SOFT-SOFT, NOT HARD ENOUGH?

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ABSTRACT Successful application of offshore wind energy requires the effective integration of two complex engineering fields: wind energy and offshore technology. As we are on the threshold of an expected boom of large offshore parks, it is essential that parties on both sides create adequate understanding of each other's technical (im-) possibilities. Situations in which a turbine manufacturer simply requests an offshore company to design and install a support structure on which he can put his rotor-nacelle assembly or, conversely, in which an offshore company comes up with a support structure design and then requests a turbine manufacturer to provide him with a rotor-nacelle assembly, must definitely be avoided. The wind energy converter and the support structure form an integrated dynamic system that must be developed in mutual interdependency and close co-operation. This paper provides a contribution to this integration process by extending the design approach initiated in the Opti-OWECS study [1] and the work of Kühn [2].

1. INTRODUCTION

In the early days of the wind industry, the effects of structural dynamics were either ignored completely, or included through the use of estimated dynamic magnification (i.e. safety) factors. This approach was adequate for the relative small (1 to 500 kW range) and rigid constant speed onshore wind turbine systems.

The wind industry is showing a growing interest towards increasingly lightweight and large (i.e. structurally flexible) wind turbine systems operating at variable speed. This development has been driven mainly by the trend to reduce cost and increase fatigue life in order to reach cost-effective wind turbine systems. However, such systems tend to be dynamically active, and are susceptible to resonances and instabilities. These problems need to be understood and solved before the full potential of such systems can be realized. This implies that our ability to model a wind turbine system's dynamic behaviour accurately becomes more important.

It should be noted that the regulations for certification of wind turbine systems are also mainly based on the experience of rather small and rigid constant rotational speed onshore wind turbine systems (see e.g. Germanischer Lloyd [4]).

2. THE BASICS OF DYNAMICS

In general, (offshore) wind turbine systems basically consist of five physical components: rotor, transmission, generator, support structure, and control system. Each of these components will have impact on the resulting dynamic behaviour of the complete wind turbine system. This implies that (offshore) wind turbine systems should be designed in their entirety as an integrated system to achieve and maintain cost-effective wind turbines.

In this paper we will focus on the structural design and its relation to control system design. The importance of proper modelling of the structural dynamics can be most conveniently illustrated by considering a single degree of freedom mass-spring-damper system as shown in figure 1. Please note that a complete (offshore) wind turbine system can be thought of as being constructed of a number of coupled mass-spring-damper systems [3].

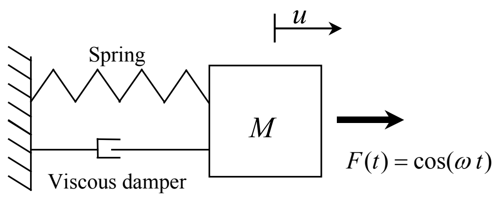

Figure 1. Single degree of freedom mass-spring-damper system

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

a) Quasi-static (or stiff)
b) Resonance
c) Inertia dominated (or soft)

For frequencies of excitation well below the natural frequency of the system the response will be quasi-static as illustrated in figure 2a: the displacement of the mass will follow the time varying force almost instantaneously (i.e. a small phase lag) as if it was excited by a static force. Figure 2b 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 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 2c. In this case the inertia of the system dominates the response and is therefore classified as "soft".

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.

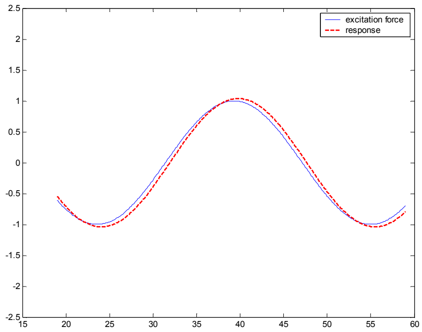

Figure 2a. Quasi-static response. Solid line: excitation force, and dashed line: simulated response.

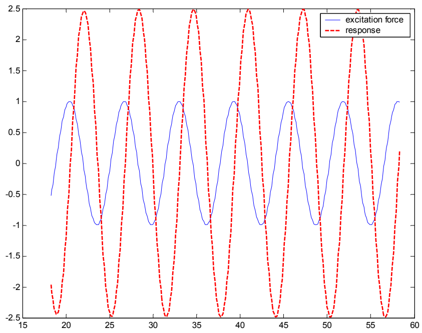

Figure 2b. Resonant response. Solid line: excitation force, and dashed line: simulated response.

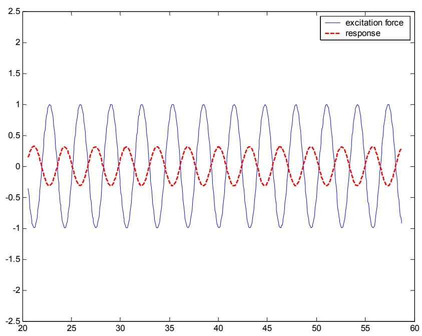

Figure 2c. Inertia dominated response. Solid line: excitation force, and dashed line: simulated response.

The normalised ratio of the amplitude of the figures 2a-2c 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 function of the frequency of excitation. Figure 3 shows the FRF of the single degree of freedom system depicted in figure 1.

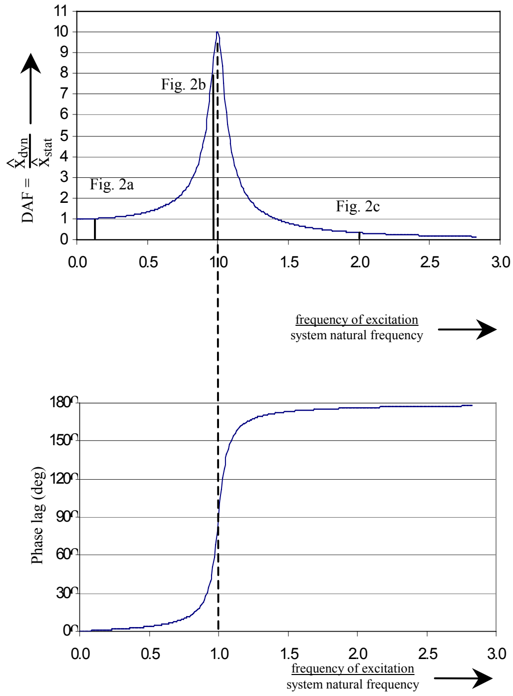

Figure 3. Frequency response function. Upper figure: magnitude versus frequency, and lower figure: phase lag versus frequency.

The peak in figure 3 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 problems. 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 part of the substructures is vital.

The normalised amplitude ratio is also known as the Dynamic Amplification Factor. The DAF is commonly used in the wind energy and offshore technology community calculations 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 2a-2c.

The important conclusion that can 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 when aiming at achieving cost-effectiveness.

3. SOFT TO STIFF

3.1 Introduction

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

3.2 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: 2P for a turbine equipped with two rotor blades, 3P for a three bladed rotor.

The varying load at both frequencies can best be described with figure 4. A turbulence bell in the wind field will cause an extra load on the blade every time it passes through. This extra load will change with a full rotation: the first blade is excited again and at the blade passing speed: every blade feels it. The difference between a static wind load spectrum and the spectrum felt by the turbine due to the rotating rotor is shown in figure 5.

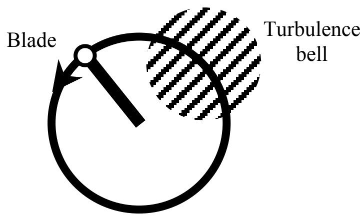

Figure 4. Blade rotating through a turbulence bell

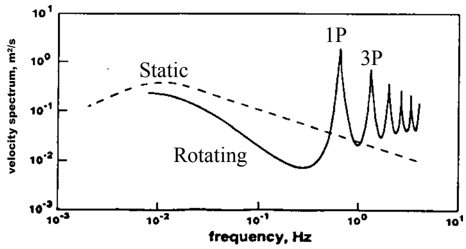

Figure 5. Velocity spectrum from a static and rotating point of view

These two frequencies are plotted in a graph as shown in figure 6. The horizontal axis represents the frequency in hertz, and the vertical axis has no value. Though higher order excitations do occur, in this paper, only 1P and 3P 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 3P. This leaves three possible intervals. A very stiff structure, with a high natural frequency, above 3P (stiff-stiff), a natural frequency between 1P and 3P: soft-stiff and a very soft structure below 1P: soft-soft.

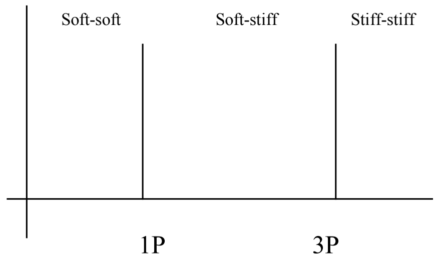

Figure 6. Soft to stiff frequency intervals of a three bladed, constant rotational speed wind turbine

3.3 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 7. In this shape it resembles the model of the mass-spring–damper system from section 2. The bending flexibility of the tower represents the spring stiffness; the damping is given in the form of a damping coefficient.

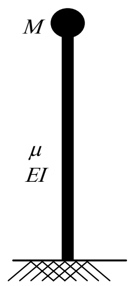

Figure 7. Structural model of a flexible wind turbine system

For this model consisting of a uniform beam with a top mass, the following approximation is valid [5] for the calculation of the first natural frequency:

$$f_1^2 \cong \frac{3.04}{4\pi^2}\frac{EI}{(M+0.227\mu L)L^3} \qquad (1)$$

With:

f_1	First natural frequency
M	Top mass
μ	Tower mass per meter
L	Tower height
EI	Tower bending stiffness

We find with:

$$I \cong \frac{1}{8}\pi D^3 t \text{ and } \mu = \rho_c \pi D t \text{ and } a = \frac{M}{\rho_c \pi D t L}$$

$$f_1 \cong \frac{D}{L^2}\sqrt{\frac{E}{104(a+0.227)\rho_c}} \qquad (2)$$

Where:

t	Tower wall thickness
D	Tower average diameter $= D_{outer} - t = D_{inner} + t$
ρ_c	Density of steel

As an example equation (2) is applied to the Opti-OWECS design. This design consists of a 2 bladed constant speed turbine. The rotational frequency 1P is 0.3Hz, the blade passing frequency 2P is 0.6Hz. A soft-soft structure would then have (for example) a first natural frequency of 0.25Hz, soft-stiff 0.5Hz and stiff-stiff 1Hz. The wall thickness is taken to be constant over the entire height: 7.5mm. The system has a top mass of 130,000kg. Equation (2) is applied to determine the diameter D corresponding to the specified natural frequencies. The results are listed in table I.

Table I. Required diameters per frequency

Type	*Frequency*	*Diameter*
Soft-soft	0.25 Hz	2.4 m
Soft-stiff	0.5 Hz	4.2 m
Stiff-stiff	1.0 Hz	7.4 m

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 best.

4. LIMITATION OF OPTIONS

4.1 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 the addition of waves in the case of offshore turbines. These influences are described in the next sections.

4.2 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 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 8.

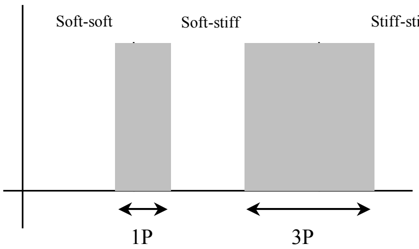

Figure 8. Frequency intervals for a variable speed turbine system

4.3 Larger turbines

The trend to create larger turbines is still strong. This means that rotor blades get longer and generator masses higher. 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 9. C_p is the power coefficient: the amount of power extracted from 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]. λ is the tip speed ratio: tip speed / wind speed.

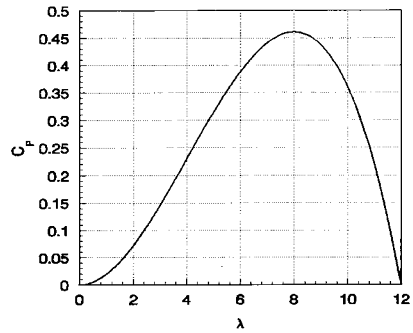

Figure 9. Typical C_p- λ curve

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

$$\lambda = \frac{V_{tip}}{V_w} = \frac{\Omega R}{V_w} = \frac{f_{1P}\pi D}{V_w} \tag{3}$$

So $$f_{1P} = \frac{\lambda V_w}{\pi D} \tag{4}$$

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 λ = 8 and rotor diameters of 80, 100 and 120m respectively (with 3 blades) are plotted in figure 10.

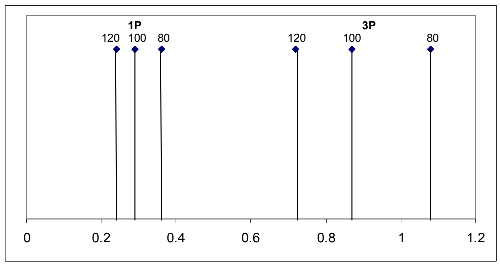

Figure 10. 1P and 3P frequencies for 80, 100, and 120 m diameter rotor operating at constant rotational speed

The increase in rotor diameter also requires a higher hub height and a more powerful, thus heavier, generator. In equation (2) the structure length L is used squared. This means large decrease of natural frequency with increasing height.

4.4 Waves

For offshore wind turbine systems an additional excitation force is present: waves. Wave frequencies are generally lower than the rotational frequency of the rotor. Because waves come in various periods they span a wide area in the frequency band. Figure 11 shows the average wave frequencies occurring per year on the NL1 location, the location of the previously mentioned Opti-OWECS turbine near the Dutch coast [1]. The histogram shows the occurrence of average wave periods per year projected in the previous figure 6. From this figure, it is clear that when the offshore wind turbine system is designed with a natural frequency below the rotation frequency to avoid resonance, it will come into the area where resonance due to waves may occur.

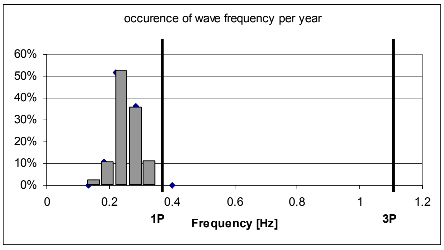

Figure 11. Occurrence of wave frequencies with plotted 1P and 3P frequencies

5. COMPENSATION

5.1 Introduction

As shown in the previous sections, the goal would 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 converge to this soft area with a major risk of resonant behaviour. But there are two comments to be made: aerodynamic damping and controllability of variable speed turbines.

5.2 Aerodynamic damping

It was shown in section 4 that when a soft-soft structure is designed to prevent excitation by the 1P frequency of the rotor, it would encounter waves with frequencies near its natural frequency for 25% of the year (see figure 11). Though resonant behaviour does occur, the dynamic excitation is significantly less than structural analysis of the support structure would reveal. The rotation of the rotor adds damping to the system that reduces the height of the peak in figure 3 considerably and with it the top displacement and the total fatigue.

It was calculated by Van der Tempel [7] that the fatigue life of the Opti-OWECS support structure is doubled when the turbine is in operation when compared to a parked turbine.

5.3 Variable Speed Frequency Skipping

The variable speed turbines are equipped with comprehensive controls to keep the system running at the optimum speed. Though the variability of the rotation speed narrows the intervals of safe frequencies for the structure, 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 area around the natural frequency. This will prevent the rotor from exciting the tower frequency. The tuning of the controller can best be done after installation and measurements of the actual first natural frequency because uncertainties in soil conditions and 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].

6. CONCLUSIONS

6.1 Soft-Soft, not complex enough?

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. But 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 structure's natural frequencies closer together. This means that integrated design and evaluation of the entire structure becomes much more important.

The models and calculations used in this paper are absolutely insufficient to describe the total offshore wind turbine system: with growing blade length, the natural frequencies of the blades also come down. They are just another line in the spectrum to watch out for. Good, basic understanding of the different key dynamic features of an offshore wind turbine system can aid in a quick scan of the entire system to identify problem areas, which can then be investigated and dealt with head-on.

6.2 Soft-Soft, not hard enough?

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 in soft to stiff intervals is also becoming a more complex matter. The use of a Campbell diagram, where excitation frequency and natural frequencies are plotted against rotational speed, offers some extra information, but more thorough investigation is always required. The understanding of the nature, impact and tunability of 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 11. 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 (400-600m), for example the Baldpate in the Gulf of Mexico in 500 m water, with a first natural period of 31.98s, i.e. a frequency of 0.03 Hz [9]. But for now for offshore wind turbine systems soft-soft will have to do.

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