Wind Turbine Structural Dynamics – A Review of the Principles for Modern Power Generation, Onshore and Offshore

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ABSTRACT
Wind turbines for electricity production have two seemingly opposing constraints; they need to be structural secure yet of low cost. To meet the first constraint, it would be an obvious choice to design a stiff structure of consequently large mass but this would drive up the cost. By reducing the mass a more cost-effective turbine can be realized. However, such lightweight structures are by definition more flexible. To design a cost-effective flexible system, thorough understanding of the dynamics is essential. This paper reviews the theoretical basics of the dynamic design options and applies these to realistic situations, including offshore machines under wave action. The wind energy converter and the support structure form an integrated dynamic system that must be developed in mutual interdependency and close cooperation. This paper provides a contribution to this integration process by extending the design approach initiated in the Opti-OW ECS study [1] and the work of Kühn [2].

1. INTRODUCTION
All newly developed wind turbines share the same characteristics: they are larger than their predecessors and of the variable speed concept [3]. ‘Multi-megawatt’ is now a synonym for ‘greater than 2 MW’, with 5 MW in reach. While the output of the turbines is boosted with larger rotors and more powerful generators, the cost is kept as low as possible by reducing the overall weight. This means that the turbine system becomes more flexible and thus more dynamically active.

To make sure the dynamic activity does not influence the system negatively, fully integrated design of the entire wind turbine system is crucial. This may even give cause to a further reduction in cost: in some cases the sum for separate analyses is more than for integrated analysis.

With increased computer power, models that are more sophisticated can be constructed to study the integrated system. But more complex systems are still subject to the principals of dynamics. After an introduction of the turbine characteristics in section 2, the basics of dynamics are described in section 3. These are applied to a “classical” constant speed turbine system in section 4. This fundamental approach is then extended to a modern day offshore wind turbine in section 5, with examples of the benefits of integrated design shown in section 6.

2. TURBINE CHARACTERISTICS
In general, wind turbine systems consist of five physical components: rotor, transmission,
generator, support structure, and control system. A straightforward modelling approach of such a system is shown in figure 1. The system has external inputs from the wind, waves and the grid.

A different approach, which can reduce the amount of calculation radically, is shown in figure 2. The functional components, not the physical components, are the sub modules to interact together. A clear advantage is the approach depicted in figure 1, which has mechanical interaction between tower, rotor, transmission and generator; whereas the approach in figure 2 solves the equations of motion directly and interacts only with other functionalities.

In this article, three example turbines are used to show the application of basic theory on real structures. The first example turbine has been designed within the Opti-OW ECS study. In this study a 3 MW, constant speed, two bladed turbine was used. The characteristics are summarised in table 1.

### Table I. Opti-OW ECS characteristics

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length from seabed to hub</td>
<td>81 m</td>
</tr>
<tr>
<td>Number of blades</td>
<td>2</td>
</tr>
<tr>
<td>Rated power</td>
<td>3 MW</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>22 r.p.m = 0.37 Hz</td>
</tr>
<tr>
<td>Top mass</td>
<td>130,000 kg</td>
</tr>
<tr>
<td>Tower diameter at seabed</td>
<td>3.5 m</td>
</tr>
<tr>
<td>Tower wall thickness at seabed</td>
<td>0.075 m</td>
</tr>
</tbody>
</table>

![Figure 1](image1.png) Modelling approach based on physical parts of the wind turbine system

![Figure 2](image2.png) Modelling approach based on functional characteristics
The next turbine is a 1.5 MW offshore turbine at Utgrunden in Sweden, a wind farm consisting of 7 turbines, which has been installed in 1999.

<table>
<thead>
<tr>
<th>Table II. Utgrunden characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length from seabed</td>
</tr>
<tr>
<td>Number of blades</td>
</tr>
<tr>
<td>Rated power</td>
</tr>
<tr>
<td>Rotational speed range</td>
</tr>
</tbody>
</table>

The third turbine is a Vestas V66 2 MW offshore turbine installed outside Blyth Harbour, northeast England.

<table>
<thead>
<tr>
<th>Table III. Blyth characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length from seabed</td>
</tr>
<tr>
<td>Number of blades</td>
</tr>
<tr>
<td>Rated power</td>
</tr>
<tr>
<td>Rotational speed range</td>
</tr>
<tr>
<td>Top mass</td>
</tr>
<tr>
<td>Tower diameter at seabed</td>
</tr>
<tr>
<td>Tower wall thickness at seabed</td>
</tr>
</tbody>
</table>

3. THE BASICS OF DYNAMICS

The importance of detailed modelling of the structural dynamics can be illustrated most conveniently by considering a single degree of freedom mass-spring-damper system, as shown in figure 3. Note that a complete (offshore) wind turbine system can be thought of as being constructed of a number of coupled mass-spring-damper systems [4].

When a harmonic excitation force \( F(t) \), i.e., a sinusoid, is applied to the mass, the magnitude and phase of the resulting displacement \( u \) strongly depends on the frequency \( \omega \). Three response regions can be distinguished:

a) Quasi-static
b) Resonance
c) Inertia dominated

For frequencies of excitation well below the natural frequency of the system, the response will be quasi-static as illustrated in figure 4a; the displacement of the mass will follow the time varying force almost instantaneously, i.e., with a small phase lag, as if it were excited by a static force. Figure 4b shows a typical response for frequencies of excitation within a narrow region around the system's natural frequency. In this region, the spring force and inertia force almost cancel, producing a response that is a number of times larger than it would be statically. The resulting amplitude is governed by the damping present in the system. For frequencies of

\[
F(t) = \cos(\omega t)
\]

Figure 3  Single degree of freedom mass-spring-damper system
excitation well above the natural frequency, the mass cannot “follow” the movement any longer. Consequently, the response level is low and almost in counter-phase, as illustrated in figure 4c. In this case the inertia of the system dominates the response.

It should be stressed, that in all three figures the magnitude of the excitation force $F(t)$ is identical, but applied at different excitation frequencies.

Figure 4a  Quasi-static response. Solid line: excitation force, and dashed line: simulated response

Figure 4b  Resonant response. Solid line: excitation force, and dashed line: simulated response

Figure 4c  Inertia dominated response. Solid line: excitation force, and dashed line: simulated response
The normalised ratio of the amplitudes in figures 4a–4c, illustrate the general fact that, in steady state, sinusoidal inputs applied to a linear system generate sinusoidal outputs of the same frequency, but differ in magnitude and phase (i.e. shift between the sinusoidal input and output).

The magnitude and phase modifying property of linear systems can be conveniently summarized in one plot: the frequency response function. The frequency response function (FRF) depicts the amplitude ratio of the sinusoidal output to input, as well as the corresponding phase shift, as a function of the frequency of excitation. Figure 5 shows the FRF of the single degree of freedom system depicted in figure 3.

The peak in figure 5 corresponds to the system's natural frequency. The height of the peak is determined by damping. Therefore any resonant problem can be counteracted with adequate damping controls, should the budget allow for it. In dynamics, the frequency of the force is at least as important as its magnitude. Resonant behaviour can cause severe load cases, even failure, but it is most feared for fatigue difficulties. For structures where dynamics are expected to be a problem, detailed knowledge of the expected frequencies of the excitation forces and the natural frequencies of the structure, or parts of the substructure, is vital.

The normalised amplitude ratio is also known as the Dynamic Amplification Factor. The DAF is commonly used in calculations by the wind energy and the offshore technology communities, in the preliminary design phase, to account for the effect of dynamic loads from static response (thereby neglecting the phase information). In general, the required DAF's are derived from time-domain simulations similar to the ones shown in figure 4a–4c.

The important conclusion to be drawn from the basic dynamics review is that the response of a wind turbine system subjected to time-varying loads needs to be carefully addressed, especially for cost-effectiveness.

Figure 5  Frequency response function. Upper figure: magnitude versus frequency, and lower figure: phase lag versus frequency.
4. SOFT TO STIFF

4.1 Introduction

In this section, the presented dynamic approach is applied to a wind turbine system. Firstly, the time varying loads are presented, and then the system properties are modelled.

4.2 The loads

To translate the basic model to a wind turbine system, first the excitation frequencies are examined. The most visible and present source of excitation in a wind turbine system is the rotor. In this example a constant speed turbine will be investigated. The constant rotational speed is the first excitation frequency, mostly referred to as 1P. The second excitation frequency is the rotor blade passing frequency: \( N_bP \) in which \( N_b \) is the number of rotor blades. This means 2P for a turbine equipped with two rotor blades and 3P for a three bladed rotor.

The varying load at both frequencies can best be described with figure 6. A turbulent stream in the wind field will cause an extra load on the blades every time they interact. This extra load will change during a full rotation: the first blade is excited again at 1P. With two blades, there is a 2P response. The difference between a static wind load spectrum and the spectrum left by a 2-bladed turbine due to the rotating rotor is shown in figure 7.

These two frequencies are indicated in a diagram, as shown in figure 8, for the constant speed, 2-bladed, Opti-OCS turbine. The horizontal axis represents the frequency in hertz, and the indicative vertical axis has no value. Though higher order excitations do occur, in this paper, only 1P and 2P are considered for the purpose of illustration. To avoid resonance, the structure should be designed such that its first natural frequency does not coincide with either 1P or 2P. This leaves three possible intervals. A very stiff structure, with a high natural frequency greater than 2P (stiff-stiff), a natural frequency between 1P and 3P: soft-stiff and a very soft structure less than 1P: soft-soft.
4.3 The structure

The structural dynamics of a flexible wind turbine system can be modelled as a flagpole with top mass $M$, as depicted in figure 9. In this shape it resembles the model of the mass-spring-damper system from section 3. The bending flexibility of the tower represents the spring stiffness; the damping is given in the form of a damping coefficient.

For this model, consisting of a uniform beam with a top mass, the following approximation is valid for the calculation of the first natural frequency $f_1$:

$$f_1^2 \approx \frac{3.04}{4\pi^2} \frac{EI}{(M + 0.227 \mu L)L^3}$$

(1)

where:

- $f_1$: First natural frequency
- $M$: Top mass
- $\mu$: Tower mass per meter
- $L$: Tower height
- $EI$: Tower bending stiffness

and with:

- $t$: Tower wall thickness
- $D$: Tower average diameter $= \frac{r_c}{\rho_c}$
- Density of steel

![Figure 9](image_url) Structural model of a flexible wind turbine system

![Figure 8](image_url) Soft to stiff frequency intervals for the constant speed Opti-OW ECS turbine
we find:

\[ I \equiv \frac{1}{8} \pi D^3 \text{t} \quad \text{and} \quad \mu = \rho_c \pi D \text{t} \quad \text{and} \quad a = \frac{M}{\rho_c \pi D \text{t} L} \]

\[ f_1 \equiv \frac{D}{L^2} \sqrt{\frac{E}{104(a+0.227)\rho_c}} \quad (2) \]

As an example, equation (2) is applied to the Opti-OWECS design with 7.5 mm wall thickness constant over the entire height. As summarized in table IV, the structure could have a first natural frequency near 0.25 Hz, a soft-stiff frequency near 0.5 Hz, and stiff-stiff frequency near 1 Hz. Equation (2) is therefore applied to determine the diameter \( D \) corresponding to the specified ‘allowed’ natural frequencies. The results are listed in table IV.

<table>
<thead>
<tr>
<th>Type</th>
<th>frequency</th>
<th>Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Soft-soft</td>
<td>0.25 Hz</td>
<td>2.4 m</td>
</tr>
<tr>
<td>Soft-stiff</td>
<td>0.5 Hz</td>
<td>4.2 m</td>
</tr>
<tr>
<td>Stiff-stiff</td>
<td>1.0 Hz</td>
<td>7.4 m</td>
</tr>
</tbody>
</table>

Because the price of procurement and handling of large tubular piles is mainly influenced by the diameter, from a cost saving point of view the selection of the “softest” structure will be the cheapest and best.

We note that using equation (2) with the data in table I, yields a natural frequency of 0.396 Hz, which would be a soft-stiff structure. The actual natural frequency of the turbine, as established in the Opti-Owec study [1], is 0.289 Hz: a soft-soft structure. This was designed by having, firstly, the tower diameter and wall thickness decrease rapidly above the water line which yields a more flexible structure, and, secondly, including the foundation flexibility as a parameter.

5. LIMITATION OF OPTIONS
5.1 Introduction
In the previous section the basic system properties of a wind turbine were described. The simple model has to be extended to include variable speed, larger turbines and, for offshore turbines, the addition of waves. These influences are described in the next sections.

5.2 Variable speed
Variable speed turbines are gaining market share from constant speed turbines. They offer higher energy capture and lower dynamic loads. For example, the Vestas 2MW turbines in the North Sea near Blyth, northeast England, have a rotational speed ranging from 10.5 to 24.5 RPM. This means that the interval for a soft-stiff design is also narrower, as shown in figure 10. Note that a two bladed turbine would have a range of blade passing frequency, \( 2P \), which would start at 0.34 Hz, which is lower than the upper bound of the \( 1P \) range; the soft-stiff area would have disappeared.

5.3 Larger turbines
The trend to create larger turbines is still strong. This means that rotor blades become longer and generator masses greater. The increase in rotor diameter has a direct effect on the soft to stiff approach. The performance of a turbine can be measured as a function of tip speed ratio, as shown in figure 11. \( C_p \) is the power coefficient, equal to the power extracted from the moving
air divided by the total amount of power in the moving air over the swept area. This curve has a theoretical maximum of 0.593, the Betz limit [6]. \( \lambda \) is the tip speed ratio, equal to the speed of the blade tip divided by the upstream wind speed.

The tip speed ratio is defined by equation (3).

\[
\lambda = \frac{V_{\text{tip}}}{V_w} = \frac{\Omega R}{V_w} = \frac{f_{1p} \pi D}{V_w}
\]  

(3)

So

\[
f_{1p} = \frac{\lambda V_w}{\pi D}
\]  

(4)

This means that the rotation frequency will decrease when the diameter increases. The results of equation (4) for a wind speed of 11.4 m/s and \( \lambda = 8 \) and rotor diameters of 80, 100 and 120m respectively (with 3 blades) are plotted in figure 12.

The increase in rotor diameter also requires a higher hub height and a more powerful, thus heavier, generator. In equation (2), the tower height \( L \) appears as the square. This produces a large decrease of natural frequency with increasing height.

Figure 10  Frequency intervals for a variable speed turbine system

Figure 11  Typical \( C_p-\lambda \) curve

Figure 12  1P and 3P frequencies for 80, 100, and 120 m diameter rotor operating at constant rotational speed
5.4 Waves
For offshore wind turbine systems, an additional excitation force is present, namely sea waves. Wave frequencies are generally lower than the rotational frequency of the rotor. Because waves come in various periods, they span a wide spread in the frequency band. Figure 13 shows the average wave frequencies occurring per year at the 'NLI' location, i.e. the location of the previously mentioned Opti-OW ECS turbine near the Dutch coast [1]. The histogram shows the occurrence of average wave periods per year projected in the previous figure 12 for 80 m diameter, 3-bladed, rotors. From this figure, it is clear that when the offshore wind turbine system is designed with a natural frequency less than the rotation frequency, to avoid resonance, it will enter the frequency domain where resonance due to waves may occur.

6. COMPENSATION
6.1 Introduction
As shown in the previous sections, the goal should be to create a soft-soft support structure, because it uses less steel and is therefore cheaper, but the trends for both structure and excitation forces seem to both converge to this soft area, with a consequent major risk of resonant behaviour. However, there are two further aspects to consider: aerodynamic damping and controllability of variable speed turbines.

6.2 Aerodynamic damping
It was shown, in section 5, that if an offshore soft-soft structure is designed to prevent excitation by the 1P frequency of the rotor, it would encounter sea waves with frequencies near its natural frequency for 25% of the year (see figure 13). However, although resonant behaviour would occur, the dynamic excitation is significantly less than predicted by just structural analysis of the support structure. The rotation of the rotor adds damping to the system that reduces the height of the peak in figure 5 considerably, and thereby the tower-top displacement and the total fatigue.

We note that it was calculated by Van der Tempel [7] that the fatigue life of the Opti-OW ECS support structure is doubled when the turbine is in operation as compared to a parked turbine.

6.3 Variable Speed Frequency Skipping
The variable speed turbines are equipped with comprehensive controls to keep the system running at optimum speed for the particular wind speed. Such variability of the rotation speed narrows the intervals of safe frequencies for the structure and, moreover, the controller can be used to create new intervals. Even though the natural frequency lies in the range of the rotation frequency band, the controller can be programmed to skip the region around the

![Figure 13 Occurrence of wave frequencies with plotted 1P and 3P frequencies](image)
natural frequency. This will prevent the rotor from exciting the tower frequency. The tuning of the controller is best done after installation and measurements of the actual first natural frequency. This is because uncertainties in the soil conditions of the foundation and in the installation works can make the actual frequency deviate appreciably from the design [8]. This frequency skipping has been applied successfully at the Utgrunden Wind Farm in Sweden [2].

7. CONCLUSIONS
In the previous sections the basics of dynamics were used to backup the design philosophy leading to softer support structures. Soft structures require less steel and are therefore cheaper. However, dynamic phenomena need to be identified and dealt with throughout the design, installation and operation phases. The future larger turbines bring both excitation frequencies and the structure's natural frequencies closer together. This means that integrated design and evaluation of the entire system becomes much more important than previously.

The models and calculations used in this paper introduce the principles of analysis, but they are certainly insufficient to describe the full complexity of actual offshore wind turbine systems. Moreover, with increasing blade length, the natural frequencies of the blades themselves reduce and may be excited; therefore these produce another line in the spectrum to watch out for. Basic understanding of the different key dynamic features of an offshore wind turbine system allows a quick scan of the entire system. This should identify problem areas, which can then be investigated and dealt with in detail.

With the popularity of variable speed turbines, the term 'soft-soft' cannot really be maintained; 'soft-stiff-soft-soft' would probably be more accurate, but too long a term to be of practical use.

Dividing the frequency spectrum into soft to stiff intervals is also becoming a more complex matter. The use of a Campbell diagram, whereby excitation frequency and natural frequencies are plotted against rotational speed, offers some extra information, but a more thorough investigation is always required. Understanding the nature, impact and controlled tuning the lines in these diagrams is more valuable than the diagrams themselves.

A final word on the softness of structures and their possibilities. There is an even softer area on the left-hand side of figure 13. If a structure can be designed such that the natural frequency is even lower than the wave frequencies, only inertia-dominated response can be expected. These structures are known as 'compliant towers' in the offshore industry. They are applied in very deep water (400–600 m), for example the Baldpate oil production structure in the Gulf of Mexico in 500 m water, with a first natural period of 31.98 s, i.e. a frequency of 0.03 Hz [9]. Nevertheless, for present offshore wind turbine systems, 'soft-soft' is accepted.

REFERENCES

[5] Vugts, J.H. *Considerations on the dynamics of support structures for an OWEC* Delft University of Technology, Faculty of Civil Engineering and Geoscience, Section Offshore Technology, 2000


