la línea con gran precisión. Los resultados indicaron la importancia de
capturar la dinámica de las líneas de fondeo para la predicción de la tensión especialmente en movimientos de alta frecuencia.

Finalmente, el código se utilizó en una exhaustiva evaluación del efecto que la dinámica de las líneas de fondeo tiene sobre las cargas extremas y de fatiga de diferentes conceptos de aerogeneradores flotantes. Las cargas se calcularon para tres tipologías de aerogenerador flotante (semisumergible, "spar-buoy" y "tension leg platform") y se compararon con las cargas obtenidas utilizando un modelo cuasi-estático de líneas de fondeo. Se lanzaron y postprocesaron más de 20.000 casos de carga definidos por la norma IEC 61400-3 siguiendo todos los requerimientos que una entidad certificadora requeriría a un diseñador industrial de aerogeneradores flotantes. Los resultados mostraron que el impacto de la dinámica de las líneas de fondeo, tanto en las cargas de fatiga como en las extremas, se incrementa conforme se consideran elementos situados más cerca de la plataforma: las cargas en la pala y en el eje sólo son ligeramente modificadas por la dinámica de las líneas, las cargas en la base de la torre pueden cambiar significativamente dependiendo del tipo de plataforma y, finalmente, la tensión en las líneas de fondeo depende fuertemente de la dinámica de las líneas, tanto en fatiga como en extremas, en todos los conceptos de plataforma que se han evaluado.
Thesis publications and funding projects

Refereed papers


Conference papers


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Nomenclature

\( \bar{X} \) Mean value of the data
\( \dot{Z}_i \) Vertical velocity of the node \( i \)
\( A \) Cable section area
\( A_i \) Adjacency matrix for the element \( i \)
\( B \) Derivative of matrix \( N \)
\( B_{gi} \) Derivative of matrix \( N_{gi} \)
\( C \) Structural damping matrix of the complete FEM system
\( C_2 \) Constant for the calculation of tangential drag force
\( C_3 \) Constant for the calculation of normal drag force
\( C_4 \) Constant for the calculation of added mass force
\( C_{dn} \) Normal drag coefficient
\( C_{dt} \) Tangential drag coefficient
\( C_i \) Element structural damping matrix in the global reference system
\( c_i \) Element structural damping matrix in the local reference system
\( C_{mn} \) Normal added mass coefficient
\( D \) Diameter of the cable for hydrodynamic calculations
\( d \) Horizontal distance between the anchor and the mean fairlead position during the wave tank tests
\( D_{sc} \) Damping of the seabed per unit length used in the cable-seabed contact model
\( d_w \) Water depth
\( d_w \) Wire diameter of the chain link in the wave tank tests
\( dl \) Infinitesimal length of cable
\( E \) Cable material Young’s modulus
\( EA \) Axial stiffness
\( f \) Frequency of the equivalent fatigue load
\( F_D \) Module of the structural damping force
\( F_{equivalent} \) Fatigue equivalent load
\( F_x \) Component \( x \) of the force at the component of the floating wind turbine considered
\( F_y \) Component \( y \) of the force at the component of the floating wind turbine considered
\( F_z \) Component \( z \) of the force at the component of the floating wind turbine considered
\( g \) Gravity constant
\( H_s \) Significant wave height
\( i \) Element number
\( j \) Index
\( K \) Stiffness matrix of the complete FEM system
\( K_i \) Element stiffness matrix in the global reference system
\( k_i \) Element stiffness matrix in the local reference system
\( K_{sc} \) Stiffness of the seabed per unit length used in the cable-seabed contact model
\( L \) Cable length
\( l \) Distance along the stretched length of the cable
\( l_{0i} \) Length along the unstretched cable to the initial node of the element \( i \)
\( l_0 \) Distance along the unstretched length of the cable
\( L_i \) Length of the element \( i \)
\( M \) Mass matrix of the complete FEM system
\( m \) Inverse S-N material slope in the computation of \( F_{equivalent} \)
$M_i$ Element mass matrix in the global reference system

$m_i$ Element mass matrix in the local reference system

$M_x$ Component $x$ of the moment at the component of the floating wind turbine considered

$M_y$ Component $y$ of the moment at the component of the floating wind turbine considered

$M_z$ Component $z$ of the moment at the component of the floating wind turbine considered

$N$ Shape functions matrix

$n$ Number of elements in the data for the computation of $\sigma$

$N'$ Shape functions matrix with staircase interpolation functions

$N'_{gi}$ Shape functions matrix with staircase interpolation functions in the global coordinate system

$N_1$ Linear shape function number 1

$N_2$ Linear shape function number 2

$N_{gi}$ Shape functions matrix in the global coordinate system

$n_i$ Number of cycles in the load range $S_i$ for the computation of $F_{equivalent}$

$P$ Point at the cable

$R$ Initial reference configuration of the mooring line

$S_i$ $i^{th}$ load range for the computation of $F_{equivalent}$

$T$ Line tension

$t$ Time

$T_h$ Duration of the original time history for the computation of $F_{equivalent}$

$T_l$ Element local to global transformation matrix

$T_i$ Local to global transformation matrix

$T_p$ Peak spectral period of the sea

$V_w$ Wind speed
\( \text{Work produced by the structural damping forces} \)
\( W_{\text{damp}} \)

\( \text{Work produced by the elastic forces} \)
\( W_{\text{elastic}} \)

\( \text{Work produced by the external forces} \)
\( W_{\text{external}} \)

\( \text{Work of the inertial forces} \)
\( W_{\text{inertia}} \)

\( \text{Virtual work} \)
\( W_V \)

\( \text{Coordinate x in the global reference system of the } i \text{ element initial node} \)
\( x_1 \)

\( \text{Coordinate x in the global reference system of the } i \text{ element final node} \)
\( x_2 \)

\( \text{ }^{i}\text{th element of the data} \)
\( X_i \)

\( \text{Coordinate y in the global reference system of the } i \text{ element initial node} \)
\( y_1 \)

\( \text{Coordinate y in the global reference system of the } i \text{ element final node} \)
\( y_2 \)

\( \text{Coordinate z in the global reference system of the } i \text{ element initial node} \)
\( z_1 \)

\( \text{Coordinate z in the global reference system of the } i \text{ element final node} \)
\( z_2 \)

\( \text{Vertical position of the node } i \)
\( Z_i \)

\textbf{Greek Symbols}

\( \beta \) Rayleigh damping coefficient proportional to stiffness

\( \gamma \) Line mass per unit of cable unstretched length

\( \gamma_r \) Equivalent mass per unit length of the cable submerged in water

\( \sigma \) Standard deviation

\( \delta S_0 \) Indentation of the line into the seabed due to self-weight

\( \dot{\varepsilon} \) Axial deformation velocity

\( \varepsilon \) Axial deformation

\( \xi_i \) Element local coordinate

\( \rho_c \) Density of the cable material

\( \rho_w \) Density of the water

\( \psi_1 \) Staircase function number 1
\psi_2 \quad \text{Staircase function number 2}

**Vectors**

\( \delta \vec{P} \) \quad \text{Vector with the virtual displacements of the system nodes in the global reference frame}

\( \delta \vec{P}_i \) \quad \text{Element } i \text{ nodal virtual displacement vector in the global reference system}

\( \delta \vec{p}_i \) \quad \text{Element } i \text{ nodal virtual displacement vector in the element reference system}

\( \delta \vec{U} \) \quad \text{Virtual displacement vector in the global reference system}

\( \delta \vec{u} \) \quad \text{Virtual displacement vector in the element reference system}

\( \vec{e}_{1i} \) \quad \text{Unit vector in the x local axis for element } i

\( \vec{e}_{2i} \) \quad \text{Unit vector in the y local axis for element } i

\( \vec{e}_{3i} \) \quad \text{Unit vector in the z local axis for element } i

\( \vec{F} \) \quad \text{Total external forces of the complete FEM system}

\( \vec{F}_{1i} \) \quad \text{Element } i \text{ resultant gravity and buoyancy force per unit of unstretched length in the global reference system}

\( \vec{f}_{1i} \) \quad \text{Element } i \text{ resultant gravity and buoyancy force per unit of unstretched length in the local reference system}

\( \vec{F}_1 \) \quad \text{Resultant gravity and buoyancy force per unit of unstretched length in the global reference system}

\( \vec{F}_{2i} \) \quad \text{Element } i \text{ tangential drag force per unit of unstretched length in the global reference system}

\( \vec{f}_{2i} \) \quad \text{Element } i \text{ tangential drag force per unit of unstretched length in the local reference system}

\( \vec{F}_2 \) \quad \text{Hydrodynamic tangential drag force per unit of unstretched length in the global reference system}

\( \vec{F}_{3i} \) \quad \text{Element } i \text{ normal drag force per unit of unstretched length in the global reference system}

\( \vec{f}_{3i} \) \quad \text{Element } i \text{ normal drag force per unit of unstretched length in the local reference system}
\( \vec{F}_3 \)  Hydrodynamic normal drag force per unit of unstretched length in the global reference system

\( \vec{F}_4 \)  Hydrodynamic inertial force per unit of unstretched length in the global reference system

\( \vec{F}_i \)  Element total external force vector in the global reference system

\( \vec{f}_i \)  Element total external force vector in the local reference system

\( \vec{n}_1 \)  Vector in the x local axis before normalization

\( \vec{N}_2 \)  Vector used in the calculation of the element local reference unit vectors

\( \vec{n}_2 \)  Vector in the y local axis before normalization

\( \vec{n}_3 \)  Vector in the z local axis before normalization

\( \vec{P}_i \)  Element \( i \) nodal displacement vector in the global reference system

\( \vec{p}_i \)  Element \( i \) nodal displacement vector in the element reference system

\( \vec{R} \)  Acceleration vector in the global reference system

\( \vec{r} \)  Acceleration vector in the element reference system

\( \vec{R} \)  Velocity vector in the global reference system

\( \vec{r} \)  Velocity vector in the element reference system

\( \vec{R} \)  Position vector in the global reference system

\( \vec{r} \)  Position vector in the element reference system

\( \vec{R}_0 \)  Initial position vector of the point \( P \) in the global reference system

\( \vec{t} \)  Vector tangential to the cable at point \( P \)

\( \vec{U} \)  Displacement vector in the global reference system

\( \vec{u} \)  Displacement vector in the element reference system

\( \vec{V} \)  Relative velocity of the water with respect to the cable in the global reference system

\( \vec{v} \)  Relative velocity of the water with respect to the cable in the local reference system
\( \vec{V}_i \) Element \( i \) nodal vector for the relative water velocity in the global reference system

\( \vec{v}_i \) Element \( i \) nodal vector for the relative water velocity in the local reference system

\( \vec{V}_n \) Normal component of the relative velocity between the water and the cable in the global reference system

\( \vec{V}_t \) Tangential component of the relative velocity between the water and the cable in the global reference system

\( \vec{X} \) Acceleration vector of the complete FEM system in the global reference system

\( \vec{X}_i \) Element \( i \) nodal acceleration vector in the global reference system

\( \vec{x}_i \) Element \( i \) nodal acceleration vector in the element reference system

\( \vec{x}_i \) Element \( i \) nodal velocity vector in the element reference system

\( \vec{X} \) Velocity vector of the complete FEM system in the global reference system

\( \vec{X}_i \) Element \( i \) nodal velocity vector in the global reference system

\( \vec{X} \) Position vector of the complete FEM system in the global reference system

\( \vec{X}_i \) Element \( i \) nodal position vector in the global reference system

\( \vec{x}_i \) Element \( i \) nodal position vector in the element reference system
Chapter 1

Introduction

1.1 Offshore wind energy

Wind energy has experienced an increasing importance as energy source in Europe since the first 80’s of the past century. By 2014, over 240,000 wind turbines were in operation in the world, providing 4% of the world’s energy and with an installed capacity of more than 336 GW, mainly in China, the United States, Germany, Spain and Italy [1]. Offshore wind energy started his development in the first 2000’s and this growth was consolidated at the end of the first decade of the century. An important factor to impulse the installation of wind turbines in offshore location is the increasingly degree of occupation of the potentially available onshore locations as wind energy develops. In addition, the installation of offshore wind turbines has several advantages in comparison to onshore locations. The energy yield of a wind turbine installed in open sea is, in general, above the production of an onshore location due to higher and steadier winds. The visual and audible impact of a wind farm is less restrictive for the design in offshore than in onshore wind turbines. Finally, most of the world’s population is located close to the coastline and transmission losses are low.

Figure 1.1 shows the evolution in time of offshore wind installations in Europe. Of the total 128.8 GW of installed wind energy capacity in the EU by the end of 2014, only around 8 GW are located offshore (6.2%). Nevertheless, the growing tendency is clear and offshore wind energy will play a very important role in the future: around 12.5% of the new wind energy capacity installed during 2014 in Europe was offshore [18].

Several studies such as [5] suggest that floating substructures could achieve a Levelized Cost of Energy (LCOE) comparable with fixed substructures for water depths greater than 50 m. For higher depths floating technology would be more competitive. Nowadays the floating technology is still in its infancy and requires research and investment to be commercially competitive.

For these reasons, most of offshore installations use bottom fixed substructures and are located at shallow waters. Figure 1.2 shows the sea depth around Europe. The North sea, with
depths below 50 m in a great part of its extension is specially suitable for offshore bottom fixed wind turbines and a great part of the existing offshore wind farms are located in this area.

Table 1.1 shows the number of offshore farms, turbines and MW fully connected to the grid by the end of 2014. UK is the country with the highest amount of offshore capacity, with 55.9% of the total MW installed in Europe. Denmark is the second country in Europe with 15.8%, followed by Germany with 13%. Spain has only one offshore wind turbine connected to the grid, due to the great depth of the sea that surrounds the Iberian Peninsula, that makes fixed bottom platforms unfeasible from an economic point of view. Spain, and other countries from the South of Europe, will have to wait for the development of a mature floating platform technology to increment the importance of the offshore wind energy in their energy balance.

Current targets of the EU for offshore wind are 40 GW of installed power by 2020 and 150 GW by 2030. These objectives will be accomplished mainly using conventional bottom fixed substructures in depths below 50 m. Average water depth of offshore wind farms has increased from 12.2 m in 2009 to 22.4 m in 2014, which also means an increase in the cost of the bottom fixed substructures. If this tendency continues in the next few years, the 2030 objective will require a greater proportion of more technologically mature floating substructures though the offshore wind deployment is expected to be still dominated by fixed substructures in this period. From 2030, it is likely that locations shallow enough for cost efficient fixed structures will start to scarcer, and floating platforms will have a starring role in the offshore wind energy scenario.
Table 1.1: Number of offshore wind farms, turbines and MW fully connected to the grid at the end of 2014

<table>
<thead>
<tr>
<th>Country</th>
<th>BE</th>
<th>GE</th>
<th>DK</th>
<th>ES</th>
<th>FI</th>
<th>IE</th>
<th>NL</th>
<th>NO</th>
<th>PT</th>
<th>SE</th>
<th>UK</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of farms</td>
<td>5</td>
<td>16</td>
<td>12</td>
<td>1</td>
<td>2</td>
<td>1</td>
<td>5</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>25</td>
</tr>
<tr>
<td>Number of turbines</td>
<td>182</td>
<td>258</td>
<td>513</td>
<td>1</td>
<td>9</td>
<td>7</td>
<td>124</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>91</td>
<td>1,301</td>
</tr>
<tr>
<td>Capacity installed (MW)</td>
<td>712</td>
<td>1,048.9</td>
<td>1,271</td>
<td>5</td>
<td>26</td>
<td>25</td>
<td>247</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>212</td>
<td>4,494.4</td>
</tr>
</tbody>
</table>

1.2 Typologies of offshore wind turbines substructures

As has been mentioned in the previous Section, nowadays, bottom fixed are the most common substructures for offshore wind turbines. The main types of bottom fixed substruc-
CHAPTER 1. INTRODUCTION

Figure 1.3: Wind turbine fixed substructures main concepts. From left: monopile, gravity base, tripod and jacket.

... are illustrated in Figure 1.3. The monopile is the most usual fixed structure for depths below 20 m. It is basically a pipe that extends the turbine’s tower into the seabed. The installation is performed by piling, which can produce a serious noise pollution. Gravity base substructures (GBS) are also widely used in depths up to 30 m. It consists on a very massive base that rests on the seabed providing stability to the structure connected to it. The main advantage of this structure is that it can be built at land and then towed to the location and submerged by adding ballast. For higher depths than 30 m, space frame substructures as tripods or jackets are usually more cost efficient. These kind of substructures are composed by steel pipes. Tripods have a main central column with three legs that are driven into the seabed. Jackets are more transparent structures because there is no central column and, consequently, less steel is needed.

Relating the floating wind turbines, three main topologies exist depending on how the system stability is achieved. These three concepts are illustrated in Figure 1.4. In the spar design, ballast is located in the lower part of the platform to bring down the the center of gravity of the system below the center of buoyancy. If the platform is displaced from the vertical position, a restoring moment created by gravity and buoyancy forces provides stability. In this type of platforms, the area that pierces the water surface is minimized to decrease the interaction with the waves. Spars are typically easy to manufacture, but the large draft (can be more than 100 m) can create logistic problems. In the semisubmersible concept, there is an important amount of volume of the platform at the water surface level. These volumes are clearly displaced from the platform center increasing the moment created by buoyancy forces and giving stability to the platform. The low draft of semisubmersibles simplifies the installation process. Finally, Tension Leg Platforms (TLP’s) are relatively light platforms with respect to its volume with an excess of buoyancy force that is compensated by the tension of the mooring lines. TLP’s usually have a shallow draft and require less...
material to be constructed, but the lines and the anchors experiment very high tensions. In addition, the installation process can be challenging and the failure of a tendon during the wind turbine operation represents a critical incident.

1.3 Mooring lines typologies

Mooring systems are composed of mooring lines that are attached to the platform in a point called fairlead and have the lower ends anchored to the seabed. Mooring systems are responsible of the station keeping of floating platforms, contributing to the stability in the case of TLP designs.

The lines can be made up of chain, wire or synthetic rope. The chain is the most common material in depths up to 300 m. For higher depths, wire rope can be more adequate because it is lighter and more flexible. The synthetic fiber rope is lighter than chain or wire, and is typically used in combination with them for mooring systems at deep waters (2000 m or more).

Basically, two main mooring configurations for the mooring systems exist: the catenary
mooring systems, used in spar and semisubmersible concepts, and the taut-leg mooring systems, used in TLP designs (see Figure 1.4). In addition, some platforms have adopted a semi-taut mooring system, which is a hybrid between the taut and the catenary moorings. The catenary is the most common mooring configuration and receives this name from the shape that a hanging chain or cable assumes under its own weight when it is supported only at its ends. In catenary mooring systems, the lines are relatively long in comparison with the water depth and part of them are lying in the seabed. In consequence, the anchor only experiments horizontal forces. The restoring forces in the catenary mooring systems are generated by the gravity force acting on the suspended portion of the lines and by the changes in the line shape due to the platform motion. As the water depth increases, the length and the weight of the lines increases rapidly, becoming excessive for deep waters.

In taut mooring systems, the lines are pre-tensioned and the restoring forces are generated mainly through the axial stiffness of the mooring lines rather than by geometry changes. The stiffness of taut mooring systems is more linear than in catenary systems and allows a better platform offset control under mean load. In addition, the mooring lines have to be elastic enough not to suffer overloading under the platform motions induced by the waves. Semi-taut mooring lines are a combination of taut moorings with catenaries. Semi-taut and taut mooring systems require less seabed space and shorter lines than the catenary, resulting in material saving and smaller footprint. Although they need more expensive anchors due to the high vertical loads and the installation procedure is challenging, it could be a more cost-efficient solution than the catenary system for high depth locations.

Mooring systems can also be classified depending on the number of lines and their distribution. Figure 1.5 represents the different types of mooring systems according to the distribution of the lines. Most of the floating wind turbines designs use spread mooring systems, that consist of multiple lines attached to the platform on several locations, usually with a symmetrical distribution. This configuration restricts the rotation of the structure in the horizontal plane due to wind, waves and currents providing an almost constant heading to the structure. The lines of a spread mooring system can be equally spread around the floating structure or grouped. The grouped moorings provide better redundancy against the failure of a line. Some floating wind turbines designs, in particular spars, have proposed a single point mooring system, where all the lines are connected to the same point of the platform, that is free to rotate in horizontal plane. In these cases the lines are sometimes attached to a turret with bearings to allow the structure rotation. This configuration allows the system to adjust to the prevailing environment, minimizing loads in harsh multi-directional environments.
Mooring systems can be also divided into passive systems, that do not need further adjustments to support adverse environmental conditions, and active systems, where the tension is adjusted depending on the severity of the environmental conditions.

1.4 Reference system and motions of a floating platform

Six degrees of freedom corresponding to the six modes of motion as rigid solid of a floating wind turbine are represented in Figure 1.6. The three translational degrees of freedom are defined based on an inertial reference system with the X axis pointing towards the main wind and wave direction, the Y axis pointing to the left when looking downwind and the Z axis pointing vertically. In the offshore wind energy sector, the origin of this degree of freedom is typically located at the SWL and at the centreline of the wind turbine’s tower when the platform is at its undisplaced location. The rotational degrees of freedom are defined based on a reference system that is fixed to the floating wind turbine and is coincident with the inertial reference system when the platform is in its undisplaced position. The six rigid-body degrees of freedom have the following particular names in the naval terminology and are defined as:

- Surge: Translation along the main wind/wave direction (inertial X axis)
- Sway: Translation along the lateral axis (inertial Y axis)
- Heave: Translation along the vertical axis (inertial Z axis)
- Roll: Rotation about the wind turbine fixed X axis
- Pitch: Rotation about wind turbine fixed Y axis
CHAPTER 1. INTRODUCTION

![Diagram of floating platform degrees of freedom](image)

Figure 1.6: Floating platform degrees of freedom (Modified from [52])

- Yaw: Rotation about the wind turbine fixed Z axis

This is the reference system and the terminology for the floating platform motions used in this dissertation.

1.5 **Current operational and planned floating wind turbines**

The first floating wind turbine prototype in full scaled was the Hywind project, which consisted in a spar-buoy concept installed in 2009 in the coast of Norway. Since then, a few full scaled demonstration projects have been developed and currently several full scale prototypes are in production or planned to be installed. These full scale demonstration projects are summarized in Table [12].
### Table 1.2: List of operational and planned floating wind turbines [42] and [74]

<table>
<thead>
<tr>
<th>Project Name</th>
<th>Country</th>
<th>Manufacturer</th>
<th>Developer</th>
<th>Technology</th>
<th>Turbine Capacity</th>
<th>Turbine Manufacturer</th>
<th>Stage</th>
<th>Commissioning</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hywind</td>
<td>Norway</td>
<td>Statoil</td>
<td>Statoil</td>
<td>Spar</td>
<td>2.3 MW</td>
<td>Siemens</td>
<td>Prototype</td>
<td>2009</td>
</tr>
<tr>
<td>Sway</td>
<td>Norway</td>
<td>Sway A/S</td>
<td>Sway A/S</td>
<td>Spar</td>
<td>0.015 MW</td>
<td>Sway A/S</td>
<td>Pilot</td>
<td>2011</td>
</tr>
<tr>
<td>WindFloat Phase 1</td>
<td>Portugal</td>
<td>Principle Power</td>
<td>EDP/R/Repsol</td>
<td>Semisub.</td>
<td>2 MW</td>
<td>Vestas</td>
<td>Prototype</td>
<td>2011</td>
</tr>
<tr>
<td>Kabashima Island</td>
<td>Japan</td>
<td>Toda Corporation</td>
<td>Toda Corporation, Fuji Heavy Industries, Univ. Kyoto, J-Power</td>
<td>Spar</td>
<td>2 MW</td>
<td>Hitachi</td>
<td>Prototype</td>
<td>2013</td>
</tr>
<tr>
<td>VolturnUS 1/8 prototype</td>
<td>USA</td>
<td>DeepCWind Consortium</td>
<td>DeepCWind Consortium</td>
<td>Semisub.</td>
<td>0.02 MW</td>
<td>Renewgy</td>
<td>Pilot</td>
<td>2013</td>
</tr>
<tr>
<td>GOTO</td>
<td>Japan</td>
<td>Mitsui</td>
<td>Marubeni</td>
<td>Spar</td>
<td>2 MW</td>
<td>Hitachi</td>
<td>Prototype</td>
<td>2013</td>
</tr>
<tr>
<td>Fukushima</td>
<td>Japan</td>
<td>MHI</td>
<td>Marubeni</td>
<td>Spar</td>
<td>7 MW</td>
<td>MHI</td>
<td>Prototype</td>
<td>2015</td>
</tr>
<tr>
<td>IDEOL</td>
<td>France</td>
<td>IDEOL</td>
<td>Floatgen</td>
<td>TLP</td>
<td>2 MW</td>
<td>Gamesa</td>
<td>Prototype</td>
<td>2015</td>
</tr>
<tr>
<td>GICON-SOF</td>
<td>Germany</td>
<td>GICON</td>
<td>GICON</td>
<td>TLP</td>
<td>2.3 MW</td>
<td>Siemens</td>
<td>Prototype</td>
<td>2015</td>
</tr>
<tr>
<td>Sea Twirl</td>
<td>Sweden</td>
<td>Sea Twirl</td>
<td>Sea Twirl</td>
<td>Spar</td>
<td>0.03 MW</td>
<td>Sea Twirl</td>
<td>Pilot</td>
<td>2015</td>
</tr>
<tr>
<td>Sea Twirl</td>
<td>Sweden</td>
<td>Sea Twirl</td>
<td>Sea Twirl</td>
<td>Spar</td>
<td>0.03 MW</td>
<td>Sea Twirl</td>
<td>Pilot</td>
<td>2015</td>
</tr>
<tr>
<td>MODEC</td>
<td>Japan</td>
<td>MODEC</td>
<td>MODEC</td>
<td>Semisub.</td>
<td>0.5 MW</td>
<td>MODEC</td>
<td>Pilot</td>
<td>2015</td>
</tr>
<tr>
<td>Fukushima</td>
<td>Japan</td>
<td>Japan Marine United</td>
<td>Marubeni</td>
<td>Semisub.</td>
<td>5 MW</td>
<td>Hitachi</td>
<td>Prototype</td>
<td>2016</td>
</tr>
<tr>
<td>Blue H</td>
<td>Netherlands</td>
<td>Blue H group</td>
<td>Blue H group</td>
<td>TLP</td>
<td>5-7 MW</td>
<td>-</td>
<td>Prototype</td>
<td>2018</td>
</tr>
<tr>
<td>Vertiwind/Nemuphar /</td>
<td>France</td>
<td>Nemuphar</td>
<td>EDF</td>
<td>semisub.</td>
<td>2.6 MW</td>
<td>Nemuphar</td>
<td>Prototype</td>
<td>2018</td>
</tr>
<tr>
<td>Maine Aqua Ventus I</td>
<td>USA</td>
<td>UMaine</td>
<td>UMaine</td>
<td>Semisub.</td>
<td>6 MW</td>
<td>-</td>
<td>Prototype</td>
<td>2018</td>
</tr>
<tr>
<td>SEA REED</td>
<td>France</td>
<td>DCNS</td>
<td>DCNS/Alstom</td>
<td>Semisub.</td>
<td>6 MW</td>
<td>Alstom</td>
<td>Prototype</td>
<td>2018</td>
</tr>
<tr>
<td>PelaStar</td>
<td>France</td>
<td>Glosten</td>
<td>Glosten</td>
<td>TLP</td>
<td>6 MW</td>
<td>Alstom</td>
<td>Prototype</td>
<td>2018</td>
</tr>
</tbody>
</table>

Figure [74] shows the proportion of platform typologies in the existing full scale demonstrative projects. The distribution between the three main typologies is balanced, with a slight prevalence of semisubmersibles, revealing that none of the technologies have been imposed over the others and all of them have a range of application depending on the conditions.
CHAPTER 1. INTRODUCTION

1.6 Design standards for floating structures

The companies that develop new technologies as floating wind turbines are usually reluctant to share sensitive information or lessons learned. The development of standards for the design of offshore floating structures allows to establish a consensus in the industry on design principles that considers the experience of engineers and companies. The application of the guidelines that consider the particular aspects of floating wind turbines facilitates the economical optimization of the designs.

DNV GL has developed the DNV-OS-J103 standard that is specific for the design of floating wind turbines. This guideline was defined within an industry joint project between 2011 and 2013 with the participation of some of the most relevant actors of the sector such as: Alstom, Gamesa, Statoil-Hydro, Glosten Associates, Sasebo Heavy Industries, Principle Power, Iberdrola, Navantia, STX Offshore & Shipbuilding, and Sumitomo Metal. This standard has to be complemented with the more general guideline DNV-OS-J101 and together provide a set of principles, technical requirements and guidance for the design, the construction, the inspection during the service and the transportation. In addition, specific standards exist for the design of tendons and mooring lines.

DNV GL standards are mainly used in Europe. In America, an equivalent set of guidelines have been developed by the American Bureau of Shipping (ABS), providing guidance in similar aspects of the design process.

The IEC 61400 standards are published by the International Electrotechnical Commission (IEC), regarding wind turbines. They consist in a set of design requirements to ensure that wind turbines are correctly engineered against damage from hazards during their lifetime. The IEC 61400-3 Edition 1 is the most commonly used standard for the calculation of loads for floating wind turbines.
1.7 Integrated codes for offshore wind turbines

The load calculation of floating offshore wind turbine requires time-domain simulation tools taking into account all the phenomena that affect the system such as aerodynamics, structural dynamics, hydrodynamics, control actions and the mooring lines dynamics. These effects present couplings and are mutually influenced. The computational tools able to compute these coupled effects are called integrated or coupled codes. Some literature also refers to these tools as aero-hidro-servo-elastic codes.

The integrated codes for the simulation of floating wind turbines use different approaches to describe the mooring system behaviour coupled with the platform motions. Some of them are simplified methods as the quasi-static approach or the force-displacement relationships. Other models represent the full dynamics equations of the lines, considering effects as inertia, added mass or the water-line drag that are not captured by the simplified methods, though this means a higher computational effort. A more detailed description of the integrated codes and mooring models is provided in Chapter 3.

Several studies have pointed out the need of dedicated and comprehensive studies of the dynamics of mooring lines specifically focusing on coupled codes for floating wind turbines, see for example [63] and [61]. The fatigue and ultimate loads provided by integrated tools are used for the structural design of the different components of the wind turbine, consequently treated as inputs for the structural computation performed by Finite Element Method (FEM). The level of loads have a great influence on the design of the components. For this reason, the accuracy and reliability of the integrated simulation codes, including the mooring system models, have an important impact on achieving an optimized and cost-effective floating wind turbine design.
Chapter 2

Objectives and methodology

2.1 Objectives

The objective of this dissertation is to investigate computational and experimental methodologies to evaluate mooring dynamics and implement them in integrated codes for the simulation of floating wind turbines. With these tools, conclusions on the applicability of mooring dynamics to the design of floating wind turbines will be obtained. In particular, the following goals are pursued in this research:

- Improvement of integrated simulation tools for floating wind turbines by developing dynamic mooring lines models
- Development, implementation and verification of mooring lines testing methodologies
- Validation of simulation tools for mooring dynamics against experimental data
- Validation of the coefficients for computation of drag and friction forces in chains provided by the guidelines and the bibliography
- Evaluation of the influence of mooring dynamics on the fatigue and ultimate loads calculated for different floating wind turbine concepts according to certification guidelines

The development of accurate and verified simulation tools will provide more exact loads to design the floating wind turbine components, contributing to the optimization of the structure and the cost reduction. In addition, the characterization of the impact of the mooring dynamics on the results of the simulation tools will allow designers to choose when these effects should be considered and when lower complexity tools can be used saving computational time.

These objectives can be also expressed as the following questions to be answered in the conclusions of our research:
CHAPTER 2. OBJECTIVES AND METHODOLOGY

- Can mooring dynamics be efficiently included in the integrated simulation tools used for design and certification of offshore floating wind turbines?
- How can mooring dynamic models be experimentally tested and validated?
- Can computational models represent accurately the tension and motions of mooring lines?
- Are the values provided by guidelines and bibliography for drag and friction coefficients adequate for the representation of the real mooring dynamics?
- What is the influence of mooring dynamics in the level of loads for the different topologies of floating wind turbines?
- When mooring dynamics must be included in the integrated simulation of floating wind turbines?

2.2 Methodology

The following tasks have been performed to fulfill the objectives enumerated in the previous Section 2.1 and are described in the following chapters of this dissertation:

- In Chapter 3 a revision on the current state of development of integrated codes for floating wind turbines is provided with particular attention to their capabilities on the simulation of mooring dynamics. It is also reviewed the existing studies performed on the experimental validation of mooring line codes. Finally, past studies on the influence of mooring dynamics on the loads of floating wind turbines are examined. This bibliographic revision served as a starting point for this research, identifying the needs of development and research that are subsequently carried out in Chapters 4, 5 and 6.

- In Chapter 4 a dynamic mooring lines code based on a Lumped Mass formulation is developed. The non linear equations of motion of a line are discretized using the Finite Element Method and solved in the time domain. The equations are implemented in a code called OPASS (Offshore Platforms Anchorage System Simulator). The OPASS code is coupled with the FAST code for integrated simulations of floating wind turbines.

- The dynamic mooring lines code is experimentally validated in Chapter 5. The experiment set up consists on a chain submerged with one end anchored at the bottom of the basin and the suspension point at the still water level. The suspension point is excited with a prescribed periodic motion at different frequencies and configurations of the line. The tension at the fairlead and the motion at several positions along the length of the
line is measured and compared with the numerical predictions of equivalent simulation cases.

- Chapter 6 quantifies the influence that including mooring dynamics have on the calculation of fatigue and ultimate loads for floating wind turbines. The OPASS code coupled with FAST is used to compute the loads including mooring dynamics. These results are compared with the loads calculated also with FAST but using a lower complexity quasi-static approach. The computation of the loads is done for the three main existing floating wind turbines topologies (semisubmersible, spar-buoy and tension leg platform) and according to the methodologies and guidelines required by the certification entities within the offshore wind energy industry. More than 20,000 simulation cases are launched and postprocessed for this study.

- Finally, the conclusions of this research and the future lines of work are presented in Chapter 7.
Chapter 3

State of the art

3.1 Overview on integrated codes for floating wind turbines

The introductory Chapter highlighted the importance of accurate and reliable integrated simulation tools for the cost-efficient design of floating wind turbines. The simulation of the dynamics of a floating wind turbine has to integrate all the phenomena that can influence the behaviour of the system as aerodynamics, hydrodynamics, control, structural dynamics, mooring lines dynamics, etc. Each of these effects are mutually influenced and a reliable calculation should evaluate them in a coupled way. The influence of mooring lines over the global dynamics of a floating wind turbine is very important and therefore, an accurate simulation of the mooring system within the integrated codes can be fundamental for a precise description of the floating wind turbine dynamics and for the load calculation in the different components of the system.

The aero-servo-hydro-elastic codes are composed by different submodels representing all the physics that affect the wind turbine dynamics as the external conditions (wind and waves), the aerodynamics, the structural dynamics, the hydrodynamics or the dynamics of the mooring lines, taking also into account the wind turbine control strategy. The following subsections give an overview of the models used in the integrated codes.

3.1.1 External conditions

The external conditions that affect a floating wind turbine are the wind loading, the waves and the currents. The codes have to generate the wind inflow including a wide variety of conditions: steady wind, turbulent winds, gusts, vertical shear, direction changes, etc. Different turbulent models such as Von Karman, Kaimal or Mann are usually available in the codes.
For the wave conditions, the codes have to generate regular and irregular waves with different periods and heights. Airy theory is widely used to describe linear waves (see, for example, [29] or [67]). It produces sinusoidal sea surface elevation but does not provide any information about the kinematics over the medium sea level. Several methods exist to extrapolate the wave kinematics to the actual sea surface called stretching methods (see, for example, [19]). Some codes generate waves including second order effects to describe higher amplitude waves or low sea depth situations. Stream function is a wave theory that describes highly non-linear waves up to the breaking limit [21].

Currents are included in the described models as they could have great influence on the system response. These currents can be near-surface or sub-surface and include different profiles to define the variation of the velocity with depth.

### 3.1.2 Aerodynamics models

The most widely used model for the calculation of the aerodynamic loads in integrated codes is the Blade Element Method (BEM) (see for example: [78] or [15]). In this approach, the blade is divided into elements independent from the adjacent elements, where the forces are calculated using the airfoil lift and drag coefficients. The calculation does not consider tri-dimensional effects along the blade. Inertial effects of the flow are not taken into account. To overcome the model limitations, several semi-empirical corrections are typically applied to the results to take into consideration the hub and the blade tip losses [33], the yaw misalignment [32], high induction factors [32] or the effect of unsteady aerodynamics (dynamic stall) [53].

Some codes also apply the Generalized Dynamic Wake (GDW) Theory [85] that is based on potential theory and is able to describe the pressure distribution over the rotor including three-dimensional and unsteady effects.

Free-wake 3D modelling using vortex particle dynamics has been implemented in some integrated tools [2]. This approach assumes a potential flow field around the airfoil that is described through the distribution of discrete sources and vortices. Like BEM, forces are based on lift and drag coefficients data of the profile that are corrected considering the effects of blade rotation. It is expected that this theory provides better simulations in conditions where the wake dynamics are important or when there is a strong time dependency of the aerodynamic characteristics, as in flow misalignments.

### 3.1.3 Hydrodynamics models

There are two main hydrodynamic models that can be applied for the computation of the loads in the substructure: models based in the Morison equation and potential theory. Morison equation [65] is applicable to slender bodies. Basically, this load calculation method is based on the use of two coefficients: the inertia and the drag coefficients. To achieve a correct prediction of the loads these coefficients have to be carefully chosen or derived from
available experimental data. The linear potential theory (see, for example [29] or [67]) is a more advanced approach that can be applied to any geometry but it neglects the viscous drag effect. From the computational point of view, is more time-consuming than the Morison approach. The missing part of the damping due to viscous mechanisms can be included using the drag term of the Morison equation or increasing the overall platform damping with a linear damping matrix derived from experiments. The potential linear theory can be extended to take into account second order hydrodynamic effects [73] that can have an important role in the dynamics of certain floating platform concepts, particularly semisubmersibles, as it is demonstrated in [57] and [13].

3.1.4 Structural dynamics models

Some codes use a modal approach for the dynamics of the flexible bodies that compose the wind turbine structure. It consists on a reduction of the degrees of freedom of the flexible bodies to a limited amount of modal degrees of freedom. The motion of the flexible body is obtained by the linear combination of the body modes multiplied by the modal degrees of freedom. This approach is linear, and therefore it cannot represent large deflections where non-linearities arise, such as the ones in the long blades typically used in offshore wind turbines.

Other approaches as Finite Element Method (FEM) or Multi-Body systems are able to capture non-linear effects due to large deflections, though they have a higher computational cost.

3.1.5 Control

Most of the codes allow setting the parameters of very simple pitch control and rotor variable speed strategies. More advanced control strategies are usually implemented in dynamic link libraries (DLLs) and interfaced with the code.

3.1.6 Mooring line modeling approaches

The equations of motion of a submerged line are non-linear and cannot be solved analytically. Instead, either simplified approaches or numerical methods are applied. Simplified methods as the quasi-static approach [45] or the force-displacement relationships (for example in [75]) are widely used in integrated codes, because they provide the computational efficiency required for the simulation of the thousands of load cases required by the guidelines. The problem of solving the full dynamic representation of the lines requires a much higher computational effort and can be approached using different techniques. The fundamentals of the main mooring line models can be briefly described as:
• **Quasi-static**
  The static equations of the catenary at every time step of the simulation are solved, given the position of the fairlead where the line is attached to the platform. This model typically accounts for weight and buoyancy, axial stiffness, and friction from variable contact on the seabed, but does not consider the inertial effects, the hydrodynamic drag from waves, currents or the relative motion of the line within the fluid field.

• **Force-displacement relationship**
  In this model, non-linear spring stiffnesses are applied to the translational and rotational degrees of freedom of the platform. The force-displacement relationship has to be derived using a line analysis code and the results obtained would be similar to the quasi-static approach.

• **Dynamic model**
  This approach considers dynamic effects as inertia, added mass or hydrodynamic drag. Several formulations have been developed that can handle the dynamics of the line, as FEM, Finite Difference Method or Multi-Body models. The Lumped Mass model can be considered a variation of the FEM approach, where the mass of the elements is concentrated in the element’s adjacent nodes.

### 3.2 Revision of mooring line models in simulation codes

In this Section, the available integrated codes for the simulation of floating wind turbines are reviewed with particular attention to their mooring line modelling approaches. First, codes with simplified moorings models are described. Second, codes for floating structures including dynamic mooring models with a wind turbine model are discussed analyzing their features and limitations. Finally, specific codes for mooring lines are also commented.

#### 3.2.1 Codes for the simulation of floating wind turbines with simplified mooring line models

BLADED \(^1\) is one of the most popular commercial codes for the simulation of wind turbines. It has been developed by Garrad Hassan, originally for the simulation of onshore wind turbines. It has been extended to model floating structures using Morison equation for the computation of the hydrodynamic loading. In BLADED, the mooring lines are included through non-linear force-displacement relationships applied to the platform degrees of freedom. The aerodynamics of the rotor are based on BEM theory and, for the structural dynamics, a modal approach has been implemented. The modes of the tower and the blades
are calculated internally using a multi-body model. The HAWC2 code was developed by Risø-DTU and applies the BEM theory for the aerodynamics and the Morison equation for the hydrodynamics. A multi-body formulation is used for the structural dynamics and the mooring lines are represented by a force-displacement relationship. The code FAST is an open-source and free publicly available software developed by NREL. FAST allows the integrated simulation of onshore wind turbines and also offshore wind turbines both with a monopile or a floating substructure. The structural dynamics are modelled using a combined modal and multi-body approach. The aerodynamics are based on the BEM theory and are solved by a module called AeroDyn. The hydrodynamics of floating platforms are calculated by the module HydroDyn using potential theory. Quadratic drag based on the Morison equation can be added to take into account for the viscous forces. A drag coefficient along the vertical centreline of the platform has to be defined.

### 3.2.2 Codes for the simulation of floating wind turbines with dynamic mooring line models

Most of the tools for floating platforms with capabilities for the modelling of mooring dynamics were originated in the oil and gas industry and do not represent accurately the overall integrated dynamics of the complete system because they tend to simplify the coupling with the wind turbine dynamics. On the other hand, many aeroelastic tools created for the simulation of wind turbines lack of full capabilities for the computation of the different floating platform topologies including the mooring lines dynamics.

For this reason, most of the advanced tools for the integrated simulation of floating wind turbines with mooring dynamics are the result of coupling between codes independently developed. Each of these codes cover different parts of the complete problem as the hydrodynamics, the mooring dynamics or aerelasticity of the wind turbine.

Table summarizes the existing codes for the integrated simulation of floating wind turbines that include the dynamic simulation of the mooring system. Although most of these tools are the result of coupling several independent codes, there are also a few tools that have been developed specifically for the simulation of floating wind turbines.
### Table 3.1: Summary of floating wind turbines simulation tools including mooring dynamics

<table>
<thead>
<tr>
<th>Design Name</th>
<th>Code</th>
<th>Developer</th>
<th>Hydrodynamics Formulation</th>
<th>Coupled Code for Aerodynamics</th>
<th>WT Structural Dynamics</th>
<th>Coupling Developer</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Orca Flex</td>
<td>Orcina</td>
<td>FEM (LM)</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>CIVDISD</td>
<td>Texas A&amp;M University &amp; Offshore Dynamics Inc.</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>SHO</td>
<td>Marintek</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>SIMORIFLEX</td>
<td>Marintek</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>SIMORIFLEX</td>
<td>Marintek</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>SIMORIFLEX</td>
<td>Marintek</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>ANSYS</td>
<td>Inc.</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>Deepwater</td>
<td>Primaira</td>
<td>MBS</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
<tr>
<td>SIMPAK</td>
<td>University of Victoria &amp; Dynamic Systems Ltd</td>
<td>FEM</td>
<td>PF-CD/CD</td>
<td>FAST</td>
<td>BEM/GDW</td>
<td>yes</td>
<td>FAST</td>
</tr>
</tbody>
</table>

OrcaFlex is a commercial software developed by Orcina for the dynamic analysis of offshore structures, including catenary systems as mooring lines, flexible risers or umbilical cables. Orcina has coupled OrcaFlex with the FAST code. In the resulting tool, FAST is responsible for the simulation of aerodynamics, the control system and the structural dynamics of flexible elements as the rotor and the tower. OrcaFlex computes the hydrodynamics of the platform and the dynamics of the mooring system.

FAST has been used to model the wind turbine in a similar coupling with the CHARM3D code that computes the floating body and the mooring system dynamics. CHARM3D is a code for purchase developed by Texas A&M and Offshore Dynamics, Inc. for the simulation of moored floating structures, but has not capabilities for the coupled aerodynamics, structural dynamics and the control actions of a wind turbine.

SIMO/RIFLEX is one of the leading software for the analysis of offshore structures. Historically, it has been widely used in the oil and gas industry. RIFLEX is a finite element module for the dynamic analysis of mooring lines and SIMO is the module that performs the time domain hydrodynamic analysis of the floating platform, using potential theory. Morison elements can also be included. A module for the calculation of the aerodynamic forces on the wind turbine rotor based on the BEM theory has been developed. The forces at the blade elements are integrated and the resulting 3 forces and 3 moments are applied to the SIMO model as a user-specified external load.

The SIMO/RIFLEX code has also been coupled with HAWC2 for the inclusion of the wind turbine’s aerodynamic loading. This coupling has been implemented by Risø and Hydro Oil & Energy.

ANSYS Aqwa is a general purpose software for the analysis of offshore structures. The mooring system can be modeled using Finite Element Theory. ANSYS, Inc. has coupled his software with the code FLEX5 but neither the tool nor the documentation are publicly available. FLEX5 is a code created by DTU for the aeroelastic computation of onshore wind turbines. It is based in the BEM theory and uses a modal representation of the flexible bodies.

DeepLines is a general code for the simulation of offshore structures based in the potential theory or, alternatively, the Morison equation. It includes the dynamic analysis of the mooring system. Currently, PRINCIPIA, the company owner of the code, is collaborating with IFPEN (Institut Français du Pétrole - Energies Nouvelles) adding to the code capabilities for the simulation of floating wind turbines. The rotor aerodynamic model is based in the BEM theory and is coupled with the DeepLines code through a DLL generated by IFPEN. A preliminary verification of the tool has been performed within the IEA Annex 30 (OC4).

The integrated tool aNySimPHATASpro, developed by MARIN, is the result of the coupling of the code aNySimpro and PHATAS. aNySim has been developed by MARIN for the analysis of offshore structures and is able to consider the mooring dynamics using a Multi-Body formulation. PHATAS is a code developed by ECN for the aeroelastic simulation of onshore wind turbines.

SIMPACK is another commercial software for the dynamic simulation of mechanisms widely
used in many different industrial sectors. It is a general purpose program that has been
applied by the University of Stuttgart to model the structural dynamics of a floating wind
turbine, including the mooring lines. The aerodynamics and hydrodynamics have been in-
troduced in the model by coupling with SIMPACK the NREL’s AeroDyn and HydroDyn
codes \[62\].
ProteusDS \[37\] is a specific software for the dynamic analysis of mooring lines, using a Finite
Element formulation with the mass of the cable lumped at the nodes. It has been developed
by the University of Victoria and Dynamic Systems Analysis Ltd. ProteusDS has been com-
piled as a DLL and then coupled with the FAST code.
There are also a few specialized codes that have been originally conceived for the simulation
of offshore floating wind turbines including the mooring dynamics.
3DFloat is a specific code for the integrated analysis of onshore and offshore wind turbines,
including mooring system dynamics. It has been developed by IFE \[68\] and uses the BEM
theory for the aerodynamics, the Finite Element Method for the structural and mooring
lines dynamics and the Morison equation for the hydrodynamics.
Hydro-GAST is another integrated code developed by NTUA that uses potential theory for
the hydrodynamics, the BEM theory or, alternatively, the Free Wake Vortex theory for the
aerodynamics and a Finite Element formulation for the mooring lines and structural dynam-
ics \[2\], \[8\].

3.2.3 Other specific codes for the dynamic analysis of mooring lines

Many other programs specific for the simulation of the dynamics of mooring lines exist, but
they do not have capabilities for the simulation of the rest of components of a floating wind
turbine and they have neither been coupled with other tools for this purpose though this
could be done in the future. Some examples of specific codes for mooring lines based on
the Finite Element method are: MDD (Mooring Design and Dynamics) (Centre for Earth
and Ocean Research, University of Victoria) \[26\], Ariane7 (VeriSTAR) \[6\], CABLE3D (Texas
A&M), Flexcom V8 (MCS Kenny), HYBER (USFOS) \[40\] and SeaDyn (US Navy) \[88\]. The
code LINES, developed by the Massachusetts Institute of Technology (MIT) is based in a
Multi-Body formulation. The Massachusetts Institute of Technology, together with Woods
Hole Oceanographic Institute has also developed the WHOIcable code \[34\], that is based in
the Finite Difference method. This code is publicly available.
3.3 Revision of previous experimental validations of dynamic mooring line codes

The achievement of reliable simulation tools requires the validation of the computational predictions with experimental results. An experimental validation of the dynamic mooring line code developed in this research is part of the methodology to achieve the objectives of the study. Some previous work have been performed in the past on the validation of dynamic mooring lines based on Lumped Mass formulation.

The Lumped Mass method was first used to model a mooring line at the end of the 50’s, though the model was simple and neglected the elasticity of the material, the hydrodynamic drag forces or the seabed-line contact and the model was not validated. This model was improved at the beginning of the 80’s including elasticity, cable-seabed interaction and drag due to the relative motion of the cable with respect to the water. This method was validated with forced harmonic oscillation tests. The agreement with computational models was good, though the tests only covered a limited set of cases that did not consider slack conditions. In the same period, a lumped mass code was developed and verified against computations. The methodology to perform the experimental tests was based in the work described a few years before. Another comparison between a computational Lumped Mass model and a submerged line with prescribed harmonic displacements at the suspension point is described. This reference shows a good agreement at the line end tension between the experimental results and the calculation, but comparisons of the cases including slack condition are not shown. In the same research, a Lumped Mass model is compared with tension experimental results with good agreement for the Harmonic condition, but presents significant differences when the line loses tension. The importance in the mooring line dynamics modelling of the accurate identification of the cable elastic stiffness and the free falling velocity is highlighted. This work also presents a comparison between the experimentally measured tension and numerical computations showing good agreement. In dynamic simulations with Orcaflex are compared to experimental data of a mooring line with prescribed harmonic motion of the fairlead. The experiments include cases with and without current. Good agreement between tension in computations and experiments is achieved except for the cases where the line loses tension. The computed results here show spikes of much higher tension.
than in the experiments. More recently, a Lumped Mass mooring line code has been validated against tension measurements of a floating wind turbine mooring lines obtained with scaled tests in a wave tank [36].

3.4 Revision of previous research on the importance of mooring dynamics on floating wind turbine systems

Some studies have been published relating the influence of the mooring dynamics on the loads of floating wind turbines. One of the most complete is presented in [37], where simulations using a quasi-static mooring line model and a dynamic model are compared for three different floating wind turbines: the OC3-Hywind spar, the ITI Energy barge and the MIT/NREL TLP. The study concludes that mooring dynamics can affect the fatigue and ultimate loads and the adequacy of quasi-static model is dependent on the support configuration, being well suited for spars with natural periods below the peak wave period. The comparison is nevertheless limited by the fact that the calculation of extreme and fatigue loads is based in a few individual cases and each platform is located at a different sea depth (320 m, 150 m and 200 m). A previous study by Kallesøe et al. [48] compared simulations of the OC3-Hywind platform using quasi-static and dynamic mooring models, concluding that calculating loads with the quasi-static approach is conservative and the use of mooring dynamics can affect the resulting fatigue tower loads. Matha et al. [61] also performed a comparative study of fatigue equivalent loads for the OC3-Hywind spar based on a single load case computed with a quasi-static and a Multi-Body dynamic model. The turbine loads were slightly affected, but significant differences in the fairlead equivalent tensions were found. They also concluded that the importance of dynamic effects of the lines have to be further investigated with a comprehensive analysis based on the requirements and load cases specified in the guidelines. Masciola et al. [60] compared scaled experimental data of the OC4 semisubmersible platform with coupled simulations using quasi-static and dynamic moorings model, concluding that the lines dynamics have a limited importance in the motions of the platform, but are relevant for the line tension.

3.5 Concluding remarks

Many codes with capabilities for the dynamic simulation of mooring lines have been enumerated. Some of them are specialized software for the analysis of moorings dynamics and others are general codes for the modeling of floating structures, mainly coming from the naval or the oil and gas industries. Most of the integrated tools for the simulation of floating wind
turbines that include mooring dynamics are the result of the coupling of different existing software.

The design loads of the wind turbines have to be calculated based in several thousands of simulation cases that are specified by the certification guidelines. This high amount of computations requires a compromise in the simulation tools between the reliability of the tools and their CPU cost. From this point of view, the simulation tools created through the coupling of different codes can present some drawbacks such as a low computational efficiency or lack of robustness during the simulations.

The development of specialized tools specifically developed for the floating wind turbines simulation is a need for the sector that requires accurate and efficient tools for the design process. The inclusion of dynamic models for the mooring lines implies an important increment in the number of degrees of freedom of the equations to be solved, and therefore represents a challenge in terms of computational effort. Dynamic mooring models developed for the offshore wind energy sector should pay particular attention to the computational cost efficiency. The use of a Lumped Mass approach can have advantages reducing the simulation time.

There is a limited number of studies on experimental validation of codes for submerged lines. All of them are based on the measurement of the tension of the line, but none has measured the motion of the line. The main disagreements with the Lumped Mass code predictions are reported in situations where the line looses the tension and suddenly recovers producing a snap load. This is a highly dynamic and challenging case that should be included in the validation of a mooring line code.

The revision on the existing work on the influence of the mooring dynamics on the fatigue and ultimate loads reveals that there is a lack of systematic and comprehensive studies. Existing works have limitations and are based on a reduced number of simulation cases and platform models. The bibliographic sources point out the need of studies based on the methodologies required by the certification standards.
Chapter 4

Dynamic Simulation Code for Mooring Lines

4.1 Abstract

In this chapter, the development of a dynamic computer code for the simulation of mooring lines is described. The code has been called OPASS (Offshore Platform Anchorage System Simulator). The basic system of equations that represents the dynamics of a line are developed. These equations are discretized using the Finite Element Method to obtain the set of equations implemented in the OPASS code. The mathematical model has been oriented to the achievement of a computationally efficient code. For this reason, it has been selected a Lumped Mass approach that results in a global mass matrix that can be inverted with low computational cost. The code can be run as an stand alone tool or coupled with FAST for the simulation of a complete floating wind turbine.

4.2 Basic dynamic equations

A mooring line has one of his ends fixed to the seabed by an anchor and the other end, called fairlead, is attached to the floating platform. The physical model considered in this code for the mooring is an slender line with constant circular section and three translational degrees of freedom at each node. Inertia, hydrodynamic added mass, gravity, hydrostatics, wave kinematics, tangential and normal hydrodynamic drag, axial elasticity and structural damping are considered. The code neglects the bending stiffness, being suitable for the simulation of chains. Part of the line can be in contact with the seabed and the line-seabed contact and friction are included.

A coordinate system $l_0$ is defined in the cable as the distance along the unstretched length of the cable, from the anchor to the cable section to be considered, as it is shown in the figure.
As the cable is a very slender structure, shear forces can be neglected. If the bending and torsion stiffness are low enough to also be neglected, the only internal forces are the tension $T$ and the structural damping $F_D$. Both internal forces are always tangential to the cable. The external forces acting on the cable are the gravity, the buoyancy and the hydrodynamic drag forces. There is also an additional inertial force due to the volume of water displaced by the line in movement (added mass).

Let us consider an infinitesimal length of cable $dl$, at point $P$, that is located at a distance $l_0$ along the unstretched length of the cable. As can be seen in the figure, the forces acting on this portion of the cable are shown in the figure. The resultant force from hydrostatic pressure and gravity $\vec{F}_1$, is vertical. The hydrodynamic drag force is split into two components: normal and tangential to the cable. The tangential component is $\vec{F}_2$ and the normal component is $\vec{F}_3$. The inertial force coming from the added mass, $\vec{F}_4$ is supposed to have only a component normal to the cable. All these forces are expressed per unit of unstretched length and in the global reference system.

According to the figure, and considering the inertial forces in the balance of forces, we can write the following equation in the global coordinate system:

$$\gamma \ddot{\vec{R}} - \frac{\partial (T \vec{t})}{\partial l_0} - \frac{\partial (F_D \vec{t})}{\partial l_0} - \left( \vec{F}_1 + \vec{F}_2 + \vec{F}_3 + \vec{F}_4 \right) = 0 \quad (4.1)$$

Where $\gamma$ is the line mass per unit of cable unstretched length, $\ddot{\vec{R}}$ is the acceleration of point $P$ in the global reference system and, $\vec{t}$ is the vector tangential to the cable at point $P$ in the global reference system.

If we relate the displacements of the cable to an initial reference cable configuration $\vec{R}$, then the current position vector $\vec{R}$ of point $P$ can be expressed in the global reference system as:

$$\vec{R} = \vec{R}_0 + \vec{U} \quad (4.2)$$
Where $\vec{R}_0$ is the initial position vector of point $P$ at the reference line configuration $R$, and $\vec{U}$ is the displacement vector. The vector tangential to the line at point $P$ in equation (4.1) can be calculated as:

$$\vec{t} = \frac{\partial \vec{R}}{\partial l_0}$$

(4.3)

### 4.2.1 Elastic forces

For the infinitesimal element that we are considering, the axial deformation $\varepsilon$ is defined as:

$$\varepsilon = \frac{\partial l}{\partial l_0} - 1 = \left| \frac{\partial \vec{R}}{\partial l_0} \right| - 1$$

(4.4)

Where $l$ is the distance along the stretched length of the cable.

The tension at point $P$ can be obtained from the constitutive equation:

$$T = EA \varepsilon$$

(4.5)

Where $E$ is the material Young’s modulus and $A$ is the section area of the cable.

### 4.2.2 Structural damping forces

The structural damping is based on the Rayleigh model (see, for example, [41]). In general, Rayleigh model assumes the damping to be proportional to the mass and the stiffness. In this model, the term proportional to the mass is neglected and the damping force $F_D$ at point $P$ is considered only proportional to the stiffness $EA$ as it is commonly assumed in the modelling of mooring lines for integrated codes [55], [75]. The proportionality coefficient is $\beta$. Thus, we can formulate:

$$F_D = \beta E A \varepsilon$$

(4.6)

Where $\dot{\varepsilon}$ is the deformation velocity.
4.2.3 Gravity and hydrostatic forces

The gravity force is a body force: it acts throughout the volume of the cable. By contrast, the hydrostatic force is not a body force: it is produced by the integration of the hydrostatic pressure over the element. Nevertheless, it can be treated as a volume force if the element considered is totally surrounded by water and can be calculated according to Archimedes’ Principle. For a cable this is not strictly true, but as the diameter of the section is small in comparison with the length, the error induced by this assumption is negligible. The resultant force from hydrostatic pressure and gravity per unit of unstretched length, expressed by $\vec{F}_1$, is the weight of the cable minus the weight of the displaced volume of water per unit of unstretched length. The direction of the force is vertical. Thus:

$$\vec{F}_1 = \begin{bmatrix} 0 \\ 0 \\ -\gamma_r g \end{bmatrix}$$

and

$$\gamma_r = \frac{\rho_c - \rho_w}{\rho_c} \gamma$$

(4.8)

Where $g$ is the gravity constant, $\gamma_r$ is the equivalent mass per unit length of the cable submerged in water, $\rho_c$ is the density of the cable and $\rho_w$ is the density of the water.

4.2.4 Hydrodynamic forces

The hydrodynamic forces considered equation (4.1) are the tangential drag $\vec{F}_2$, the normal drag $\vec{F}_3$ and the hydrodynamic inertial force $\vec{F}_4$, that is also normal to the cable. These hydrodynamic forces are represented in the figure 4.3.

![Figure 4.3: Tangential drag, normal drag and added mass force along the cable](image)

Using the Morison equation for slender cylinders \[65\], the value of the tangential drag force per unit length of the unstretched cable can be calculated as:
\[ \vec{F}_2 = C_2 |\vec{V}_t| \vec{V}_t (1 + \varepsilon) = C_2 \left( |\vec{V} \cdot \vec{t}| \vec{V} \cdot \vec{t} \right) (1 + \varepsilon) \vec{t} \]
\[ \quad C_2 = \frac{1}{2} C_{dt} D \rho_w \]

Where \( C_{dt} \) is the tangential drag coefficient and \( D \) is the diameter of the cable. The vector \( \vec{V} \) is the relative velocity between the water and the cable and \( \vec{V}_t \) is the component tangential to the element, both expressed in the global reference system.

In a similar way can be obtained the drag force normal to the cable per unit length:
\[ \vec{F}_3 = C_3 |\vec{V}_n| \vec{V}_n (1 + \varepsilon) = C_3 \left( \vec{V} \cdot \vec{V} - \left( \vec{V} \cdot \vec{t} \right) \vec{t} \right)^{\frac{1}{2}} \left( \vec{V} - \left( \vec{V} \cdot \vec{t} \right) \vec{t} \right) (1 + \varepsilon) \]
\[ \quad C_3 = \frac{1}{2} C_{dn} D \rho_w \]

Where \( C_{dn} \) is the normal drag coefficient and \( \vec{V}_n \) is the component normal to the element of the relative velocity between the water and the cable.

The hydrodynamic inertial force per unit of unstretched cable length is:
\[ \vec{F}_4 = -C_4 (1 + \varepsilon) \left[ \vec{R} - \left( \vec{R} \cdot \vec{t} \right) \vec{t} \right] \]
\[ \quad C_4 = C_{mn} \frac{\pi D^2}{4} \rho_w \]

Where \( C_{mn} \) is the normal added mass coefficient.

Substituting equations (4.5), (4.6) and (4.11) into (4.1) finally results in:
\[ \gamma \vec{R} + C_4 (1 + \varepsilon) \left[ \vec{R} - \left( \vec{R} \cdot \vec{t} \right) \vec{t} \right] - \frac{\partial(E A \varepsilon \vec{t})}{\partial l_0} - \frac{\partial(\beta E A \varepsilon \vec{t})}{\partial l_0} - \vec{F}_1 - \vec{F}_2 - \vec{F}_3 = 0 \]

4.3 The Finite Element equations

4.3.1 Virtual Works principle

According to the Virtual Works Principle, the path followed by a system is the one for which the difference between the work performed by the forces along this path and other nearby paths is zero. If we apply a small (virtual) displacement that satisfies the boundary conditions \( \delta \vec{U} \) with respect to a certain configuration at time \( t \), the virtual work \( W_V \) done by the forces along the cable length must be zero. The virtual work can be obtained by multiplying equation (4.12) by the virtual displacement \( \delta \vec{U} \) and integrating along the cable.
length:

\[
W_V = \int_0^L \left\{ \left( \gamma \vec{R} + C_4(1 + \varepsilon) \left[ \vec{R} - (\vec{R} \cdot \vec{i}) \vec{i} \right] \right) \cdot \delta \vec{U} - \left( \vec{F}_1 + \vec{F}_2 + \vec{F}_3 \right) \cdot \delta \vec{U} + \\
+ EA \varepsilon \frac{\partial \delta \vec{U}}{\partial l_0} \cdot \vec{i} + \beta EA \varepsilon \frac{\partial \delta \vec{U}}{\partial l_0} \cdot \vec{i} \right\} dl_0 - \left[ EA \varepsilon \vec{i} \cdot \delta \vec{U} \right]_0^L - \left[ \beta EA \varepsilon \vec{i} \cdot \delta \vec{U} \right]_0^L = 0
\]  

(4.13)

Where \( L \) is the total cable length.\(^1\) The last two terms represent the work performed by the end forces at the initial and final faces of the cable. If the anchor and the fairlead of the cable are fixed or their displacements are prescribed, these terms are zero. Considering the boundary conditions of our problem, the anchor will remain fixed and the fairlead position will be determined by the platform displacements, thus, these terms are neglected in the following discussion.

### 4.3.2 Interpolation by shape functions

The mooring line is discretized into \( n \) finite elements using straight bar members. A bar has two main characteristics:

- The axial direction is much larger than the transversal directions.
- The bar resists an internal force in the axial direction

A local coordinate \( \xi_i \) is defined for each element \( i \). \( \xi_i \) is 0 at the beginning of the element and it is 1 at the end as is described in the figure \[1\] .

Thus, the position of a point \( P \) along the unstretched length of the cable, \( l_0 \), can be expressed as:

\[
l_0 \approx l_{0,i} + \xi_i L_i
\]  

(4.16)

\(^1\)The last four terms of equation (4.13), related to the elastic and structural damping forces, are obtained integrating by parts. For example, for the elastic forces, we can write:

\[
\nu = \delta \vec{U} \rightarrow \frac{dv}{dl_0} = \frac{\delta \vec{U}}{dl_0} \\
dv = -\frac{\partial EA \varepsilon \vec{i}}{\partial l_0} dl_0 \rightarrow \nu = -EA \varepsilon \vec{i}
\]  

(4.14)

Then:

\[
\int_0^L -\frac{\partial EA \varepsilon \vec{i}}{\partial l_0} \cdot \delta \vec{U} dl_0 = \int_0^L EA \varepsilon \frac{\partial \delta \vec{U}}{\partial l_0} \cdot \vec{i} dl_0 - [EA \varepsilon \vec{i} \cdot \delta \vec{U}]_0^L
\]  

(4.15)

The terms for the structural damping forces can be derived with a parallel method.
Where $L_i$ is the length of the element $i$ and $l_{0i}$ is the unstretched length to the initial node of the element $i$:

$$l_{0i} = \sum_{j=1}^{i-1} L_j$$  

These parameters are shown in the figure 4.4.

All the magnitudes are assumed to be continuous along the finite element and they are approximated by interpolation of the values at the element nodes based in the following linear shape functions:

$$N_1 = \xi_i$$
$$N_2 = 1 - \xi_i$$  

The interpolations of these magnitudes can be expressed in a compact way by means of the shape functions matrix $N$:

$$N = \begin{bmatrix} 1 - \xi_i & 0 & 0 & \xi_i & 0 & 0 \\ 0 & 1 - \xi_i & 0 & 0 & \xi_i & 0 \\ 0 & 0 & 1 - \xi_i & 0 & 0 & \xi_i \end{bmatrix}$$  

Then, for the element $i$, we can write:

$$\vec{r}(\xi_i, t) \approx N(\xi_i) \vec{x}_i(t)$$
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\[
\vec{u}(\xi_i, t) \simeq N(\xi_i) \vec{p}_i(t) \quad (4.21)
\]

\[
\delta\vec{u}(\xi_i, t) \simeq N(\xi_i) \delta\vec{p}_i(t) \quad (4.22)
\]

\[
\vec{r}(\xi_i, t) \simeq N(\xi_i) \vec{x}_i(t) \quad (4.23)
\]

\[
\vec{v}(\xi_i, t) \simeq N(\xi_i) \vec{v}_i(t) \quad (4.24)
\]

Where \(\vec{r}\) and \(\vec{r}'\) are the 3 x 1 position and velocity vectors in the element reference system and \(\vec{x}_i\) and \(\vec{x}_i'\) are the 6 x 1 element nodal position and element nodal velocity vectors, both in the element reference system. The vectors \(\vec{u}\) and \(\delta\vec{u}\) are the 3 x 1 displacement and virtual displacement vectors and \(\vec{p}_i\) and \(\delta\vec{p}_i\) are the respective 6 x 1 element \(i\) nodal displacement and element \(i\) nodal virtual displacement vectors, all in the element reference system. Finally, \(\vec{v}\) is the relative velocity of the water and \(\vec{v}_i\) is the element \(i\) nodal vector for the relative water velocity in the local reference system.

For the inertial forces, instead of linear shape functions, discontinuous step functions are defined, resulting on a Lumped Mass model. The resulting global mass matrix is composed by 3 x 3 submatrices located at the diagonal. This is an important advantage for the inversion of the matrix that can be done with much less computational effort.

\[
N' = \begin{bmatrix}
\psi_1 & 0 & 0 & \psi_2 & 0 & 0 \\
0 & \psi_1 & 0 & 0 & \psi_2 & 0 \\
0 & 0 & \psi_1 & 0 & 0 & \psi_2 \\
\end{bmatrix} \quad (4.25)
\]

Where:

\[
\begin{aligned}
\psi_1 &= 1 & \text{if } \xi_i \in [0, 0.5] \\
\psi_1 &= 0 & \text{if } \xi_i \in [0.5, 1]
\end{aligned}
\]

So, the accelerations within element \(i\) are approximated as:

\[
\ddot{\vec{r}}(\xi_i, t) \simeq N'(\xi_i) \ddot{\vec{x}}_i(t) \quad (4.27)
\]

Where \(\ddot{\vec{r}}\) is the acceleration vector and \(\ddot{\vec{x}}_i\) is the 6 x 1 element vector with the nodal accelerations, both in the local element system.

Finally, the matrix \(B\) is defined as the derivative of \(N\):

\[
B = \frac{\partial N}{\partial \xi_i} = \begin{bmatrix}
-1 & 0 & 0 & 1 & 0 & 0 \\
0 & -1 & 0 & 0 & 1 & 0 \\
0 & 0 & -1 & 0 & 0 & 1
\end{bmatrix} \quad (4.28)
\]
4.3.3 Local to global transformation matrix

The magnitudes in the local reference system of the \(i\) element and in the global reference system can be related through the \(3 \times 3\) local to global transformation matrix \(T_i\). To find the unit direction vectors, \(\vec{e}_{1i}, \vec{e}_{2i}\) and \(\vec{e}_{3i}\), that compose the local reference system attached to the bar element \(i\), together with the element initial node (node 1) and final node (node 2), an additional node has to be defined. The position of this third node is arbitrary, though it has to be located out of the element, to define the plane containing the \(\vec{e}_{2i}\) unit vector. The element \(i\) local reference system is illustrated in the figure 4.6.

\[
\begin{align*}
\vec{n}_1 & = \begin{bmatrix} x_2 - x_1 \\ y_2 - y_1 \\ z_2 - z_1 \end{bmatrix} \\
\vec{N}_2 & = \begin{bmatrix} x_3 - x_1 \\ y_3 - y_1 \\ z_3 - z_1 \end{bmatrix} \\
\vec{n}_3 & = \vec{n}_1 \wedge \vec{N}_2 \\
\vec{n}_2 & = \vec{n}_3 \wedge \vec{n}_1 
\end{align*}
\]  

(4.29)

Where \(\wedge\) denotes cross product.

Then, the unit vectors of the local reference system for the \(i\) element can be calculated as:

\[
\vec{e}_{1i} = \frac{\vec{n}_1}{|\vec{n}_1|} \quad \vec{e}_{2i} = \frac{\vec{n}_2}{|\vec{n}_2|} \quad \vec{e}_{3i} = \frac{\vec{n}_3}{|\vec{n}_3|} 
\]  

(4.30)

And then, the local to global transformation matrix is just:

\[
T_i = \begin{bmatrix} \vec{e}_{1i} & \vec{e}_{2i} & \vec{e}_{3i} \end{bmatrix}
\]  

(4.31)
In the case of the element nodal vectors, we define a 6 x 6 transformation matrix, \( T_I \) as:

\[
T_I = \begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
\end{bmatrix}
\] (4.32)

The matrices \( N \) and \( N' \) interpolate the nodal magnitudes in the element local reference system. By means of the transformation matrices, we can define new matrices \( N_{gi} \), \( N'_{gi} \) and \( B_{gi} \) to operate with the magnitudes at element \( i \) in the global system:

\[
N_{gi} = T_i NT_I^T
\] (4.33)

\[
N'_{gi} = T_i N'T_I^T
\] (4.34)

\[
B_{gi} = T_i BT_I^T
\] (4.35)

Thus:

\[
\vec{R}(\xi_i, t) \simeq N_{gi}(\xi_i) \vec{X}_i(t)
\] (4.36)

\[
\vec{U}(\xi_i, t) \simeq N_{gi}(\xi_i) \vec{P}_i(t)
\] (4.37)

\[
\delta \vec{U}(\xi_i, t) \simeq N_{gi}(\xi_i) \delta \vec{P}_i(t)
\] (4.38)

\[
\vec{R}(\xi_i, t) \simeq N_{gi}(\xi_i) \vec{X}_i(t)
\] (4.39)

\[
\vec{R}(\xi_i, t) \simeq N'_{gi}(\xi_i) \vec{X}_i(t)
\] (4.40)

\[
\vec{V}(\xi_i, t) \simeq N_{gi}(\xi_i) \vec{V}_i(t)
\] (4.41)

Where \( \vec{R} \) is the 3 x 1 velocity vector, and \( \vec{X}_i, \vec{X}_i \) and \( \vec{X}_i \) are the 6 x 1 element \( i \) nodal position, velocity and acceleration vectors, all in the global reference system. \( \vec{P}_i \) and \( \delta \vec{P}_i \) are the element nodal displacement and virtual displacement vectors, also in the global system. Finally, \( \vec{V}_i \) is the element nodal relative water velocity in the global reference system.

### 4.3.4 Definition of the adjacency matrix

We define for each element \( i \) an adjacency matrix \( A_i \), composed by elements with value 0 or 1, that relates the 6 x 1 element \( i \) nodal vectors with the whole 6\((n+1)\) x 1 discretized...
system global vector. So, the dimension of $A_i$ is $6 \times (n+1)$. If the nodes are numbered consecutively, then the adjacency matrix of the element $i$ has the following structure:

$$
\begin{bmatrix}
0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\
\end{bmatrix}
$$

\text{Index: } 1 \ldots i - 1 \ i \ i + 2 \ i + 3 \ i + 4 \ i + 5 \ i + 6 \ i + 7 \ \ldots \ 6n + 6

(4.42)

And then:

$$
\begin{align*}
\vec{X}_i &= A_i \vec{X} \\
\vec{X}_i &= A_i \vec{X} \\
\vec{X}_i &= A_i \vec{X} \\
\delta \vec{P}_i &= A_i \delta \vec{P}
\end{align*}
$$

(4.43) (4.44) (4.45) (4.46)

This is a mathematical way to express the assembly of the discretized matrices implemented in the code.

### 4.3.5 Discretization of elastic forces

As we have neglected the elastic forces acting at the end sections of the cable, the term for the work produced by the elastic forces $W_{\text{elastic}}$ in the equation (4.13) is:

$$
W_{\text{elastic}} = \int_0^L EA \varepsilon \left. \frac{\partial \delta \vec{U}}{\partial l_0} \right| \cdot \vec{t} \, dl_0
$$

(4.47)

The discretized element deformation $\varepsilon_i$ is obtained deriving the expression (4.30) and introducing it into equation (4.44):

$$
\varepsilon = \left| \frac{\partial \vec{R}}{\partial l_0} \right| - 1 \simeq \varepsilon_i = \sqrt{\frac{B_{gi} \vec{X}_i}{L_i} \cdot \frac{B_{gi} \vec{X}_i}{L_i}} - 1 = \sqrt{\vec{X}_i^T B_{gi}^T B_{gi} \vec{X}_i} - 1
$$

$$
\sum_{j=1}^{i-1} jL_j < l_0 \leq \sum_{j=1}^{i} jL_j
$$

(4.48)

On the other hand, from derivation of equation (4.38) we have:

$$
\frac{\partial \delta \vec{U}}{\partial l_0} \simeq B_{gi} \frac{\delta \vec{P}_i}{L_i} \sum_{j=1}^{i} jL_j < l_0 \leq \sum_{j=1}^{i} jL_j
$$

(4.49)
And the tangential vector \( \vec{t} \) can be expressed introducing into (4.3) the derivative of the expression (4.36):

\[
\vec{t} = \frac{\partial \vec{R}}{\partial l} = \frac{B_{gi} \vec{X}_i}{L_i (1 + \varepsilon_i)} \approx B_{gi} \vec{X}_i \\
\sum_{j=1}^{i-1} jL_j < l_0 \leq \sum_{j=1}^{i} jL_j
\]  

(4.50)

Introducing into (4.47) the expressions (4.43), (4.46), (4.49) and (4.50), we have:

\[
W_{\text{elastic}} = \sum_{i=1}^{n} \int_{0}^{1} EA \varepsilon_i \delta \vec{P}_T A_i B_{gi} A_i \vec{X} L_i (1 + \varepsilon_i) d\xi
\]  

(4.51)

4.3.6 Discretization of structural damping forces

The term in equation (4.43) for the work produced by the structural damping forces \( W_{\text{damp}} \) is:

\[
W_{\text{damp}} = \int_{0}^{L} \beta EA \varepsilon_i \frac{\partial \delta \vec{U}}{\partial l_0} \cdot \vec{t} dl_0
\]  

(4.52)

The element deformation velocity can be obtained by the time derivation of the expression (4.48):

\[
\dot{\varepsilon} \approx \frac{d\varepsilon_i}{dt} = \frac{\dot{X}_i B_{gi} B_{gi} \vec{X}_i}{L_i^3 (1 + \varepsilon_i)} \sum_{j=1}^{i-1} jL_j < l_0 \leq \sum_{j=1}^{i} jL_j
\]  

(4.53)

If we follow a parallel reasoning as to discretize the work of the elastic forces, we obtain the following expression:

\[
W_{\text{damp}} \approx \sum_{i=1}^{n} \int_{0}^{1} \beta EA \varepsilon_i \frac{\delta \vec{P}_T A_i B_{gi} B_{gi} \vec{X}_i}{L_i (1 + \varepsilon_i)} d\xi
\]  

(4.54)

And using (4.44) and (4.52), we have:

\[
W_{\text{damp}} = \sum_{i=1}^{n} \int_{0}^{1} \beta EA \frac{\delta \vec{P}_T A_i B_{gi} B_{gi} \vec{X}_i \vec{X}_i B_{gi} B_{gi} A_i \vec{X}}{L_i^3 (1 + \varepsilon_i)^2} d\xi
\]  

(4.55)

4.3.7 Discretization of external forces

The external forces considered are the gravity and the buoyancy \( \vec{F}_1 \), the tangential hydrodynamic force \( \vec{F}_2 \) and the normal hydrodynamic drag \( \vec{F}_3 \). The force produced by the hydrodynamic added mass is studied separately, as an inertial force.

Thus, the work due to these external forces in equation (4.43), \( W_{\text{external}} \), is:

\[
W_{\text{external}} = \int_{0}^{L} \left( \vec{F}_1 + \vec{F}_2 + \vec{F}_3 \right) \cdot \delta \vec{U} \cdot dl_0
\]  

(4.56)
The discretization of the distributed external forces \( \vec{F}_1, \vec{F}_2 \) and \( \vec{F}_3 \) respectively results in the resultant gravity and buoyancy elemental force: \( \vec{F}_{1i} \); the tangential drag elemental force: \( \vec{F}_{2i} \) and the normal drag elemental force: \( \vec{F}_{3i} \).

The expression for \( \vec{F}_{1i} \) can be easily obtained since \( \vec{F}_1 \) is constant and does not depend on the element local coordinate \( \xi \). Thus, the discretized gravity and buoyancy force for the element \( i \) is just:

\[
\vec{F}_1 \simeq \vec{F}_{1i} = \begin{bmatrix} 0 \\ -\gamma g \end{bmatrix} \tag{4.57}
\]

\( \vec{F}_{2i} \) is calculated from expression (4.44), using (4.41) and (4.50):

\[
\vec{F}_2 \simeq \vec{F}_{2i} = \frac{C_2}{L_i^2(1 + \epsilon_i)^2} \left| \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i \right| \left( \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i \right) B_{g_i} \dot{X}_i
\]

\[
\sum_{j=1}^{i-1} jL_j < l_0 \leq \sum_{j=1}^{i} jL_j \tag{4.58}
\]

And in the same way, \( \vec{F}_{3i} \) is obtained introducing (4.41) and (4.50) in the expression (4.10):

\[
\vec{F}_3 \simeq \vec{F}_{3i} = C_3 (1 + \epsilon_i) \left[ \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i - \left( \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i \right)^2 \right]^{\frac{1}{2}} \left( \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i \right)^2
\]

\[
\left[ \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i - \left( \dot{V}_i^T N_{g_i} B_{g_i} \dot{X}_i \right)^2 \right] \sum_{j=1}^{i-1} jL_j < l_0 \leq \sum_{j=1}^{i} jL_j \tag{4.59}
\]

Including into (4.56) the expressions (4.45), (4.41), (4.57), (4.58) and (4.59) we have:

\[
W_{\text{external}} \simeq \sum_{j=1}^{i-1} \int_0^{l_0} -\delta \vec{F}_i^T \dot{A}_i^T N_{g_i} \left( \vec{F}_{1i} + \vec{F}_{2i} + \vec{F}_{3i} \right) L_i d\xi \tag{4.60}
\]

### 4.3.8 Discretization of inertial forces

Finally, the term for the work produced by the inertial forces \( W_{\text{inertial}} \) in equation (4.13) is:

\[
W_{\text{inertial}} = \int_0^L \left( \gamma \ddot{R} + C_4 (1 + \epsilon) \left[ \ddot{R} - (\ddot{R} \cdot \vec{t}) \vec{t} \right] \right) \cdot \delta \vec{U} dl_0 \tag{4.61}
\]
If we substitute into (4.61) the expressions (4.38), (4.40), (4.46) and (4.50), we have:

\[ W_{\text{inertia}} \simeq \sum_{j=1}^{i-1} \int_0^1 \delta \vec{P}_T \vec{A}_i \left( \gamma L_i N_{g_i}^T A_i + C_4 \left[ L_i (1 + \varepsilon_i) N_{g_i}^T A_i - \frac{B_{g_i} \vec{X}_i \vec{X}_i^T B_{g_i} N_{g_i}^T A_i}{L_i (1 + \varepsilon_i)} \right] \right) \vec{X} d\xi \]

(4.62)

### 4.3.9 Discretized equations of motion

Once we have obtained the discretized expressions for the different terms of work within the equation (4.13) (expressions (4.51), (4.54), (4.60) and (4.62)), we can build the following equation for the total virtual work:

\[ W_V \simeq \int_0^1 \left( \gamma L_i A_i^T N_{g_i}^T N_{g_i}^T A_i \vec{X} + C_4 \left[ L_i (1 + \varepsilon_i) A_i^T N_{g_i}^T N_{g_i}^T A_i - \frac{N_{g_i}^T B_{g_i} \vec{X}_i \vec{X}_i^T B_{g_i} N_{g_i}^T A_i}{L_i (1 + \varepsilon_i)} \right] \right) \vec{X} d\xi = 0 \]

(4.63)

The virtual displacement \( \delta \vec{P} \) can be eliminated from the expression (4.63) since it is an arbitrary displacement. This allows us to finally find the equations of motion of the system in the following form:

\[ M \ddot{\vec{X}} + C \dot{\vec{X}} + K \vec{X} - \vec{F} = 0 \]

(4.64)

Where \( M \) is the mass matrix of the system:

\[ M = \sum_{i=1}^{n} A_i^T \int_0^1 \left( \gamma L_i N_{g_i}^T N_{g_i}^T + C_4 \left[ L_i (1 + \varepsilon_i) N_{g_i}^T N_{g_i}^T - \frac{N_{g_i}^T B_{g_i} \vec{X}_i \vec{X}_i^T B_{g_i} N_{g_i}^T A_i}{L_i (1 + \varepsilon_i)} \right] \right) d\xi_i A_i \]

(4.65)

\( C \) is the structural damping matrix of the system:

\[ C = \sum_{i=1}^{n} A_i^T \int_0^1 \beta E A \frac{B_{g_i} \vec{X}_i \vec{X}_i^T B_{g_i} N_{g_i}^T A_i}{L_i^3 (1 + \varepsilon_i)^2} d\xi_i A_i \]

(4.66)

The stiffness matrix of the system, \( K \) is:

\[ K = \sum_{i=1}^{n} A_i^T \int_0^1 E A \varepsilon_i \frac{B_{g_i}^T B_{g_i} N_{g_i}^T A_i}{L_i (1 + \varepsilon_i)} d\xi_i A_i \]

(4.67)
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4.3.10 Elemental matrices and force vectors

The equations (4.65), (4.66), (4.67) and (4.68) provide the expressions to build the mass matrix, the structural damping matrix, the stiffness matrix and the external force vector of the complete system in the global reference system. These global matrices and vectors are the result of the assembly of the elemental matrices and forces. This assembly procedure is expressed mathematically by the products with the adjacency matrix \( A_i \).

It is trivial to derive from these equations the expressions for the elemental mass matrix \( M_i \), the structural damping matrix \( C_i \), the structural stiffness matrix \( K_i \) and the force vector \( \vec{F}_i \) in the global reference system:

\[
M_i = \int_0^1 \left( \gamma L_i N' \bar{N}' + C_4 \left[ L_i (1 + \varepsilon_i) \bar{N}' - \frac{N' \bar{B} \bar{X}_i \bar{X}' \bar{B} \bar{T} \bar{N}'}{L_i (1 + \varepsilon_i)} \right] \right) d\xi_i \tag{4.69}
\]

\[
C_i = \int_0^1 \beta EA \frac{B' \bar{B} \bar{X}_i \bar{X}' \bar{B} \bar{T} \bar{B}}{L_i (1 + \varepsilon_i)^2} d\xi_i \tag{4.70}
\]

\[
K_i = \int_0^1 EA \varepsilon_i \frac{B' \bar{B} \bar{L}_i}{L_i (1 + \varepsilon_i)} d\xi_i \tag{4.71}
\]

\[
\vec{F}_i = \int_0^1 L_i N' \bar{F}_1 + \bar{F}_2 + \bar{F}_3 d\xi_i \tag{4.72}
\]

These elemental matrices and forces can also be expressed in the elemental reference system:

\[
m_i = \int_0^1 \left( \gamma L_i N' \bar{N}' + C_4 \left[ L_i (1 + \varepsilon_i) \bar{N}' - \frac{N' \bar{B} \bar{X}_i \bar{X}' \bar{B} \bar{T} \bar{N}'}{L_i (1 + \varepsilon_i)} \right] \right) d\xi_i \tag{4.73}
\]

\[
c_i = \int_0^1 \beta EA \frac{B' \bar{B} \bar{X}_i \bar{X}' \bar{B} \bar{T} \bar{B}}{L_i (1 + \varepsilon_i)^2} d\xi_i \tag{4.74}
\]

\[
k_i = \int_0^1 EA \varepsilon_i \frac{B' \bar{B}}{L_i (1 + \varepsilon_i)} d\xi_i \tag{4.75}
\]
\[ \mathbf{f}_i = \int_0^1 L_i N^T \left( \mathbf{f}_{1i}^T + \mathbf{f}_{2i}^T + \mathbf{f}_{3i}^T \right) d\xi_i \] (4.76)

Where:

\[ \mathbf{f}_{1i} = T_i^T \begin{bmatrix} 0 \\ -\gamma_r g \end{bmatrix} \] (4.77)

\[ \mathbf{f}_{2i} = \frac{C_2}{L_i^2(1+\varepsilon_i)^2} \left( \mathbf{v}_i^T N^T B \bar{x}_i \right) \left( \mathbf{v}_i^T N^T B \bar{x}_i \right)^T B \bar{x}_i \] (4.78)

\[ \mathbf{f}_{3i} = C_3(1+\varepsilon_i) \left[ \mathbf{v}_i^T N^T N \mathbf{v}_i - \frac{(\mathbf{v}_i^T N^T B \bar{x}_i)^2}{(1+\varepsilon_i)^2 L_i^2} \right] \left[ N \mathbf{v}_i - \frac{(\mathbf{v}_i^T N^T B \bar{x}_i) B \bar{x}_i}{(1+\varepsilon_i)^2 L_i^2} \right] \] (4.79)

4.4 Code implementation of the mooring lines dynamics

A computer program in Fortran 90 based on the theoretical development described in section 4.3, has been developed with the capability of simulating the dynamics of a mooring line submerged in water, under the action of waves and in contact with the seabed. The code has been named OPASS (“Offshore Platform Anchoring System Simulator”).

4.4.1 Implementation of the Finite Element Method

The equation of motion of the system, (4.64), are built according to the classical steps of the Finite Element Method:

1. Discretize the cable in a finite number of bar elements.
2. Built the elemental mass, damping and stiffness matrices using equations (4.73), (4.74), (4.75) and the external force vectors according to (4.76).
3. Transform the elemental matrices and force vectors into the global coordinate system using the local to global transformation matrix (4.32).
4. Assembly the global mass, damping and stiffness matrices and the global external forces vector using the adjacency matrices (4.42).

The resultant system of equations of motion are ordinary differential equations. Alternatively, the steps 2 and 3 can be performed in only one step using the equations (4.69),
These are the expressions that have been implemented in the code, providing a better computational efficiency.

As has been explained in subsection 4.3.2, the interpolation function chosen for the inertial forces are step functions instead of linear functions. This results in a mass matrix composed by 3 x 3 submatrices around the diagonal. Thus, the assembly of the matrices and force vector (step 4) can be simplified and the inversion of the mass matrix is performed with much lower computational cost.

The efficiency in the computation of the matrices and also in the inversion of the mass matrix is important to obtain a fast simulation code, because matrices are calculated at every time step.

### 4.4.2 Initial configuration

The static solution of the catenary shape formed by the mooring line that is hanging between the fairlead and the anchor is used for the initial reference cable configuration $R$. This static shape is calculated using the analytical formulation for a cable suspended between two points that was implemented in FAST, to be used in the quasi-static mooring line model, as it is described in [14]. This formulation considers the elastic stiffness of the mooring line, the cable weight and buoyancy, the contact with a horizontal seabed and static tangential seabed friction and the resulting system of equations is solved using a Newton-Raphson iteration scheme. As the ”Catenary” subroutine included in the FAST code already solves these equations, this subroutine has been adapted and used in OPASS for the computation of the mooring line initial reference configuration $R$ before starting the time integration of the equations of motion.

### 4.4.3 Seabed contact model

A contact model of the line with the seabed has been implemented using bi-linear springs. When a node is in contact with the seabed, a spring with stiffness $K_{sc}$ provides the floor reaction force per indentation depth and per unit of line length. A damping $D_{sc}$ is also included in the model. Indentation of the line into the seabed due to self-weight is:

$$\delta S_0 = \frac{\gamma_c g}{K_{sc}}$$

(4.80)

If $d_w$ is the water depth and $Z_i$ is the vertical position of the node $i$ in the global reference system, the seabed only introduces a force on the node $i$ when the condition:

$$Z_i < (d_w - \delta S_0)$$

(4.81)
is fulfilled. This force provided by the spring at the node \( i \), denoted as \( \vec{F}_{sc}^i \), is calculated as:

\[
\vec{F}_{sc}^i = \begin{bmatrix}
0 \\
0 \\
-K_{sc}(Z_i - d_w - \delta S_0) - D_{sc}\dot{Z}_i
\end{bmatrix}
\]  \hspace{1cm} (4.82)

Where \( \dot{Z}_i \) is the vertical velocity of the node \( i \).
Thus, the elements resting at the seabed will be located at \( D_w \). If the elastic forces compensate only part of the weight of the element, the node will be located between \( D_w \) and \( D_w - \delta S_0 \).

![Cable-seabed contact model](image)

**Figure 4.7**: Cable-seabed contact model

### 4.4.4 Time integration of the equations

The system of equations that we have obtained through the Finite Element method, (4.64), can be rewritten as:

\[
\ddot{\vec{X}} = M^{-1} \left( \vec{F} - C \dot{\vec{X}} - K \vec{X} \right)
\]  \hspace{1cm} (4.83)

To integrate this equation in time, three different integration schemes were implemented. The first one is a simple explicit scheme described in [54] and [55], based in the central difference formula. The second integrator is the Runge-Kutta-Nyström scheme described in [38]. This method requires four evaluations of the equations of the lines (4.83) per time step, increasing the computational effort, but, on the other hand, it allows to increase the size of the time step. Finally, the third integrator is the Adams-Moulton-Bashforth predictor-corrector scheme [28], where the solution at each time step is achieved by evaluating twice the equations of motion (4.83). In the first evaluation, called predictor stage, the accelerations are obtained
from the equations of motion of the system and they are used to calculate the positions and velocities of the line’s degrees of freedom at the next time step. In a second stage, called corrector, this solution is refined using the next time step positions and velocities to obtain more accurate accelerations for the next step. A final estimation of positions and velocities from these accelerations is performed and one time step is advanced.

4.5 Verification of the OPASS code

The OPASS code has been successfully verified against computations with the 3DFloat code [19], [20]. In addition, it has been coupled with the FAST code [21], whose original FAST mooring lines model is a quasi-static approach, expanding its capabilities for the integrated simulation of floating offshore wind turbines. This coupled version will be used in the research of Chapter 6. A verification of OPASS coupled with FAST was satisfactorily carried out within the IEA Annex 30 benchmark (OC4) [22], [23].

4.6 Concluding remarks

The equations of motion of a submerged line are set and solved using a FEM technique. A Lumped Mass formulation oriented to achieve high computational efficiency has been adopted. The global mass matrix is composed by 3 x 3 submatrices in the diagonal resulting in easily invertible matrix with low CPU cost. The code has been verified against computations with other existing tools for mooring lines and also, coupled with FAST, by taking part in the IEA Annex 30 (OC4) international simulation codes benchmark.
Chapter 5

Experimental validation of the code

5.1 Abstract

In this chapter, the OPASS numerical simulation code developed in Chapter 4 is experimentally validated against a set of tests performed at the École Centrale de Nantes (ECN) wave tank in France. The tests consist of a suspended chain submerged into a water basin, where the suspension point of the chain is excited with harmonic motions of different periods. Equivalent simulations of the chain setup are launched to compare against the experimental results. A description of the tests, the results and the conclusions presented in this chapter has been published in [9] and [7]. The code is able to predict the tension at the suspension point and the motions at several positions of the line with high accuracy. Even for those cases where the line loses and subsequently recovers tension, the resulting snap load and motions are well captured with a slight overprediction of the maximum tension. The drag coefficients for chains used in the computations have been taken from the DNV guidelines and, in general, predict correctly the hydrodynamic loads. In addition, sensitivity studies and verification against another code show that highly dynamic cases are sensitive to the seabed-cable contact and friction models. The results show the importance of capturing the evolution of the mooring dynamics for the prediction of the line tension, especially for the high frequency motions.

5.2 Description of the experiments

The dimensions of the École Centrale de Nantes wave tank are 50m length, 30m width and 5m depth. The chain was submerged into the water basin, forming a catenary shape with the bottom end anchored to the tank floor. The suspension point was connected through a load cell to a mechanical actuator with the capability of reproducing a prescribed motion that was located at the water free surface. Figure 5.1 shows the configuration of the experiment. During the test, the anchor remains fixed at the bottom of the basin, at a depth of
CHAPTER 5. EXPERIMENTAL VALIDATION OF THE CODE

5m. The fairlead, located at the water plane, is excited by the mechanical actuator with a
sinusoidal prescribed horizontal motion in the plane of the catenary, around a mean position.
The distance $d$ is the horizontal distance between the anchor and the mean position of the
fairlead during the tests. Two different configurations of the mooring chain corresponding
to different values of the parameter $d$ are tested: Configuration 1, where $d = 19.364$ m, and
Configuration 2, where $d = 19.870$ m. Figure 5.1 also shows the reference system that will
be used to present the results.

The steel chain selected for the tests is a DIN5685A design with a link diameter of 2 mm.

The length of the chain including the load cell at the fairlead is 21m. Some preliminary
simulations have been performed in order to confirm that the presence of the load cell has
not an important impact on the tension and dynamics of the chain. The mass per unit
length was measured and a value of 69g/m was found, though the original manufacturer
CHAPTER 5. EXPERIMENTAL VALIDATION OF THE CODE

specifications was 70g/m. This chain model was chosen because the resulting full scale mass scales by an approximate factor of $1/40^2$ when compared to the properties of the OC4 floating model mooring lines [75]. With a scale factor of 1/40, the 5m depth of the basin represents a 200m depth sea location. Therefore, this chain represents a typical scaled mooring line used in wave tank tests for floating wind turbines.

First, two static cases with the chain suspension point fixed corresponding to Configuration 1 and Configuration 2 are studied. In addition, 6 dynamic cases are simulated combining the 2 chain configurations with three different oscillation periods of the motion imposed at the fairlead. Similar criteria based on scale factors used for the chain dimensions were also used in the selection of the excitation periods of 1.58s, 3.16s and 4.74s. These periods correspond to oscillation periods of 10s, 20s and 30s in full scale when a 1/40 scale factor is used. As a matter of fact, a typical surge period of a moored platform is 10s or higher. The cases that have been considered are described in Table 5.1.

<table>
<thead>
<tr>
<th>Case ID</th>
<th>Configuration</th>
<th>Anchor - Fairlead Mean Distance d (m)</th>
<th>Amplitude (m)</th>
<th>Period (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>19.364</td>
<td>Static</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>19.872</td>
<td>Static</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>19.364</td>
<td>0.25</td>
<td>1.58</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>19.364</td>
<td>0.25</td>
<td>3.16</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>19.364</td>
<td>0.25</td>
<td>4.74</td>
</tr>
<tr>
<td>6</td>
<td>2</td>
<td>19.872</td>
<td>0.25</td>
<td>1.58</td>
</tr>
<tr>
<td>7</td>
<td>2</td>
<td>19.872</td>
<td>0.25</td>
<td>3.16</td>
</tr>
<tr>
<td>8</td>
<td>2</td>
<td>19.872</td>
<td>0.25</td>
<td>4.74</td>
</tr>
</tbody>
</table>

During the dynamic tests, the position of 8 reflecting markers located at different chain positions and the tension at the fairlead of the chain were measured. The distance between two adjacent reflecting markers along the chain is approximately 0.5m. The exact position of these markers is provided in Table 5.2. For the static cases (cases 1 and 2), in addition to the markers in Table 5.2, the positions of 4 additional reflecting markers located at the lower part of the chain were also monitored. The distance between these additional reflecting markers was also 0.5m approximately. The fairlead static tension was also measured.

The motion tracking system that captures the motion of the reflecting markers at the different chain positions is composed by 6 Qualisys underwater cameras, type Oqus 3+, with 1296x1024 pixels. The lenses have 20mm and 24mm of focal length with an averaged residual of 2mm. This results in an uncertainty of the marker position of approximately 2mm. The
Table 5.2: Position of the markers

<table>
<thead>
<tr>
<th>Marker Number</th>
<th>Position Along the Line from the Fairlead (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.656</td>
</tr>
<tr>
<td>2</td>
<td>1.155</td>
</tr>
<tr>
<td>3</td>
<td>1.655</td>
</tr>
<tr>
<td>4</td>
<td>2.149</td>
</tr>
<tr>
<td>5</td>
<td>2.646</td>
</tr>
<tr>
<td>6</td>
<td>3.152</td>
</tr>
<tr>
<td>7</td>
<td>3.655</td>
</tr>
<tr>
<td>8</td>
<td>4.164</td>
</tr>
</tbody>
</table>

Load cell used to measure the tension at the fairlead is a submersible DDEN model with a range of 0-500N and an accuracy of ±0.25%.

5.3 Parameters of the computational model

As has been mentioned in Section 5.2, the measured weight per unit length of the chain is 69 g/m and the length is 21m. As the computational model assumes that the chain is a line with a constant circular section, an equivalent hydrodynamic diameter has to be determined. To do this, the volume of the total length of the chain was measured and the diameter of the circular section that provides that volume for the same length was calculated assuming that the material density (steel) had a value of 7850 kg/m³.

The hydrodynamic coefficients, in particular the drag coefficients, have a great influence on the mooring line dynamics, therefore a realistic selection of the values is critical to obtain accurate simulation results. An added mass coefficient, $C_{mn}$, of 1 was chosen according to Bureau Veritas [16]. The normal drag coefficient, $C_{dn}$, and the tangential drag coefficient, $C_{dt}$, were obtained following the indications of DNV [23]. For a studless chain, this guideline provides a value for $C_{dn}$ of 2.4 and for $C_{dt}$ of 1.15. These values are referred to the wire diameter of the chain link. For the implementation in our code, the value has to be referred to the equivalent hydrodynamic diameter, resulting in values of 1.4 and 0.67, respectively.

The axial stiffness of the chain has been estimated based on the link diameter according to Equation (5.1), see [69]:

$$EA = 0.854 \cdot 10^8 d_w^2 \, (kN),$$  \hspace{1cm} (5.1)

Where $d_w$ is the wire diameter of the chain link in meters.

The characteristics of the chain are summarized in Table 5.3:

A line-seabed contact model has been defined, including the friction of the chain with the seabed. The parameters related to this model are collected in Table 5.4.
Table 5.3: Characteristics of the chain

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>21 m</td>
</tr>
<tr>
<td>Mass per unit length</td>
<td>0.069 kg/m</td>
</tr>
<tr>
<td>Density</td>
<td>7850 kg/m³</td>
</tr>
<tr>
<td>Axial stiffness</td>
<td>3.4E5 N</td>
</tr>
<tr>
<td>Structural damping</td>
<td>0.1 %</td>
</tr>
<tr>
<td>Equivalent hydrodynamic diameter</td>
<td>0.0034 m</td>
</tr>
<tr>
<td>Added mass coefficient</td>
<td>1.0 -</td>
</tr>
<tr>
<td>Normal drag coefficient</td>
<td>1.4 -</td>
</tr>
<tr>
<td>Tangential drag coefficient</td>
<td>0.67 -</td>
</tr>
</tbody>
</table>

Table 5.4: Characteristics of the line-seabed contact and friction models. The stiffness and damping values are per unit length of the line.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical seabed stiffness</td>
<td>20 N/m²</td>
</tr>
<tr>
<td>Vertical seabed damping</td>
<td>0.1 Ns/m²</td>
</tr>
<tr>
<td>Tangential friction coefficient</td>
<td>0.5 -</td>
</tr>
<tr>
<td>Normal friction coefficient</td>
<td>0.5 -</td>
</tr>
</tbody>
</table>

A minimum of 30 elements were used to discretize the chain in the computations. A sensitivity analysis was performed running several simulations with double number of elements with no significant changes in the results.

5.4 Comparison of computational and experimental results

In this section, a comparison between the experimental results and computations is presented, with the objective of validating the code for the dynamic simulation of mooring lines. First, computed and experimental data of the shape and tension are presented for the static cases. Afterwards, results comparing measurements and computations of the tension at the suspension point and the positions of different chain points are shown for the dynamic cases. Results in this section are presented according to the reference system in Figure 5.1.
5.4.1 Static cases

These cases were used to adjust the exact horizontal position of the chain anchor that was fixed to the basin bottom by a diver. In contrast to other parameters of the test that can be easily measured or derived, as the chain length, the weight per unit length, or the equivalent hydrodynamic diameter; the accurate determination of the anchor position is a difficult step due to the difficulties of measuring and placing it under the water at the exact specified distance of several meters. For this reason, looking at the static shape measured by the underwater cameras, we concluded that the exact anchor position measured from the basin wall was 5.305m (See Figure 5.1). The results, in particular the dynamic tension, are sensitive to small variations of the anchor position. Assuming this anchor position, the agreement of the computational and the experimental static shape of the chain is excellent in both configurations, as it is shown in Figure 5.2.

![Figure 5.2: Comparison of the computed and experimental static shapes for both line configurations](image)

Table 5.5 shows the difference between the computed tension at the fairlead and the experimental measurement for both configurations. The error in the predicted static tension is below 2%.

5.4.2 Dynamic cases: tension at the fairlead

Figures 5.3 and 5.4 show the comparison between the numerical predictions and the experiments for the different periods of the motion prescribed at the fairlead of the chain for
Table 5.5: Comparison of static fairlead tension expressed in Newtons.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Configuration 1</th>
<th>Configuration 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experimental</td>
<td>8.13</td>
<td>14.48</td>
</tr>
<tr>
<td>Computation</td>
<td>8.10</td>
<td>14.70</td>
</tr>
<tr>
<td>Difference</td>
<td>0.37%</td>
<td>1.52%</td>
</tr>
</tbody>
</table>

Configuration 1 and Configuration 2, respectively. The plots show the fairlead tension of the chain against the fairlead position. The initial transients have been eliminated and the data is plotted once the steady state has been reached. The arrows represent the sense of the dynamic loop: higher tensions are achieved when the fairlead of the chain is moving away from the anchor and the lower tensions appear when it is approaching the anchor. For the experimental data, the loops have been generated from time series containing several tens of periods of the prescribed motion. As the measured signal presents a certain amount of noise, instead of plotting the direct out-coming data, the mean value and the standard deviation of the data have been computed. These statistical data is represented in the graphs instead of the raw data for clarity. The mean value is represented by a circle and the standard deviation by a vertical bar centered at the mean and limited by horizontal lines. The definition of the standard deviation, \( \sigma \), is given in Equation (5.2):

\[
\sigma = \sqrt{\frac{\sum_{i=1}^{n} (X_i - \bar{X})^2}{n}},
\]

Where \( n \) is the number of elements in the data, \( \bar{X} \) is the mean value of the data and \( X_i \) is the \( i^{th} \) element of the data.

In addition, the plots show the fairlead tension of the chain for the initial static position (red circle) and also the tension provided by the quasi-static model for the different positions of the fairlead (gray line).
Figure 5.3: Fairlead tension for the different periods used in the dynamic simulations for Configuration 1. The tension at the fairlead for the initial static position (red circle) and also the tension provided by the quasi-static model for the different positions of the fairlead (gray line) are also represented.
As can be observed in Figures 5.3 and 5.4, the tension in the Configuration 1 is lower than in the Configuration 2. In fact, the initial static tension for Configuration 1 is 8N and for Configuration 2 it is 15N, and the maximum tension, for the 1.58s period, in Configuration 2 is three times the value in Configuration 1. As has been mentioned, the tension signal acquired during the tests, presents a certain amount of noise. This is the reason why the relative importance of the tension standard deviation in Configuration 1 is higher than in the second one. The figures also illustrate the importance of dynamic effects, in particular as the excitation frequency increases. The maximum tension for the dynamic computation of the case with 1.58s of excitation period can be between 2 (Configuration 1) and 3 (Configuration 2) times the value predicted by the quasi-static approach. For fatigue calculations of the fairlead and line, the dynamic effects are even more significant. For Configuration 2 with 1.58s excitation period, the dynamic tension amplitude is more than ten times times the corresponding quasi-static value.

The agreement between computations and measured data is, in general, very good for both configurations and all the excitation periods. The two highest excitation periods, 3.16s and 4.74s, correspond to the chain’s Harmonic Condition described in [84]. In these cases the
agreement is good, in particular for Configuration 2, where the relative importance of the noise, and the standard deviation of the data, is lower.

For the lowest period of excitation, 1.58s, the dynamics of the chain correspond to the Snap Condition which is more complex. In Configuration 1 the chain partially loses tension, and in Configuration 2 the situation is more extreme and the chain totally loses tension; the measured tension at the fairlead is 0, during almost half of the period. Nevertheless, the agreement with computations is still good and a discrepancy of around 4.5% on the estimation of the maximum tension appears in both configurations. This maximum tension appears approximately when the velocity of the chain’s fairlead is maximum. As described in Section 5.3, we have chosen the tangential and normal drag coefficients for the computations based on the values provided by the guidelines. These values provide an accurate matching with experimental results, though further research on the characterization of these parameters for the regime with highest excitation frequency may result on a better prediction of the maximum tension for the 1.58s period case.

Another slight discrepancy appears in both configurations for the period of 1.58s. When the fairlead moves towards the anchor, the chain loses tension. The fairlead motion reverses, the tension is recovered, and the links that are falling freely suddenly are pulled upwards, producing a snap load; a sudden increase of the tension. During this event, the measured tension temporarily reaches a plateau, before again increasing towards the peak value. For the experimental data this decrease in the slope appears later and it is more pronounced than in the computational results. This effect is even more important in the Configuration 2 case.

The case for Configuration 2 and oscillation period of 1.58s, where the loss of tension of the line is the most pronounced, is very sensitive to the modelling of the seabed-cable interaction. For this reason, this case was studied more in detail and was simulated with the alternative code 3DFloat, developed by IFE. 3DFloat is based on a nonlinear co-rotated FEM approach, with three translational and three rotational DOFs at each node. The cable elements are Euler-Bernoulli beams, where the bending stiffness is turned off for chain mooring lines. Both codes have elements that maintain axial stiffness also when tension is lost, in contrast to the real chain. This fact can affect the behaviour for the part of the chain resting on the seafloor. Both models take the seafloor into account with a seabed contact model. Nodes that drop below the defined seafloor, get an applied vertical force that is proportional to the vertical distance to the seafloor. In addition, linear damping is applied to nodes below the seafloor in the vertical direction. Both codes scale the linear damping with the distance up to the seafloor. 3DFloat also applies a limiter to the damping. 3DFloat uses no friction for the computations in this paper, but applies a horizontal damping to the nodes in contact with the seabed, that is not present in OPASS.

Figure 5.5 shows the effect of eliminating the seabed-cable friction on the tension computed with OPASS for the mentioned sensitive case of period 1.58s and Configuration 2. After a few periods of excitation, an oscillation is generated in the part of the loop where the suspension point moves from the minimum position towards the maximum position and the
chain is being lifted quickly off the seafloor. This effect does not arise in any of the other cases studied in this work when the seabed friction model is disabled. When the considered case is simulated with 3DFloat, a similar effect arise (see Figure 5.6); if no damping is applied to the nodes in contact with the seabed, an oscillation appears when the line is recovering the tension. But if enough damping is applied in the vertical and horizontal direction to the nodes in contact with the seabed, this dynamics disappears. Apparently, these dynamics are caused by small motions of the nodes of the line resting on the seabed during the tension-less portion of the dynamic loop. The introduction of seabed friction in the OPASS model or seabed damping in the 3DFloat model avoid these small nodal displacements and, consequently, the oscillations are not generated. These dynamics are not present in the experimental results, due to the friction of the chain with the bottom of the basin. This case highlights the importance of a correct modelling of the line-seabed interaction to avoid the generation of instabilities in highly dynamic situations with pronounced loss of tension in the chain.

![Figure 5.5: Effect of seabed friction on the fairlead tension computed with OPASS for 1.58s period and Configuration 2](image-url)

Figure 5.5: Effect of seabed friction on the fairlead tension computed with OPASS for 1.58s period and Configuration 2
Figure 5.6: Effect of seabed damping on the fairlead tension computed with 3DFloat for 1.58s period and Configuration 2

Figure 5.6 shows how the two models agree well for the 1.58s period case, considering the modelling differences, once the friction is included in OPASS and the seabed damping is included in 3DFloat. The results are not sensitive to further increases of friction or damping levels beyond damping the oscillations seen in the Figure 5.3 and Figure 5.4. For the larger periods the results are not sensitive to the seabed model and both tools provide almost identical tension.
In summary, the computational model is able to predict with good precision the tension in all the cases considered. In those cases where chain loses tension and the dynamics is more complex, the computations also predict the tension accurately, though special attention has to be paid to the seabed-cable interaction model.

5.4.3 Dynamic cases: motion of the chain

A comparison between the measured and the computed trajectories of the 8 positions along the chain length specified in Table 5.2 is discussed in this section. Figures 5.8 to 5.23 show the trajectories of the markers for both configurations and for the different periods of excitation of the chain fairlead. According to the reference system described in Figure 5.1, the $x$ and $z$ coordinates represent the horizontal and vertical positions of the marker respectively. Similarly to the procedure conducted in Section 5.4.2, the loops of experimental data have been generated based on several tens of periods of the prescribed motion and graphically represented by the mean value and the standard deviation. The dispersion of data for the motion is lower than in the tension results. For the calculation of the mean value and standard deviation, data has been stored into bins. The classification of data in these bins has been performed in most of cases based on the $x$ axis, but in some particular cases, as for
example the motion of markers 5 and 6 for Configuration 2 and period 1.58s (see Figure 5.17 and Figure 5.19), the discretization is based in the z axis, due to the shape of the curve and the requirements of the algorithm used in the data processing. In those cases, the dispersion bars on the plots are shown with their corresponding direction.

As before, the red dot in the plots show the initial static position of the marker before the chain is excited and the gray line represents the displacement of the marker according to the quasi-static model. The arrows indicate the sense of the dynamic loops.

The scale and length of both the x and z axis is the same in all the plots from Figure 5.8 to Figure 5.23 to keep a constant reference frame for the chain at each marker position. The loop corresponding to period 1.58s represented in Figure 5.24 is not complete due to the loss of visibility of the marker by the tracking system during part of the cycle.
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Figure 5.8: Trajectory described by the marker 1 (Conf. 1)

Figure 5.9: Trajectory described by the marker 1 (Conf. 2)

Figure 5.10: Trajectory described by the marker 2 (Conf. 1)

Figure 5.11: Trajectory described by the marker 2 (Conf. 2)

Figure 5.12: Trajectory described by the marker 3 (Conf. 1)

Figure 5.13: Trajectory described by the marker 3 (Conf. 2)
CHAPTER 5. EXPERIMENTAL VALIDATION OF THE CODE

Figure 5.14: Trajectory described by the marker 4 (Conf. 1)

Figure 5.15: Trajectory described by the marker 4 (Conf. 2)

Figure 5.16: Trajectory described by the marker 5 (Conf. 1)

Figure 5.17: Trajectory described by the marker 5 (Conf. 2)

Figure 5.18: Trajectory described by the marker 6 (Conf. 1)

Figure 5.19: Trajectory described by the marker 6 (Conf. 2)
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Figure 5.20: Trajectory described by the marker 7 (Conf. 1)

Figure 5.21: Trajectory described by the marker 7 (Conf. 2)

Figure 5.22: Trajectory described by the marker 8 (Conf. 1)

Figure 5.23: Trajectory described by the marker 8 (Conf. 2)
In general, the agreement between the measured motions and the simulations is very good for both configurations and all oscillation periods and positions. The highest disagreement appears for the Configuration 2 at the highest frequency of oscillation (period 1.58s), where the line totally loses tension. Nevertheless, the agreement is still good.

The motion of the chain markers is predominantly horizontal in the positions closer to the fairlead of the chain, where the horizontal motion was prescribed, but it transforms to a predominantly vertical motion as the distance to the fairlead increases. The reason is that the tension of the chain decreases as we get further from the fairlead. For the markers located deeper, when a low motion period (1.58s) is imposed, the chain links feel zero load during the portion of the loop where the tension in the chain is lower and consequently fall freely. This corresponds to the right part of the loops, that is almost vertical, especially for the lower excitation periods and markers located far from the fairlead (see for example Figures 5.14 to 5.19). When the tension at the marker position recovers, the predominately vertical motion changes to a negative $x$-direction and positive $z$-direction, corresponding approximately to the left part of the loops. The effect of tension loss is even more clear on the short period cases (period 1.58s), because the decrease in tension along the chain is more pronounced. For the cases with longer periods, the tension remains positive in a longer portion of the chain length, or even all along. In those cases (periods 3.16s and 4.74s), the motion of the marker is close to the quasi-static solution.

Figure 5.24 represents the $x$ displacement for the marker 5 in Configuration 2 (motion period of 1.58s) and the tension at the fairlead of the chain as function of the time. Figure 5.25 represents the $z$ displacement for the same case and marker, also shown with the tension. These Figures illustrate how, as the tension approaches to 0, the marker stops moving in the $x$ direction and freely falls in the negative $z$ direction. During the period when the line remains slack, the marker displacement in $x$ suffers a small oscillation giving a N-shape: the marker is initially moving in the positive $x$ direction, it goes backwards for a while and then continues forwards again. This N-shape effect is responsible for the secondary loops that appear at the deeper markers in the $x-z$ trajectories of Configuration 2 for period 1.58s, see Figures 5.21 or 5.23. The N-shape is more pronounced as the distance along the chain to the fairlead increases. For this reason, no secondary loops are present at the markers closer to the fairlead, but they arise and increase their amplitude with the increasing distance to the excitation source. The dynamics of this effect on the loops is well predicted by the code, though it seems to be a complex effect and very sensitive to small variations in parameters. Thus, the trajectories of the markers located lower with the highest oscillation frequency (period 1.58s) and for Configuration 2 are the ones where the inaccuracy of the computations is higher, in particular for the $x$ coordinate.
Figure 5.24: X position and tension for marker 5 (Conf. 2, period 1.58s)
5.5 Concluding remarks

The dynamic mooring line code developed in this research has been successfully validated against experimental data for static and dynamic conditions. Two configurations of the chain have been studied, with different tension levels. The agreement between computations and experimental results is very good for the chain fairlead tension and also for the motion of the chain at the diverse positions considered.

In the static cases, both the shape of the line and the tension at the suspension point compare very well between the computations and the experiments.

For the dynamic validation, the chain has been excited with prescribed motion at the fairlead with three oscillation periods, producing different dynamic conditions, including harmonic response, loss of tension and snap loading. The code is able to predict the motion of the chain and the tension with precision in all these conditions.

The added mass and drag coefficients for the chain that have been chosen from guidelines, in general, represent with accuracy the hydrodynamic loads.

The importance of including dynamic effects on the prediction of the mooring line loads has been shown; the maximum tension can be between 2 and 3 times the value computed by the quasi-static approach when high excitation frequency is imposed. The tension amplitude can be more than ten times times the corresponding quasi-static value.
For the cases with the lowest excitation period, where the line slacks, the computed and measured tension agree well, although a slight difference appear during the snap loads once tension recovers. In these cases, the maximum tension is overpredicted by the code by around 4.5%. These differences could be due to the fact that the drag coefficients were not tuned for the cases with the highest excitation frequency. In the chain motion prediction study, those cases where the chain loses tension present the highest inaccuracies in the numerical results, especially in the lowest part of the chain, though the results are still good.

In the cases with higher tension level and higher excitation frequency, the simulation results are particularly sensitive to the cable-seabed interaction model. This was confirmed also by verification against the 3DFloat code of IFE. The computational model is able to predict the tension with accuracy, but the modelling of the seabed friction or the seabed damping should not be neglected.
Chapter 6

Mooring Dynamics on fatigue and ultimate loads

6.1 Abstract

In this chapter, the OPASS code, developed and experimentally validated in previous chapters, is used for an extensive assessment of the effect of mooring dynamics on the fatigue and ultimate loads of different floating wind turbines. The loads were computed following the floating wind turbines IEC 61400-3 Edition 1 guideline [43].

Three platforms topologies (semisubmersible, spar-buoy and TLP), representing the three main methods to achieve stability, are assumed to be installed at the same 200 m depth location in the Irish coast supporting the same 5 MW wind turbine. For each platform, the fatigue and ultimate loads are computed with the FAST integrated floating wind turbine simulation code using both, a quasi-static and the OPASS fully dynamic moorings model. More than 20,000 load cases are launched and postprocessed following the IEC 61400-3 guideline and fulfilling the conditions that a certification entity would require to an offshore wind turbine designer. The methodology and results presented in this chapter, have been published in [12] and [7].

As Chapter 3 discussed, several authors have compared coupled simulations using a quasi-static and a dynamic model for different floating platforms, having most of these studies important limitations. The following aspects of the research described in this chapter have to be highlighted, because they contribute to overcome the weaknesses of the previous studies:

1. Fatigue equivalent loads are computed based on all the load cases requested by the guideline, not only in individual cases, and are weighted based on their importance on the turbine lifetime.

2. Extreme loads are also calculated based in all the load case groups requested by the guideline and not only in a limited number of cases.
3. The environmental data have been computed for a real location.

4. The buoyancy stabilized platform concept is represented by a semisubmersible, that is closer to a industrial design than other less optimized designs as the barge.

5. The TLP design has a limited draft of 19.72 m and is not ballasted, being a very representative example of a tension stabilized concept.

6. All the platforms are assumed to be in the same location and water depth of 200 m to compare them in similar conditions.

The objective of this chapter is to identify in which conditions and for which platform concepts the mooring dynamic effects have influence on the simulation results. The effect on the level of fatigue and ultimate loads of the different wind turbine components is quantified providing future designers information for the selection of the adequate mooring lines model. The results showed that the impact of mooring dynamics in both fatigue and ultimate loads increases as elements located closer to the platform are evaluated; the blade and the shaft loads are only slightly modified by the mooring dynamics in all the platform designs, the tower base loads can be significantly affected depending on the platform concept, and the mooring lines tension strongly depends on the lines dynamics both in fatigue and extreme loads in all the platform concepts evaluated.

6.2 Floating wind turbines models

The three platform designs selected for this study, shown in Figure 6.1, are the UMaine Hywind spar [44], the OC4 DeepCwind semisubmersible [75] and the UMaine TLP [83]. The description of the three platforms is public and all of them support the NREL offshore 5MW baseline wind turbine [46].

The UMaine Hywind spar floating platform is an adaptation for 200 m sea depth by the University of Maine of the OC3-Hywind concept for 320 m depth used in the IEA Annex 23 project (also known as OC3 project) [44]. The concept used in the OC3 project was based on the publicly available information of the Hywind project. The draft of the UMaine Hywind spar platform is 120 m below sea water level (SWL). The substructure consists of two cylindrical regions connected by a tapered conical region. The diameter above the taper (6.5 m) is more slender than below (9.4 m) to reduce the hydrodynamic loads near the surface. The lower part of the platform is ballasted to bring down the center of gravity increasing the metacentric height and the stability.

The semisubmersible concept was created within the DeepCwind project [35] and used as a benchmark model in the IEA Annex 30 project [75] (also known as OC4 project). It consists of a main column and three offset columns that are connected to the main column with pontoons and cross members. The buoyancy of the offset columns provides stability to the platform. The draft of the platform is 20 m and the distance between the external columns
Figure 6.1: Sketch of the three platform concepts: UMaine Hywind spar (left), OC4 DeepCwind semisubmersible (center) and UMaine TLP (right)

is 50 m. Three heave plates at the bottom of the columns damp the vertical motion. The UMaine TLP was created by the University of Maine inspired by the Glosten Associates design. The draft of the platform is 30 m. The platform is light and the excess of buoyancy is compensated by the mooring lines that are in tension and attached at the end of three 30 m length spokes that provide the restoring moment.

Table 6.1 collects the main parameters of the three platforms studied in this research.

<table>
<thead>
<tr>
<th>Platform draft (m)</th>
<th>Spar</th>
<th>Semisubmersible</th>
<th>TLP</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>120</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>Platform mass, including ballast (t)</td>
<td>7,466</td>
<td>13,473</td>
<td>775</td>
</tr>
<tr>
<td>CM location below SWL along platform centreline (m)</td>
<td>89.91</td>
<td>13.46</td>
<td>19.72</td>
</tr>
<tr>
<td>Platform roll inertia about CM (kgm$^2$)</td>
<td>$4.23 \times 10^9$</td>
<td>$6.827 \times 10^9$</td>
<td>$1.5078 \times 10^8$</td>
</tr>
<tr>
<td>Platform pitch inertia about CM (kgm$^2$)</td>
<td>$4.23 \times 10^9$</td>
<td>$6.827 \times 10^9$</td>
<td>$1.5078 \times 10^8$</td>
</tr>
<tr>
<td>Platform yaw inertia about centreline (kgm$^2$)</td>
<td>$1.64 \times 10^8$</td>
<td>$1.226 \times 10^9$</td>
<td>$9.885 \times 10^7$</td>
</tr>
<tr>
<td>Displaced water volume (m$^3$)</td>
<td>8,035</td>
<td>13,917</td>
<td>2,767</td>
</tr>
</tbody>
</table>

In the spar platform, a viscous drag coefficient of 0.6 is applied to the submerged portion of the platform cylinder introducing an additional damping in the direction perpendicular to
the cylinder axis. Following the model description in [ref], additional linear damping coefficient in the surge, sway, heave and yaw degrees of freedom was applied to capture accurately the platform dynamics. This additional damping is provided in Table 6.2.

Table 6.2: Additional linear damping for the spar platform

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Surge</td>
<td>100 kNs/m</td>
</tr>
<tr>
<td>Sway</td>
<td>100 kNs/m</td>
</tr>
<tr>
<td>Heave</td>
<td>130 kNs/m</td>
</tr>
<tr>
<td>Yaw</td>
<td>13,000 kNms/rad</td>
</tr>
</tbody>
</table>

For the semisubmersible platform, a viscous drag coefficient was also applied in the cylindrical elements. The value of the coefficient depends on the element diameter and is shown in Table 6.3. For the heave plates a coefficient of 4.8 was defined, using the diameter of the plates as reference length.

Table 6.3: Viscous drag damping coefficients in the semisubmersible platform elements

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Main column</td>
<td>0.56</td>
</tr>
<tr>
<td>Upper columns</td>
<td>0.61</td>
</tr>
<tr>
<td>Base columns</td>
<td>0.68</td>
</tr>
<tr>
<td>Pontoons and cross members</td>
<td>0.63</td>
</tr>
<tr>
<td>Heave plates</td>
<td>4.80</td>
</tr>
</tbody>
</table>

For the TLP, a viscous drag coefficient of 0.6 was defined for the three spokes and for the submerged portion of tower. In addition, the surface at the platform keel was considered as a heave plate with a drag coefficient of 4.8 and a effective radius of 7.5 m.

The mooring system of the three platforms is composed by three lines. The spar design has one downwind line aligned with the nominal wind direction (line 1) and two upwind lines forming 60° with the wind direction (lines 2 and 3). For the semisubmersible and the TLP, the configuration is the opposite: one line is upwind and parallel to the nominal wind direction (line 2) and the other two are downwind forming 60° with the wind direction (lines 1 and 3).

The characteristics of the mooring systems are summarized in Table 6.4. A line-seabed contact model with no friction was included.
Table 6.4: Main parameters of the three mooring systems

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Spar</th>
<th>Semisubmersible</th>
<th>TLP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of mooring lines</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Angle between adjacent lines</td>
<td>120°</td>
<td>120°</td>
<td>120°</td>
</tr>
<tr>
<td>Depth to anchors below SWL (m)</td>
<td>200</td>
<td>200</td>
<td>200</td>
</tr>
<tr>
<td>Depth to fairleads below SWL (m)</td>
<td>70.0</td>
<td>14.0</td>
<td>28.5</td>
</tr>
<tr>
<td>Radius to anchors from platform centreline (m)</td>
<td>445</td>
<td>837.6</td>
<td>30.0</td>
</tr>
<tr>
<td>Radius to fairleads from platform centreline (m)</td>
<td>5.2</td>
<td>40.868</td>
<td>30.0</td>
</tr>
<tr>
<td>Unstretched length (m)</td>
<td>468</td>
<td>835.5</td>
<td>171.399</td>
</tr>
<tr>
<td>Line diameter (m)</td>
<td>0.09</td>
<td>0.0766</td>
<td>0.2216</td>
</tr>
<tr>
<td>Equivalent mass density (kg/m)</td>
<td>145</td>
<td>113.35</td>
<td>302.89</td>
</tr>
<tr>
<td>Equivalent extensional stiffness (kN)</td>
<td>384,243</td>
<td>753,600</td>
<td>7,720,000</td>
</tr>
</tbody>
</table>

Parameters in Table 6.4 are sufficient for the construction of a quasi-static mooring model, while for the dynamic model the additional parameters presented in Table 6.5 are required. According to [16] the added mass coefficient was set to 1. A normal drag coefficient of 1.6 and no tangential drag were set for all lines based on [23]. A seabed vertical stiffness and damping were defined for the dynamic line-seabed contact model. The number of elements of each line were selected after performing a sensitivity study running several simulations with double number of elements with no significant changes.

Table 6.5: Additional parameters of the three mooring systems used in the dynamic model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Spar</th>
<th>Semisubmersible</th>
<th>TLP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Added mass coefficient</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>Normal drag coefficient</td>
<td>1.6</td>
<td>1.6</td>
<td>1.6</td>
</tr>
<tr>
<td>Tangential drag coefficient</td>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Structural damping (%)</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>Seabed vertical stiffness (N/m^2)</td>
<td>500,000</td>
<td>500,000</td>
<td>500,000</td>
</tr>
<tr>
<td>Seabed vertical damping (Ns/m^2)</td>
<td>30,000</td>
<td>30,000</td>
<td>30,000</td>
</tr>
<tr>
<td>Number of elements of the line</td>
<td>45</td>
<td>80</td>
<td>40</td>
</tr>
</tbody>
</table>

Finally, for the spar design the lines are attached to the platform using a delta connection to increase the mooring system stiffness in yaw. Instead of modelling the delta connection, the platform yaw stiffness of the platform was increased with 98340 kNm/rad, as indicated in [44]. The wind turbine supported by the three platforms is the NREL offshore 5 MW baseline,
which is described in [46]. It is a 3-bladed upwind rotor with variable speed control and collective blade pitch angle. The controller used is described in [44]. The main parameters of the wind turbine are collected in Table 6.6.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal power</td>
<td>5 MW</td>
</tr>
<tr>
<td>Rotor diameter</td>
<td>126 m</td>
</tr>
<tr>
<td>Hub height</td>
<td>90 m</td>
</tr>
<tr>
<td>Cut-in, rated, cut-out wind speed</td>
<td>3 m/s, 11.4 m/s, 25 m/s</td>
</tr>
<tr>
<td>Cut-in, rated rotor speed</td>
<td>6.9 rpm, 12.1 rpm</td>
</tr>
<tr>
<td>Rotor mass</td>
<td>110,000 kg</td>
</tr>
<tr>
<td>Nacelle mass</td>
<td>240,000 kg</td>
</tr>
<tr>
<td>Tower mass</td>
<td>347,460 kg</td>
</tr>
</tbody>
</table>

### 6.3 Environmental conditions

A location with a sea depth of approximately 200 m in the coast of Ireland was selected for the computation of the environmental conditions. The exact coordinates of this location are 52° 10′N and 11° 45′W (see Figure 6.2).

![Figure 6.2: Theoretical location selected (Source: Google Maps)](image-url)
An hourly wind and wave database between 2004 to 2011 was generated using the Skiron and the WAM models for the wind and the wave respectively. These data were used to generate the scatter tables that determine the significant wave height ($H_s$) and reference peak spectral period ($T_p$) as function of the wind speed ($V_w$) required for the definition of each particular simulation load case according to the guideline. A Pierson-Moskowitz spectrum was assumed to represent the wave energy distribution. As the metoceanic model used did not include surface current data, a 1-year surface current velocity of 0.609 m/s and a 50-year surface current velocity of 1.31 m/s were assumed. A summary of the environmental parameters is provided in Table 6.7.

Table 6.7: Summary of the location metoceanic data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sea depth</td>
<td>200 m</td>
</tr>
<tr>
<td>1-year significant wave height</td>
<td>9.82 m</td>
</tr>
<tr>
<td>1-year peak spectral period</td>
<td>14.30 s</td>
</tr>
<tr>
<td>50-year significant wave height</td>
<td>11.50 m</td>
</tr>
<tr>
<td>50-year peak spectral period</td>
<td>15.48 s</td>
</tr>
</tbody>
</table>

6.4 Design load cases simulated for the fatigue and ultimate loads

The computation of the fatigue and the ultimate loads followed the requirements of the IEC 61400-3 guideline. This guideline defines design load cases (DLC) covering all the significant conditions that an offshore wind turbine may experience. The guideline combines the wind turbine operational modes or design situations with external conditions including the control actions and the protection system. The load cases are divided into fatigue and ultimate cases. The corresponding safety factor was applied to the ultimate loads according to the guideline specifications. The fatigue cases recreate different scenarios, such as power production, power production followed by grid loss, normal shut down and parked. Table 6.8 shows a description of the fatigue load cases. Cases of start-up, parked and fault and installation are usually considered with low impact on fatigue loads and were not included in the equivalent loads computation.
Table 6.8: Fatigue load cases definition

<table>
<thead>
<tr>
<th>Design situation</th>
<th>DLC</th>
<th>Wind Condition</th>
<th>Waves</th>
<th>Wind-Waves directionality</th>
<th>Sea currents</th>
<th>Water level</th>
<th>Other</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power production</td>
<td>1.2</td>
<td>NTM</td>
<td>NSS</td>
<td>COD, MUL</td>
<td>No</td>
<td>NWLR or MSL</td>
<td></td>
</tr>
<tr>
<td>Power production + fault</td>
<td>2.4</td>
<td>NTM</td>
<td>NSS</td>
<td>COD, UNI</td>
<td>No</td>
<td>NWLR or MSL</td>
<td>Control, protection or electrical faults</td>
</tr>
<tr>
<td>Normal shut down</td>
<td>4.1</td>
<td>NWP</td>
<td>NSS</td>
<td>COD, UNI</td>
<td>No</td>
<td>NWLR or MSL</td>
<td></td>
</tr>
<tr>
<td>Parked (standing still/idling)</td>
<td>6.4</td>
<td>NTM</td>
<td>NSS</td>
<td>COD, MUL</td>
<td>No</td>
<td>NWLR or MSL</td>
<td></td>
</tr>
</tbody>
</table>

COD: Co-Directional NSS: Normal Sea State NTM: Normal Turbulence Model NWLR: Normal Water Level Range
NWP: Normal Wind Profile MSL: Mean Sea Level MUL: Multi-Directional UNI: Uni-Directional

The Rainflow cycle counting method was applied to the fatigue load cases time series to obtain the number of cycles and range. For the calculation of the fatigue equivalent loads, the cycles of each load case are weighted according to the relative importance of the case in the turbine lifetime. The fatigue equivalent loads were calculated based on $10^7$ cycles in 20 years of lifetime with the expression:

$$F_{\text{equivalent}} = \left( \frac{\sum n_i S_i^m}{T_h f} \right)^{\frac{1}{m}}$$  \quad (6.1)

where $n_i$ is the number of cycles in the load range $S_i$, $T_h$ is the duration of the original time history, $f$ is the frequency of the equivalent load and $m$ is the inverse S-N material slope.

The ultimate load cases recreate extreme scenarios as emergency shutdowns, parked and fault conditions and extreme environmental conditions. A brief description of the ultimate load cases is in Table 6.9. More details can be found in the guideline [43]. All the load cases requested by the guideline were computed, with the exception of the case group 8 and a certain number of subcases of the 2.2 group. The case group 8 simulates transport, assembly, maintenance and repair and are usually considered benign. The discarded subcases of the 2.2 group simulate a fault in the nacelle yaw angle control, provoking a yaw rotation of the nacelle (yaw runaway) of 360°. When the yaw angle of the nacelle reaches 180°, the wind attacks the rotor from a direction opposite to the design condition, and the wind turbine behaviour is very sensitive to the control design. The controller used is a baseline controller that does not take into account such specific conditions, consequently not being representa-
tive of the real conditions.
## Table 6.9: Ultimate load cases definition

<table>
<thead>
<tr>
<th>Design situation</th>
<th>DLC</th>
<th>Wind Condition</th>
<th>Waves</th>
<th>Wind-Waves directionality</th>
<th>Sea currents</th>
<th>Water level</th>
<th>Other</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Power production</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>1.1</td>
<td>NTM</td>
<td>NSS</td>
<td>COD, UNI</td>
<td>NCM</td>
<td>MSL</td>
<td></td>
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<tr>
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More than 3,500 load cases were simulated for each platform and for each mooring modelling approach (around 400 fatigue cases and 3,100 extreme cases). In total more than 20,000 cases were computed. The cases were launched in a cluster with 450 cores, taking around 5 days the simulation of the whole set. The FAST and OPASS output files were converted to BLADED binary format and BLADED postprocessing tools were used for the computation of the fatigue and ultimate loads. Figure 6.3 shows the reference system for the loads at the different wind turbine components: blade, low speed shaft (LSS) and tower.

![Reference system for the loads at the different wind turbine components: blade root (left), rotating low speed shaft (center) and tower base (right) (Source: GL [31])]({})

The three forces and moments ($F_x$, $F_y$, $F_z$, $M_x$, $M_y$ and $M_z$) were computed at the blade root, the low speed shaft and the tower base of the wind turbine. In addition, the tension at the anchor and fairlead of each mooring line were also computed. The equivalent fatigue loads for the steel elements (moorings, tower and shaft) are based on a S-N slope of 4, while a slope of 9 was used for the blade. The blade root ultimate loads were calculated based on the extreme loads found at any of the three blades.

In the following sections the results for the fatigue loads (Section 6.5) and for the ultimate loads (Section 6.6) are discussed. For the sake of brevity, only the three moments at each component are presented in these sections, skipping the forces whose behaviour is similar to the corresponding moment. The tension of the mooring lines is presented only at the anchor, having the tension at the fairleads a similar behaviour. Although for some of the platforms and elements selected, the differences introduced by the moorings model in the fatigue or ultimate loads may not be significant, the graphical comparisons are not skipped.
for completeness.

6.5 Impact of mooring lines dynamics on fatigue loads

6.5.1 Spar fatigue loads

The pitch and roll motions have a great impact on the level of loads of the wind turbine components. The simulations using the quasi-static and the dynamic moorings models provide very similar pitch and roll motions for the spar concept because the fairleads are located very close to the center of rotation of the system. The fatigue analysis reveals no significant differences (below 1%) in the equivalent loads obtained with the two mooring models for the wind turbine components (blades, shaft and tower), see Figures 6.4, 6.5 and 6.6. On the other hand, the additional drag introduced by the dynamic model of the lines slightly reduces the translational surge and sway motions of the platform in comparison with the quasi-static model. Though this implies opposite effects on the tension of the lines (the drag would increase the line tension for a prescribed movement, but the additional damping reduces the motions of the line fairlead decreasing the peaks of tension), the combined effect results in a reduction on the tension peaks. Figure 6.7 shows significant differences for the fatigue equivalent tension of the moorings computed with the quasi-static and the dynamic models. The quasi-static model provides up to 8% higher tensions than the dynamic. As the environmental conditions of the fatigue cases are moderate, the accelerations of the platform are not high and the inertial effect of the lines does not have an important impact on the computation of the spar fatigue loads.
CHAPTER 6. MOORING DYNAMICS ON FATIGUE AND ULTIMATE LOADS

Figure 6.4: Spar combined blade root fatigue equivalent loads

Figure 6.5: Spar low speed shaft fatigue equivalent loads

Figure 6.6: Spar tower base fatigue equivalent loads

Figure 6.7: Spar moorings equivalent tension at anchor
6.5.2 Semisubmersible fatigue loads

In contrast with the spar concept, in the semisubmersible model the lines are attached at a considerable radius from the platform center, providing an important restoring arm for the pitch and roll rotations. In consequence, the additional damping of the dynamic model decreases the amplitude of the platform rotations. The effect on the blades and the shaft fatigue loads is minimal (see Figures 6.8 and 6.9). The differences in the platform rotations have a significant impact on the tower base moment, see Figure 6.10, due to the gravity and inertial forces of the tower top mass. The dynamic model introduces reductions of 13% in the tower base $M_x$ component and 2% in the $M_y$ component.

The effect of the mooring dynamics is very relevant for the fatigue of the lines. Figure 6.11 shows that the dynamic model increases the tension equivalent load in 26% and 30% for the downwind lines 1 and 3, respectively and 57% for the upwind line 2, in comparison with the values provided when the quasi-static model is used.

![Figure 6.8: Semisubmersible combined blade root fatigue equivalent loads](image1)

![Figure 6.9: Semisubmersible low speed shaft fatigue equivalent loads](image2)
The great impact of the mooring dynamics on the fatigue of the lines is caused by the presence of the semisubmersible heave natural period (17.5 s) inside the wave spectrum. It has been verified that the cases with $T_p = 17.5$ s provide 60% of the total fatigue load when the dynamic mooring model is used, and only 9.6% if it is the quasi-static. When a natural period of the platform is inside the wave spectrum, the guideline requires the simulation of cases with that particular wave peak period. In the UMaine Hywind spar these cases with $T_p = 17.5$ s produce a great excitation of the platform motion, activating the inertial effects of the lines and increasing the tensions for the dynamic model. The cases with $T_p = 17.5$ s have the highest contribution to the tension equivalent fatigue loads.

Figure 6.12 presents the Power Spectral Density (PSD) of the platform heave displacement for 4 power production fatigue load cases computed with the OPASS dynamic mooring model. Each of these cases has different wave spectrum peak period (7.05 s, 10.53 s, 13.29 s and 17.50 s) and the same turbulent wind of 12.1 m/s mean speed. The PSD shows that the platform heave motion increases as the peak frequency approaches to the heave natural frequency, becoming very important for the case with $T_p = 17.50$ s. Figure 6.13 shows the PSD of the upwind line 2 tension for the same 4 cases. A peak at the platform surge natural frequency (0.009 Hz) is present in all the simulations, but the highest peak appears for $T_p = 17.50$ s (0.057 Hz), showing that the platform heave excitation plays an important role on the fatigue of the lines.
Figure 6.12: PSD of the semisubmersible heave displacement with different $T_p$ using dynamic mooring model

Figure 6.13: PSD of the semisubmersible line 2 tension with different $T_p$ using dynamic mooring model

Figure 6.14 shows, for a case with $T_p = 17.50$ s, that the amplitude of the heave displacement is slightly lower due to the effect of damping when the dynamic model is used if compared to the quasi-static, where the damping is absent. Figure 6.15 shows how the inertial effects captured by the dynamic model produce an important increase on the tension peaks. This behaviour of the tension is typically observed in cases computed with the dynamic moorings model and important accelerations of the line fairlead.
6.5.3 TLP fatigue loads

The motions of the TLP platform, both translations and rotations, are more damped in the cases simulated with the dynamic moorings model that includes the drag on the lines than in those using the quasi-static model, where this effect is absent. Nevertheless, Figures 6.14 and 6.15 show that there are no significant differences on the fatigue loads of the blade root and the shaft obtained with the quasi-static and the dynamic mooring models. The reduced rotations of the platform obtained with the dynamic model are reflected in a reduction of the tower base loads, see Figure 6.18. This decrease is around 30% for the $M_x$ component of the tower base moment and only 2.5% for the $M_y$ component, where the effect of the lines damping is hidden by the aerodynamic loading at the rotor.

The fatigue equivalent tensions obtained with the dynamic model are lower in comparison with the quasi-static approach for the three lines, as Figure 6.19 shows. The differences are of 2% for the line 2 (upwind), and 4.5% and 4% for lines 1 and 3 (downwind). Again, the damping captured the dynamic model reduces the platform motions resulting in a decrease of the amplitude of the tension peaks.
CHAPTER 6. MOORING DYNAMICS ON FATIGUE AND ULTIMATE LOADS

Figure 6.16: TLP combined blade root fatigue equivalent loads

Figure 6.17: TLP low speed shaft fatigue equivalent loads

Figure 6.18: TLP tower base fatigue equivalent loads

Figure 6.19: TLP moorings equivalent tension at the anchors
6.5.4 Summary of the effect of mooring lines dynamics on the fatigue loads of the three platforms

In this Subsection, the relative differences between the loads computed using the quasi-static and the dynamic moorings model are compared for the three platforms to identify in which components and platform designs the effect of the lines dynamics is more important. A positive value in the differences means higher loads calculated using dynamic lines model and a negative value means lower loads calculated by the dynamic model. The reference value for the calculation of the percentages of difference is the quasi-static.

Figures 6.20 and 6.21 show that the effect of the lines dynamics only slightly affect the equivalent fatigue loads at the blade root and the shaft for any of the three platforms, with differences below 0.3% in the blade root and 2% in the shaft. Nevertheless, small differences in fatigue loads of an element can have an impact on the design.

The mooring lines dynamic model produces a decrease in the tower fatigue loads (Figure 6.22) for the three platform designs. This decrease is higher for the component $M_x$ of the tower base moment, in particular for the TLP platform (31%), being also important for the semisubmersible (13%). The decrease in the $M_y$ component is lower due to the effect of the rotor aerodynamic thrust: around 2% and 2.5% for the semisubmersible and the TLP, respectively. For the spar, the reduction in the tower base moment is very small due to the reduced influence of the mooring system on the rotations. Figure 6.23 shows that the effect of dynamics is very important on the line tension, reducing the fatigue loads for the spar and the TLP due to the extra damping not captured by the quasi-static approach. For the spar, the decrease is around 8% for the upwind lines 2 and 3, and 2% for the downwind line. For the TLP the mooring approach is not so critical: using the dynamic model decreases less than 5% the equivalent tension in the upwind lines and around 2% in the downwind line. For the semisubmersible platform, where the heave natural frequency is inside the wave spectrum, the effect of the mooring dynamics is different than in the other two platforms. The excitation of the platform’s heave displacement produces high velocities and accelerations in the lines increasing the loads due to the drag and inertia. In this case, the dynamic model provides 57% higher equivalent tension loads for the upwind line (line 2) with respect to the quasi-static, and between 25% - 30% for the downwind lines (lines 1 and 3).
6.5.5 Importance of the DLC’s groups on the fatigue equivalent loads

Finally, the relative importance of each case group in the total fatigue equivalent loads was analyzed and can be summarized in the following points:
• DLC 1.2 provides around 99% of the fatigue load for almost all the wind turbine components in the three platforms. The only exception is the tower base load in the TLP, where DLC 6.4 contributes with 3-5% of the load. The importance of DLC 2.4 and DLC 4.1 is negligible.

• For the equivalent fatigue tension of the lines, the role of DLC 6.4 is higher than in the wind turbine components, specially when dynamic mooring model is used. DLC 2.4 and DLC 4.1 are also negligible.

• For the spar, DLC 6.4 provides 6% of the fatigue load in the downwind line with the dynamic model, but less than 1% with the quasi-static.

• For the semisubmersible, if the dynamic model is used, DLC 6.4 contributes with 14% of the fatigue load in the two downwind lines and around 7% in the upwind line. But if the quasi-static model is used, the contribution of DLC 6.4 is reduced to 3% for the downwind lines and below 1% for the upwind.

• Finally, for the TLP, the relative importance of the DLC’s in the fatigue calculation is similar for both mooring models; DLC 6.4 contributes with 6-7% of the total fatigue damage in the downwind lines and 3.6% in the upwind line, no matter which model is used. The rest of the fatigue load is provided by DLC 1.2.

### 6.6 Impact of mooring lines dynamics on ultimate loads

In this section, the ultimate loads are presented in bar diagrams, and at the top of each bar, the load case group providing each ultimate load is indicated for information. A discussion on the importance of the different load case groups is given at the end of the section.

#### 6.6.1 Spar ultimate loads

The effect of the mooring line dynamics on the ultimate loads of the wind turbine components is very low for the spar, as can be seen in Figures 6.24 (blade root), 6.25 (shaft) and 6.26 (tower base). The differences on the modelling of the lines does not affect the platform motions in roll and pitch due to the position of the fairleads close to the platform rotation center.

The lines extreme tensions are sensitive to the lines model, as it is appreciated in Figure 6.27. The extreme environmental conditions defined for the simulation of the ultimate loads cases induce higher velocities and accelerations to the platform than in the fatigue cases. This leads to higher loads in the lines when the dynamic model is used, due to the drag and...
the inertial effects. The maximum tension of the lines is increased by 11% for line 1 and by 15% for the lines 2 and 3. The minimum tensions are also decreased by the dynamic model.

Figure 6.24: Ultimate loads at blade root for the spar platform

Figure 6.25: Ultimate loads at the low speed shaft for the spar platform
Figure 6.26: Ultimate loads at tower base for the spar platform

Figure 6.27: Ultimate tension at the lines anchors for the spar platform


### 6.6.2 Semisubmersible ultimate loads

For the semisubmersible platform, the effect of the mooring dynamics in the ultimate loads of the wind turbine components is also low, see Figures 6.28 (blade root), 6.29 (low speed shaft) and 6.30 (tower). An exception is the negative minimum $M_y$ component of the tower bending moment, increased around 20% (in absolute value) when dynamics are included. Nevertheless the absolute value of the positive $M_y$ component is higher, and therefore the negative component is not a design driver.

The impact of the line dynamics in the line tension is important, producing an increment of the tension measured at the anchors. The increase of the maximum tension is 12% and 18% for the two downwind lines (lines 1 and 3, respectively) and 42% for the upwind line (line 2) which has the highest mean tension. The mooring dynamics also produce a decrease of the tension minimum values. In particular, for the line 2, the dynamic model predicts a loss of tension in the line that is not captured by the quasi-static approach.

![Figure 6.28: Ultimate loads at blade root for the semisubmersible platform](image-url)
Figure 6.29: Ultimate loads at the low speed shaft for the semisubmersible platform

Figure 6.30: Ultimate loads at tower base for the semisubmersible platform
6.6.3 TLP ultimate loads

The influence of the mooring dynamics on the ultimate loads of the turbine components is more relevant for the TLP than in the other two platforms. For the blade root, differences between the loads provided with the quasi-static and the dynamic models are moderate and without a clear tendency, see Figure 6.32. For the maximum loads, all the differences are below 2%. Regarding the negative values of the moments, the dynamic model increases 5% the absolute value of $M_x$, but decreases 8% the absolute value of the $M_y$ component compared to the quasi-static.

The ultimate loads at the low speed shaft are compared in Figure 6.33. The dynamic and quasi-static models provide similar loads, with differences below 1%, with the exception of the negative value of the torsional moment $M_x$, being the value provided by the dynamic model one fifth of the quasi-static.

The loads at the tower base, showed in Figure 6.34, are the most affected by the dynamic model. The damping of the pitch and roll motions of the platform by the dynamic model is reflected in an important decrease of the maximum bending moments at the tower base. This decrease is around 34% for the $M_x$ component and 24% for the $M_y$ component.

Finally, Figure 6.35 shows that the extreme tension at the lines is also decreased if the dynamic model is used. Once more the drag of the lines damps the platform motions, reducing
the tension of the lines. The inertial effects in the TLP lines are low because the high stiffness of the taut mooring system avoids high accelerations of the platform even in the hard environmental conditions defined for the ultimate cases. The reduction in the maximum tension is between 25% (lines 1 and 2) and 45% (line 3). Both models predict the loss of tension of the lines 1 and 3. For the line 2 (in the upwind direction), the quasi-static model also predicts a loss of tension, but the dynamic model provides a minimum tension of 1072 kN.

Figure 6.32: Ultimate loads at blade root for the TLP platform
Figure 6.33: Ultimate loads at the low speed shaft for the TLP platform

Figure 6.34: Ultimate loads at tower base for the TLP platform
6.6.4 Summary of the effect of mooring lines dynamics on the ultimate loads of the three platforms

The relative differences between the ultimate loads computed using the quasi-static model and the dynamic model are compared in this section for the three platforms. For the calculation of the differences, the absolute value of the maximum and minimum load of each component has been considered. A positive value of the relative differences corresponds to higher ultimate loads calculated with the dynamic model and negative differences correspond to lower loads calculated with the dynamic model, in comparison with the quasi-static. The quasi-static results are the reference for the calculation of the percentages of difference. Figures 6.36 and 6.37 show that the effect in the ultimate loads introduced by the dynamic model are not relevant for the blade root and the shaft in any of the platforms, having differences always below 2.5%.

Figure 6.38 reveals that the bending moments at the tower base are considerably affected by the lines dynamics in the TLP platform. The $M_x$ and $M_y$ components are decreased 34% and 24% respectively when compared with the quasi-static. In the spar and semisubmersible platforms the line dynamics affects much less the tower base moments with maximum differences between both models lower than 2%.

Regarding the lines maximum tensions, Figure 6.39 shows that the effect is important for the three designs. In the case of the spar, the use of the dynamic mooring lines model in-
creases the maximum tension in the lines between 11% and 15%. For the semisubmersible, the increase is higher, up to 42% for the upwind line. On the other hand, the moorings dynamics produces an important decrease in the line tension for the TLP. In this concept with taut mooring lines, the inertial effects have lower importance and the drag of the dynamic approach reduces the amplitude of the platform motions resulting in a reduction of the maximum tension in the lines between 25% and 45%.

Figure 6.36: Differences in the blade ultimate loads computed with moorings dynamics vs. quasi-static

Figure 6.37: Differences in the shaft ultimate loads computed with moorings dynamics vs. quasi-static

Figure 6.38: Differences in the tower base ultimate loads computed with moorings dynamics vs. quasi-static

Figure 6.39: Differences in the lines anchor ultimate tension computed with moorings dynamics vs. quasi-static
6.6.5 Importance of the DLC’s groups on the ultimate loads

An overview of the importance of the load case groups in the ultimate loads is given in this subsection, although a deeper analysis will be done in future work. The group of cases providing each ultimate load was indicated in the plots of the ultimate loads (Figures 6.24 to 6.35).

- The group of cases DLC 6.1 is driving a large number of extreme loads. It always determines the maximum tension at the anchor lines for all the platforms, with the exception of the maximum tension at the semisubmersible with the quasi-static mooring model, given by DLC 6.2. In addition, DLC 6.1 determines the maximum $M_x$ component of the blade root moment for the spar and the TLP and the maximum $M_x$ component of the tower base moment for the three platforms.

- DLC 1.3 has also great importance, marking the extreme shaft bending moment in the three platform concepts and the blade root bending moment for the semisubmersible ($M_x$ and $M_y$ components) and the TLP ($M_y$ component). It also provides the extreme torsional moment of the tower for the three platforms.

- DLC 1.6 is particularly relevant for the spar platform, where it is responsible of the extreme loads for the blade $M_y$, the shaft $M_z$ and the tower $M_y$ components of the bending moments. For the TLP it also determines the maximum tower base $M_y$ moment with the dynamic model.

- DLC 5.1 provides the maximum torsional moment at the shaft for all the platform designs.

- Finally, DLC 2.1 provides the maximum $M_y$ component of the tower base moment for the semisubmersible.

6.7 Concluding remarks

A comprehensive study on the influence of mooring line dynamics on the calculation of wind turbine loads following the standard certification requirements for the wind energy industry has been presented. Three different platform designs (spar, semisubmersible and TLP) were studied to characterize the effect of the mooring dynamics on each concept. The loads were computed following the methodology defined by the IEC 61400-3 Edition 1 guideline, which implied the computation and postprocessing of more than 20,000 cases. The fatigue and ultimate loads obtained for each platform design using a quasi-static and a dynamic moorings model were compared. The impact of moorings dynamics in the results provided by integrated simulation codes
depends on the topology of the floating platform and the component considered. In general, the influence of mooring dynamics in both fatigue and ultimate loads increases as elements located closer to the platform are evaluated; the blade and the shaft loads are only slightly modified by the mooring dynamics in all the platform designs, the tower base loads can be significantly affected depending on the platform concept, and the mooring lines tensions strongly depend on the lines dynamics both in fatigue and extreme loads in all the platform concepts evaluated.

The equivalent fatigue tension at the anchors of the three platforms are impacted by the moorings dynamics, but the effect depends on the platform considered. For the semisubmersible, with the heave natural period inside the wave spectrum, dynamics increase the equivalent tension up to 57%. For the spar and the TLP, the equivalent tensions are moderately decreased (up to 8%).

For the spar, the effect of mooring dynamics on the fatigue of the wind turbine components (blades, shaft and tower) is low due to the low influence of the mooring system on the pitch and roll motions. For the semisubmersible and the TLP, moorings dynamics introduce a significant decrease in the fatigue of the wind turbine components, in particular for the tower. Regarding ultimate loads, mooring dynamics cannot be neglected for the calculation of the extreme line tensions in any of the three platforms studied. Including mooring dynamics in the integrated simulations can have diverse effects on the extreme line tensions depending on the platform type, with an increase (spar and semisubmersible) or decrease (TLP) of up to 40% with respect to the quasi-static.

Mooring dynamics are definitely needed for the calculation of the tower base ultimate loads in the TLP design, where the extreme moments are decreased between 24% and 34%. For the other platforms and wind turbine elements, the effect on the ultimate loads is much more limited.

As a final remark, this study reveals that mooring dynamics has the greatest impact in the computation of:

- Fatigue equivalent tensions and ultimate tensions of the mooring lines in any platform configuration.

- In particular, fatigue equivalent tensions of the moorings for the semisubmersible, with one natural frequency inside the wave spectrum.

- Fatigue and ultimate loads of the components of the wind turbine supported by the TLP.

- Tower base fatigue for platforms with fairleads located far from the platform center, as the semisubmersible and the TLP.
Chapter 7

Conclusions and future lines of research

7.1 Conclusions

In this research, a dynamic mooring lines simulation code has been built, verified within the IEA Annex 30 simulation codes benchmark (OC4) and then experimentally validated with results from water tank tests. Once validated, the code has been integrated in a simulation tool for floating wind turbines and it has been used to evaluate the effect of mooring dynamics on the computation of ultimate and fatigue loads for the different existing topologies of floating wind turbines. This evaluation has provided indications of great interest for the designers to determine the required level of complexity on the mooring system modelling for integrated floating wind turbine simulations, depending on the type of platform, the simulation case and the expected results.

In Chapter 2 the objectives of this research were formulated as questions to be answered in the conclusions of the dissertation. In the following paragraphs the conclusions obtained in relation to each of these questions are presented.

Can mooring dynamics be efficiently included in the integrated simulation tools used for design and certification of offshore floating wind turbines?

The dynamic mooring lines code developed in this work is based on a Finite Element Method formulation. The computational cost has been reduced using a Lumped Mass formulation. The efficiency of this approach to include mooring dynamics in the integrated floating wind turbine model has been demonstrated in this work: the developed model has been used to simulate the complete set of load cases required by the IEC 61400-3 guideline for the certification of ultimate and fatigue loads. The certification load calculation was performed for the three main existing concepts of floating wind turbine platforms, requiring the launching of several thousands of simulation cases and fulfilling the requirements that a certification entity would request to an industrial manufacturer.

How can mooring dynamic models be experimentally tested and validated?
A test campaign at the École Centrale de Nantes (ECN) wave tank has been performed with the purpose of validating the mooring lines dynamic code. An original testing setup for the mooring dynamics has been designed consisting on the installation of a submerged chain in a water basin, with one end anchored at the bottom of the basin and the suspension point being excited with harmonic horizontal motions of different frequencies. The tension of the line at the fairlead was measured using a load cell. It is remarkable that the trajectories of markers located at different positions along the length of the line were also measured using a system of submerged cameras, resulting in a very detailed dataset of measurements. The different cases tested produce different dynamic states of the line, including situations where the line totally loses tension and subsequently recovers producing a snap load. This is a highly dynamic situation that is challenging and has to be considered when validating a dynamic mooring model.

**Can computational models represent accurately the tension and motions of mooring lines?**

The accuracy of the dynamic model has been evaluated against the experimental data obtained at the water tank tests with good results. Two configurations of the chain have been studied, with different tension levels. In the static cases, both the shape of the line and the tension at the suspension point compare very well between the computations and the experiments, presenting negligible discrepancies. The dynamic experimental results included different dynamic conditions as harmonic response, loss of tension and snap loads. In all these conditions, the code was able to predict the motion of the markers at different chain positions accurately and also the tension at the fairlead was captured with precision. The highest disagreement appeared in the cases with the highest frequency of the excitation motion, where the maximum tension was overpredicted by the code by around 4.5%.

**Are the values provided by guidelines and bibliography for drag and friction coefficients adequate for the representation of the real mooring dynamics?**

The experimental results also showed that the drag coefficients for the chain simulations, that were chosen from guidelines, in general, represent accurately the hydrodynamic loads. As has been mentioned, the maximum tension is slightly overpredicted (around 4.5%) by the code in the cases with the highest frequency of excitation. These differences could be due to the fact that the drag coefficients were not tuned for the cases with the highest excitation frequency. Nevertheless even in these cases the agreement both in motions and in tension is still good in most of the loop.

In the cases with the highest pretension level and highest excitation frequency, the line totally loses the tension during part of the fairlead excitation period and the simulation results are particularly sensitive to the cable-seabed interaction model. In this situation, the modelling of the seabed friction must be considered because it plays an important role avoiding the slide of the line nodes on the seabed during the period with no tension in the line. This small slide of the nodes, that is not present in the experiments due to the friction of the chain with the bottom of the basin, can greatly affect the chain tension after several periods of the
fairlead motion and has to be avoided using an adequate friction value to predict the tension with accuracy. The value of 1 for the friction coefficient, typically used in the bibliography has provided good results in our study. In the cases where the excitation frequency is lower and the lines does not lose tension, the effect of the seabed-line friction coefficient is not significant.

What is the influence of mooring dynamics in the level of loads for the different topologies of floating wind turbines?

An important contribution of this research is the characterization of the effect that the mooring dynamics have on the computation of the certification loads for the three main concepts of platform designs (spar, semisubmersible and TLP). The fatigue and ultimate loads obtained for each platform design using a quasi-static and a dynamic moorings model were compared. These loads were computed according to the IEC 61400-3 Edition 1 guideline, which implied the computation and postprocessing of more than 20,000 cases. In general, the influence of mooring dynamics in both fatigue and ultimate loads increases as elements located closer to the platform are evaluated; the blade and the shaft loads are only slightly modified by the mooring dynamics in all the platform designs, the tower base loads can be significantly affected depending on the platform concept, and the mooring lines tensions strongly depend on the lines dynamics both in fatigue and extreme loads in all the platform concepts evaluated.

Looking in particular to the equivalent fatigue tension of the lines, they are impacted by the moorings dynamics for the three platforms studied, but the effect is more pronounced for the semisubmersible, that presents the heave natural period inside the wave spectrum. For this platform concept, the dynamic moorings model increases the equivalent tension up to 57% and for the other two platforms, equivalent fatigue tensions are moderately decreased. The equivalent fatigue loads at the wind turbine elements, in particular at the tower base, are significantly decreased by the mooring dynamics.

Relating the ultimate loads, the consideration of mooring dynamics in the integrated simulations can have diverse effects on the extreme line tensions depending on the platform type, with an increase (spar and semisubmersible) or decrease (TLP) of up to 40% with respect to the quasi-static. In addition, mooring dynamics decrease the tower base ultimate loads for the TLP design between 24% and 34%. For the other platforms the effect on the tower ultimate loads is much more limited.

When mooring dynamics must be included in the integrated simulation of floating wind turbines?

The study of the impact of mooring dynamics on the computation of fatigue and ultimate loads according to the certification guidelines has shown that dynamic models should be included for the computation of:

- Fatigue equivalent tensions and ultimate tensions of the mooring lines in any platform configuration.

- In particular, fatigue equivalent tensions of the moorings for the semisubmersible design
studied, that presents the heave natural frequency inside the wave spectrum.

- Fatigue and ultimate loads of the components of the wind turbine supported by the TLP.

- Tower base fatigue for platforms with fairleads located far from the platform center, as the semisubmersible and the TLP.
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