DOWEC 6 MW PRE-DESIGN

Aero-elastic modelling of the DOWEC 6 MW pre-design in PHATAS

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public version

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Acknowledgement / Preface

This report is issued within the framework of the DOWEC project, supported by the EET programme of the Dutch Ministry of Economic Affairs. It is a re-print of a confidential report issued in August 2002 (ECN-CX--01-135) including minor modifications.

A computer model has been composed based on a conceptual design of the 6 MW offshore wind turbine. Calculations with this model should support the further design. This report provides a description of the pre-design and an analysis of the stationary aerodynamic performance and natural frequencies of the 6 MW OWEC, based on calculations with the ECN program PHATAS. Results from dynamic calculations will be described in a next report.

Colleagues Ben Hendriks, Bernard Bulder and Johan Peeringa are thanked for their contributions to the determination and verification of the OWEC description. Danny Winkelaar of ECN contributed text on the modelling of wave loading, chapter 7.2, and Eric van der Hoof wrote about the control of turbine rotor speed, chapter 4.7.

Frank Goezinne of NEG Micon Holland, Jeroen Kooij of LM Glasfiber Holland, and Ton Topper of Ballast Nedam are thanked for providing specifications of the rotor, turbine, and tower and foundation respectively. Also Michiel Zaaijer of Delft University of Technology and others who commented on the report are thanked for their contributions.

Henk-Jan Kooijman, September 2003

Abstract

The aim of the DOWEC project is to develop an integrally optimised 6 MW offshore wind turbine for the use in large wind farms in the North Sea, scheduled for the year 2007. Underlying is a description of the pre-design properties of the DOWEC 6 MW turbine provided by the project partners. Based upon this information an input file has been compiled for computer calculations with the PHATAS program. The results of these calculations will be used to modify and further improve the specifications. It will also contribute to an integrated cost analysis of a 500 MW large wind farm with the 6 MW turbine.

Some remarks regarding the pre-design can already be made. The aerodynamic performance proves to be very good while further dynamic analysis is recommended. Three possible resonance frequencies are indicated in the report. Also recommended are parameter sensitivity studies to verify the correct modelling of wave loading.

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### ABBREVIATIONS

- **b.t.c.** blade tip - tower clearance (vertically w.r.t. MSL, horizontally w.r.t. tower)
- **c.o.g.** centre of gravity
- **CD** Chart Datum
- **DOWEC** Dutch Offshore Wind Energy Converter (project name)
- **DU** Delft University
- **GL** Germanischer Lloyd
- **MSL** Mean Sea Level
- **MT** Metric Ton (1,000 kg)
- **NA** NACA, name of airfoil series
- **NESS** North European Storm Study
- **OWEC** Offshore Wind Energy Converter
- **PHATAS** Program for Horizontal Axis wind Turbine Analysis and Simulation
- **ROWS** Random Ocean Wave Simulator
- **TOR** Terms Of Reference, ref. [10].

### SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
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<tbody>
<tr>
<td>$a$</td>
<td>[-]</td>
<td>slope parameter used in relation between turbulence and wind speed</td>
</tr>
<tr>
<td>$A$</td>
<td>[-]</td>
<td>rotor blade aspect ratio</td>
</tr>
<tr>
<td>$c$</td>
<td>[kg/s]</td>
<td>damping</td>
</tr>
<tr>
<td>$c_a$</td>
<td>[-]</td>
<td>coefficient of added mass (also called hydronamic mass coefficient)</td>
</tr>
<tr>
<td>$c_c$</td>
<td>[kg/s]</td>
<td>critical damping</td>
</tr>
<tr>
<td>$c_d$</td>
<td>[-]</td>
<td>drag coefficient</td>
</tr>
<tr>
<td>$H_s$</td>
<td>[m]</td>
<td>significant wave height (section 7.2)</td>
</tr>
<tr>
<td>$i$</td>
<td>[-]</td>
<td>transmission ratio</td>
</tr>
<tr>
<td>$I$</td>
<td>[kgm$^2$] or [%]</td>
<td>inertia or turbulence intensity</td>
</tr>
<tr>
<td>$J$</td>
<td>[kgm$^2$]</td>
<td>inertia</td>
</tr>
<tr>
<td>$k$</td>
<td>[Nm/rad] or [-]</td>
<td>torsional stiffness or Weibull shape parameter</td>
</tr>
<tr>
<td>$P$</td>
<td>[W]</td>
<td>power</td>
</tr>
<tr>
<td>$p$</td>
<td>[-]</td>
<td>exponent used to describe wind shear</td>
</tr>
<tr>
<td>$Q$</td>
<td>[Nm]</td>
<td>torque</td>
</tr>
<tr>
<td>$R$</td>
<td>[m]</td>
<td>rotor radius</td>
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<tr>
<td>$Re$</td>
<td>[-]</td>
<td>Reynolds number, $Re = V \cdot ('characteristic length') / \nu$</td>
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<td>$T_z$</td>
<td>[s]</td>
<td>mean zero up-crossing wave period (section 7.2)</td>
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<tr>
<td>$V$</td>
<td>[ms]</td>
<td>wind speed</td>
</tr>
<tr>
<td>$y$</td>
<td>[m]</td>
<td>local blade deflection, used to describe eigen mode</td>
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<tr>
<td>$\Omega$</td>
<td>[rpm]</td>
<td>rotor speed</td>
</tr>
<tr>
<td>$\delta$</td>
<td>[-]</td>
<td>logarithmic decrement for damping, $\delta = 2\pi \cdot c / c_c$</td>
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<tr>
<td>$\eta$</td>
<td>[-]</td>
<td>efficiency, here used for drive train</td>
</tr>
<tr>
<td>$\phi$</td>
<td>[deg]</td>
<td>nacelle yaw angle</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>[-]</td>
<td>tip speed ratio, $\lambda = \Omega R / V$</td>
</tr>
<tr>
<td>$\nu$</td>
<td>[m$^2$ / s] or [rad/s]</td>
<td>kinematic viscosity or frequency</td>
</tr>
<tr>
<td>$\theta$</td>
<td>[deg]</td>
<td>blade tip angle w.r.t. rotor plane</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>[-]</td>
<td>standard deviation, here used for wind turbulence</td>
</tr>
<tr>
<td>$\omega$</td>
<td>[rpm]</td>
<td>yaw motor speed</td>
</tr>
<tr>
<td>$\zeta$</td>
<td>[-]</td>
<td>non-dimensional damping, $\zeta = c / c_c$</td>
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SUMMARY

The DOWEC project (Dutch Offshore Wind Energy Converter) is performed by a consortium of industry and research institutes with the final aim to develop a 6 MW offshore wind turbine ready for testing offshore in 2006. The six partners in this EET programme subsidised project are Ballast Nedam, Van Oord ACZ, NEG Micon Holland, LM Glasfiber Holland, Delft University of Technology, and the Energy research Centre of the Netherlands.

A model description has been prepared for computer calculations based on pre-design specifications. Analysis of calculated structural loading and dynamic behaviour will support further design improvements. Results will also be used for an integrated cost analysis of the eventual situation where the turbine is placed in a large offshore wind farm in the North Sea waters.

The first results from calculations show a very good aerodynamic performance of the turbine.

A more elaborate frequency analysis is recommended, particularly on possible resonance between 6P and collective as well as symmetric edgewise bending of the rotor blades. Also the properties that determine the structure bending frequency should be kept in check as the risk of resonance with the nominal rotor passing frequency looms if the bending frequency decreases.

Figure 1: sketch of the DOWEC 6 MW turbine.

The sea bottom consists of sand with an intermediate clay layer between 15 m and 20 m below the seabed.

1 The definitions for reference water depth, tower and support structure may differ from those in other DOWEC reports.
General specifications DOWEC 6 MW pre-design

Nominal power 6.0 MW
Rotor diameter 129 m
Design tip speed 80.0 m/s
Nominal rotor speed 11.844 rpm (at maximum torque)
Air density 1.225 kg/m³
Cut in speed 3.0 m/s
Rated wind speed 12.1 m/s
Cut out speed 25.0 m/s
Rotor hub cone angle -2.5° (upwind)
Rotor tilt angle 5°
Pre bend of the blades 2.0 m at tip, upwind, additional to the -1.0° hub coning.
Support mono pile: 75 m long, 6 m wide cylinder, 30 to 45 m in the seabed, depending on sand wave trough.
tower: 80 m long, tubular shaped.
Hub height above sea level 91.4 m above MSL.
Total mass (approximation) 663 MT (mono pile) + 225 MT (tower) + 189 MT (nacelle) +
83 MT (rotor + hub) + 7 MT (miscellaneous) = 1167 MT.

Figure 2: transport and installation of the DOWEC turbine with the 'Svanen' vessel.
Acknowledgement: A. Vos, Ballast Nedam Engineering B.V.
1. INTRODUCTION

The final result of the DOWEC project should be the realisation of a 6 MW offshore wind turbine. This turbine is optimised for exploitation in large wind farms in the North Sea addressing all design and operational aspects in a cost-effective way. The turbine should be ready in the year 2007. Offshore testing should take place in 2006.

An input file has been compiled by ECN for analysis with the computer program PHATAS-IV release "JAN-2002", hereafter simply referred to as PHATAS.

ECN started the modelling of the pre-design of the DOWEC 6 MW turbine around September 2001. First sources of information were input files prepared by Bernard Bulder and Koert Lindenburg of ECN based upon the NM3000/96 turbine, ref. [17] and [19] (including updates for the 3.6 MW version). Results with the program BLADOPT served as input for the blade manufacturer LM Glashoover Holland to define the geometrical and structural properties of the rotor blade, denoted as LMH64-5. Specific information for the 6 MW turbine and tower given by NEG Micon Holland, and soil and foundation properties provided by Ballast Nedam, completed the specifications for the DOWEC 6 MW pre-design.

The DOWEC 6 MW pre-design is explained in the following chapters. First the geometric rotor design and quasi-stationary aerodynamic performance are given in chapter 2, followed by the structural blade properties in chapter 3. Next the turbine and support are described in chapter 4 and 5 respectively. The dynamics of the OWEC are explained in chapter 6. As a guideline to further calculations recommendations on the modelling of wind and wave loading are given in chapter 7. Finally conclusions and recommendations follow in chapter 8.
2. AERODYNAMIC MODELLING ROTOR

This chapter contains a description of the geometric properties of the pre-design of the 'LMH64-5' rotor blade: the aerodynamic representation of the blade, the structural damping, and the aerodynamic performance based on quasi-stationary calculations. All specifications were provided by LM Glasfiber Holland B.V.

2.1 Blade geometry

The outset for the blade geometry was done through optimisations with BLADOPT. These specifications were then re-designed by LM Glasfiber Holland, ref. [14]. The blade length is 62.7 m. Attached to a hub radius of 1.80 m this gives a rotor radius of 64.5 m (129 m rotor diameter). The blade axis runs along the centre line of the blade root and continues outside the maximum chord position along the points that lie 37.5% aft of the leading edge. The aerodynamic centres are taken at 25% chord distance aft of the leading edge. From the maximum chord position to the cylindrical part of the blade this is linearly increased to 50%. The position of trailing and leading edge, mass centres and aerodynamic centres are plotted in Figure 3. The aerodynamic solidity of the rotor is calculated as 4.936%.

Figure 3: chord wise blade properties (untwisted projection).

Figure 4 and Figure 5 show the chord and twist distribution both according to specifications as well as values printed in the PHATAS output file 'model file'.

Note: The rotor radius of the maximum chord location is 13.8 m, which is relatively near to the rotor centre. For comparison, for the LMH46-5-X00 rotor blade this distance is (1.5 m + 12.0 m) = 13.5 m.

2 For comparison: the solidity of the LMH46-5-X00 rotor blade (baseline) is 5.036%.
"Pre bend" and blade tip tower clearance

According to specifications from LM Glasfiber Holland the pre bend of the blade suction side starts around the maximum chord position and continues with a varying radius of curvature to the blade tip, yielding a flapwise (non twisted) tip displacement of -2.0 m, positive downwind, see Figure 6.

In PHATAS this pre bend is described with a uniform curvature between rotor radii 17.9 m and 53.7 m. Linear extrapolations are used to start the pre bend from zero curvature at the root and to extend it to the specified value at the blade tip. Figure 6 shows that the pre bend in PHATAS matches well with the specifications.

Note: Because LMGH describes the pre bend of the blade suction side an opposite bend is seen for the blade tip section. Because the deflection at the tip of -2.07 m in PHATAS is little more than the value of -2.0 m given by LMGH, the results for minimum horizontal blade tip clearance from PHATAS will be non-conservative with regard to tip tower clearance.
The horizontal blade tower clearance (b.t.c.) between the tip and the tower wall is in unloaded conditions approximately:

\[ \text{b.t.c. (horizontal)} = x_{\text{hub}} + R_{\text{rotor}} \cdot \sin (\alpha_{\text{tilt}} - \alpha_{\text{cone}}) - ('\text{pre bend at blade tip}') - \frac{d_{\text{tower, tip}}}{2} \]

\[ = 5.0 \text{ m} + 64.5 \text{ m} \cdot \sin (5^\circ - (-2.5^\circ)) + 2.0 \text{ m} - 5.51 \text{ m} / 2 = 12.6 \text{ m} \]

Here \((x_{\text{hub}})\) is the distance between tower centre and hub. The tilt angle is positive upwards and the cone angle is here taken positive in downwind direction. \((d_{\text{tower, tip}})\) is the tower diameter at the same height of the blade tip when it points vertically downwards.

### 2.2 Airfoil distribution

The aerodynamic blade properties are described with 5 "DU" airfoils and 1 "NACA64-6" airfoil, having thicknesses between 40% and 18%. After consulting with LM Glasfiber Holland the specified relative blade thickness distribution was smoothed by ECN, see Figure 7. Based upon these modified data the airfoils were positioned according to their relative thickness, see Table 1.

<table>
<thead>
<tr>
<th>input file name</th>
<th>file name</th>
<th>2D data</th>
<th>relative thickness</th>
<th>modelled from blade thickness</th>
<th>modelled from rotor span distance (input)</th>
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<tr>
<td>cylinder1</td>
<td>cylinder1 ((c_d = 0.5))</td>
<td>100%</td>
<td>100%</td>
<td>1.80 m</td>
<td></td>
</tr>
<tr>
<td>cylinder2</td>
<td>cylinder2 ((c_d = 0.35))</td>
<td>-</td>
<td>100%</td>
<td>5.98 m</td>
<td></td>
</tr>
<tr>
<td>DU40_A17</td>
<td>du99_W_405LM_Re7.in</td>
<td>40.50%</td>
<td>~ 60%</td>
<td>10.15 m</td>
<td></td>
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<tr>
<td>DU35_A17</td>
<td>du99_350_Re7.in</td>
<td>35.09%</td>
<td>37.80%</td>
<td>15.00 m</td>
<td></td>
</tr>
<tr>
<td>DU30_A17</td>
<td>du97_W_300LM_Re7.in</td>
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<td>32.55%</td>
<td>20.49 m</td>
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<td>DU25_A17</td>
<td>du91_W2_250_Re7.in</td>
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<td>27.50%</td>
<td>26.79 m</td>
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<tr>
<td>DU21_A17</td>
<td>du93_W_210LM_Re7.in</td>
<td>21.00%</td>
<td>23.00%</td>
<td>34.22 m</td>
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<tr>
<td>NA64_A17</td>
<td>nac64618.in</td>
<td>18.00%</td>
<td>19.50%</td>
<td>42.47 m</td>
<td></td>
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</table>

3 It would have been better if the absolute thickness had been used to smooth the thickness distribution but this was only realised afterwards. The differences are small for a linear chord distribution.
Ruud van Rooij of Delft University of Technology provided 2D measured coefficients for the DU-airfoils for a Reynolds number of 7 million. Table 2 shows that the Reynolds numbers (and aerodynamic performance) for the blade part outside rotor radius 20 m are significantly more for nominal operating conditions. Even more so, the airfoil data for the NACA 64-618 as copied from appendix IV of Abbott and Von Doenhoff, apply to a Reynolds number of 6 million.

For all airfoils measured coefficients were available between angles of attack of approximately -10° and +20°, depending on the airfoil. These coefficients were extended to -180° and +180° using results from the ECN tool StC (Stall Coefficients), release “MAR-2001”, ref. [21]. The hereto-required geometrical properties of the airfoil are the same as used in ref. [17]. The aspect ratio of the blade used for the StC calculations was (A) = 17, which is on the conservative side when comparing it with the PHATAS estimation (A) = 16.4. The complete 2 dimensional airfoil characteristics are given in appendix A.

A correction for the rotational effects on the aerodynamic coefficients is included in the PHATAS code 4. A distribution of the blade with 14 elements of equal width in PHATAS gives an airfoil matching of 93.0%. This is considered to be a good compromise between aerodynamic accuracy and the calculation time (the calculation time increases with the number of blade elements used). The length of the elements is (64.5 m - 1.80 m) / 14 = 4.48 m.

Note: in PHATAS the element at the blade tip is modelled with two aerodynamic annuli.

Next to the cylindrical part of the blade, the DU40 airfoil (40.5% thickness) is modelled between 10.76 m and 15.24 m. For comparison: the location of the largest chord is 13.8 m from the rotor centre.

The first blade element is modelled as purely cylindrical (cd = 0.5) while the second blade element has a lower cd to account for its more ellipsoidal shape. The cd values are irrespective of the angle of attack and are based on figure 8 from the ESDU paper no. 79026, ref. [6], while considering the range of Reynolds number for the rotor blades in actual operation, see Table 2.

---

4 PHATAS applies no correction for rotational effects for blunt sections, i.e. the correction is suppressed if the lift coefficient is smaller than cl = 0.8.

5 Airfoil matching is the proportion between the specified and modelled airfoil distribution in terms of the rotor swept area by the different airfoils.
Table 2: Reynolds number for different radial positions at rated conditions.

2.3 Aerodynamic imbalance

The aerodynamic imbalance in the computer model should take into account possible inaccuracies at the blade hub connection, perturbations due to contamination, due to deformations as a result of imperfect curing of the blade, and more. This imbalance is accounted for by using different set points for the blade pitch angle of the rotor blades. Blade 1 serves by definition as the reference blade. Blade numbers 2 and 3 have an offset in pitch angle of +0.5° and -0.5° respectively with respect to blade 1.

2.4 Aerodynamic performance

Figure 8 shows the power coefficients as function of tip speed ratio calculated with PHATAS for the following conditions:
- zero wind elevation;
- no yaw misalignment;
- not including blade flap, lead lag or torsional deformations;
- exponent in wind shear profile (p) = 0.082, as prescribed in the DOWEC TOR, ref. [10];
- no tower dynamics included.

The optimum operating conditions in partial load are a tip speed ratio of $\lambda = 8.25$ and blade tip angle $\theta = 0^\circ$, see Table 3. Optimising the design between maximum energy yield and minimum design driving loading, different combinations of $\lambda$ and $\theta$ may be considered. The currently defined generator curve in partial load fits a tip speed ratio of $\lambda = 7.75$, see also chapter 4.3, page 24.

| $V$ [m/s] | 12 |
| $\Omega$ [rpm] | 11.8 |
| $\lambda$ [-] | 6.6 |
| $R_{\text{rotor}}$ | chord | $Re$ |
| 1.801 | 3.5 | 2.39E+06 |
| 2.200 | 3.5 | 2.41E+06 |
| 2.799 | 3.5 | 2.44E+06 |
| 3.799 | 3.614 | 2.60E+06 |
| 6.799 | 3.957 | 3.24E+06 |
| 7.299 | 4.014 | 3.37E+06 |
| 8.799 | 4.186 | 3.79E+06 |
| 13.799 | 4.692 | 5.47E+06 |
| 17.499 | 4.599 | 6.36E+06 |
| 24.299 | 4.252 | 7.69E+06 |
| 31.801 | 3.793 | 8.72E+06 |
| 41.801 | 3.193 | 9.47E+06 |
| 51.801 | 2.593 | 9.44E+06 |
| 56.801 | 2.293 | 9.13E+06 |
| 59.801 | 2.023 | 8.47E+06 |
Figure 8: cp-lambda characteristics.

Table 3: aerodynamic power coefficients around optimum tip speed.

<table>
<thead>
<tr>
<th>tip speed ratio</th>
<th>$\theta = -2^\circ$</th>
<th>$\theta = -1^\circ$</th>
<th>$\theta = 0^\circ$</th>
<th>$\theta = 1^\circ$</th>
<th>$\theta = 2^\circ$</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.50</td>
<td>0.4854</td>
<td>0.4927</td>
<td>0.4907</td>
<td>0.4784</td>
<td>0.4567</td>
</tr>
<tr>
<td>7.75</td>
<td>0.4849</td>
<td>0.4955</td>
<td>0.4947 (Q-$\Omega$ design)</td>
<td>0.4825</td>
<td>0.4602</td>
</tr>
<tr>
<td>8.00</td>
<td>0.4827</td>
<td>0.4956</td>
<td>0.4967</td>
<td>0.4853</td>
<td>0.4639</td>
</tr>
<tr>
<td>8.25</td>
<td>0.4791</td>
<td>0.4935</td>
<td>0.4970 (optimum)</td>
<td>0.4872</td>
<td>0.4661</td>
</tr>
<tr>
<td>8.50</td>
<td>0.4733</td>
<td>0.4899</td>
<td>0.4961</td>
<td>0.4882</td>
<td>0.4678</td>
</tr>
</tbody>
</table>

For the NL-3 site, which is used as a reference location in the DOWEC project, the following ambient conditions have been defined (Bierbooms, ref. [3]): Weibull shape parameter (k) = 2.1 and average wind speed ($V_{ave}$) = 9.2 m/s. The corresponding energy yield calculated with PHATAS is 25.317 GWh. This includes the effect of blade pitch control and turbulence but does not include array losses or losses due to yaw misalignment or downtime. Figure 9 shows the energy yield and capacity factor as function of annual average wind speed at hub height.

Figure 9: annual gross energy yield and turbine capacity factor.
3. ROTOR STRUCTURAL PROPERTIES

This chapter contains a description of the mass and stiffness distribution of the LMH64-5 rotor blade in the PHATAS turbine model based on specifications from LM Glasfiber Holland. It also includes an evaluation of the dynamic properties of the rotor.

3.1 Blade mass and stiffness distribution

For the description of the blade mass distribution the following criteria were considered to be most important:
- particularly describing the specified mass distribution of the outside blade part well;
- matching the static mass moment, the blade moment of inertia and the total mass.

The modelling of the bending stiffness distribution was more focused on:
- a good description of the specified stiffnesses at the inner part of the blade;
- a good description of the cross stiffness to model the direction of the eigen modes, and as such a correct aero elastic behaviour of the blade.

A verification of the description in PHATAS is given in Figure 10 to Figure 12 and Table 4.

![Figure 10: cumulative blade mass distribution.](file://table.frb)

LMH64-5: cumulative mass distribution
Figure 11: verification of blade flatwise stiffness w.r.t. local chord axes reference system.

Figure 12: verification of blade edgewise stiffness w.r.t. local chord axes reference system.
3.2 Mass imbalance

Following directives given by NEG Micon Holland based upon discussions they had earlier with LM Glasfiber Holland, the mass imbalance is specified as a variation in blade mass of \( f = 0.5\% \). Therefore the mass and static moment for blade 1, 2, and 3 relative to the reference values are 100\%, 100.5\%, and 99.5\% respectively.

(S) Represents the static moment. The distance of the rotor centre of gravity from its centre (not including hub mass) is now calculated as:

\[
    r_{cog\ rotor} = \frac{S_2 \cos 30 - S_3 \cos 30}{m_1 + m_2 + m_3} = \frac{S_1 (1 + f) \cdot \sqrt{3} - S_1 (1 - f) \cdot \sqrt{3}}{m_1 (1 + (1 + f) + (1 - f))} = \frac{f \cdot S_1}{\sqrt{3} \cdot m_1}
\]

For the pre-design it is found with PHATAS that \( S_1 = 334.96 \cdot 10^3 \) kgm and \( m_1 = 17648 \) kg, hence \((r_{cog\ rotor}) = 0.055 \) m. Relative to the rotor span this is:

\[
    \frac{r_{cog\ rotor}}{R} = \frac{0.055 \ m}{64.5 \ m} \cdot 100\% = 0.0849\%
\]

It is interesting to see that this value is almost 6 times smaller than the radial displacement of 0.5\% for "balanced rotors" prescribed by Germanischer Lloyd in chapter 4-4-3 of ref. [7].

3.3 Natural rotor frequencies

The structural properties and first blade natural frequencies calculated with PHATAS are compared in Table 4 with results from LM Glasfiber Holland derived with the tool FOCUS and with another ECN tool called BLADMODE.

The static mass moment for the FOCUS result is calculated by taking the centre of mass for each subsection in the middle of the blade element, which yields a small over-prediction compared to the PHATAS result.
Table 4: structural and dynamic blade properties.

<table>
<thead>
<tr>
<th></th>
<th>FOCUS 0 rpm</th>
<th>PHATAS 0 rpm</th>
<th>FOCUS 11.0 rpm</th>
<th>PHATAS 11.8 rpm (rated speed)</th>
<th>BLADMODE 11.8 rpm (rated speed)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; flatwise frequency</td>
<td>0.652 Hz</td>
<td>0.621 Hz</td>
<td>0.702 Hz</td>
<td>0.675 Hz</td>
<td>0.683 Hz</td>
</tr>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; edgewise frequency</td>
<td>1.117 Hz</td>
<td>1.099 Hz</td>
<td>1.140 Hz</td>
<td>1.107 Hz</td>
<td></td>
</tr>
<tr>
<td>Blade mass</td>
<td>17648 kg</td>
<td>17648 kg</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Static moment (w.r.t. rotor centre)</td>
<td>369.71 · 10&lt;sup&gt;3&lt;/sup&gt; kgm</td>
<td>366.72 · 10&lt;sup&gt;3&lt;/sup&gt; kgm</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

The rotating blade frequencies for the reference blade calculated with PHATAS are 0.675 Hz (3.4P) for flap and 1.107 Hz (5.6P) for the reaction-less edgewise frequency<sup>7</sup>. The second flap wise and reaction-less edgewise frequency are 1.80 Hz and 3.99 Hz respectively. The collective edgewise frequency coupling with torsion, is 1.25 Hz. A more elaborate analysis on this property is given on page 36.

Both blade torsion and shaft bending are not modelled as a degree of freedom because no specifications are available (yet).

![Figure 14: rotating blade natural bending modes (PHATAS model file).](image)

3.4 Structural and numerical damping

**Rotor**

After discussions with LM Glasfiber Holland the structural damping of flapwise and leadwise blade bending are given the same values as were used in the NM3000 analysis, ref. [17]; logarithmic decrement (δ) = 3 %. The PHATAS input values are expressed as fractions of critical damping (ζ = c / c<sub>c</sub> = δ / (2 π)).

---

<sup>6</sup> Here the first edgewise frequency is the reaction-less rotor frequency, sometimes also referred to as symmetric in plane blade frequency.

<sup>7</sup> The collective edgewise frequency is sometimes also referred to as asymmetric in plane frequency. In this mode all three blades are in phase. The frequency depends on drive train inertia and the stiffness of drive train and generator.
Support structure

The structural damping of the support structure, i.e. tower and the part of the mono pile above the seabed is assumed to have a damping of 1% of critical damping. Some verification on this can be found in the thesis of Martin Kühn, ref. [15], and in ref. [2]. These sources give values of 0.6% to 1.0% (fraction of critical damping) for bolted steel chimneys. Soil damping is neglected in the calculations. According to ref. [15] this is permissible, with exception of an extreme situation, e.g. an extreme operational gust in combination with an emergency shut-down of the turbine.

Numerical damping

No correction has been applied for additional numerical damping in the PHATAS calculations. The consequence of this omission can be estimated using the following formula from ref. [18]:

\[ \zeta_{\text{numerical}} = 0.5 \cdot (\nu \cdot \Delta t)^3 \]

\( \Delta t \) [s] is the time increment used in the calculations and the frequency \( \nu \) is expressed in [rad/s]. A time increment of 0.02 s in the PHATAS calculations seems appropriate. For example, all dynamics with a frequency \( \nu > 2.16 \text{ Hz} \) have in that case a numerical damping \( \zeta_{\text{num}} \geq 1\% \). 2.16 Hz is more than the 2nd flapwise bending frequency and also more than the first reactionless and collective edgewise frequency.
4. TURBINE CHARACTERISTICS

In this chapter an explanation to the modelling of the turbine properties is given. Most information comes from NEG Micon Holland, see ref. [8] to [11].

4.1 Generator properties

Rated tip speed 80 m/s
Nominal rotor speed slow shaft 80 m/s / 64.5 m = 1.24 rad/s = 11.844 rpm
Synchronous nominal generator speed 1000 rpm
Gearbox ratio, defined at 110% of synchronous generator speed (rated) 1100 rpm / 11.844 rpm = 92.873
Inverter range 70% < synchronous speed < 130%
Minimum rotor speed 1000 rpm · (1 - 0.3) / 92.873 = 7.54 rpm
Maximum rotor speed 1000 rpm · (1 + 0.3) / 92.873 = 14.00 rpm
Nominal generator torque 6 MWe / (1100 · π /30) = 52.087 kNm
Time lag constant of generator characteristic < 0.1 s (ref. [9]). Refer page 36 for more.
Speed variation below lambda-optimum range 1%
Speed variation above lambda-optimum range 10% (id. NM3600), viz. at fast shaft:
52.087 kNm / (0.1 · 1000 rpm) = 521 Nm / rpm

The asynchronous generator behaviour when reaching maximum rotor speed is described by a 1% increase in rotor speed for a doubling in power.

4.2 Power conversion losses

Conversion losses consist of gearbox losses and generator losses. Following is a summary of a more detailed specification given in ref. [8].

The specified generator efficiency is defined as a tabulated relation with the relative net power. Gearbox losses consist of a power dependent part (80% at rated) and a speed dependent part (20% at rated):

\[ P_{\text{loss (gearbox)}} = 154 \text{ kW} \times (P_{\text{aero}} / P_{\text{aero, nom}}) + 38 \text{ kW} \times (\Omega / \Omega_{\text{nom}})^3. \]

A constant loss at the slow speed shaft and a part proportional to the fast shaft torque describe the power losses in PHATAS:

\[ P_{\text{loss (PHATAS)}} = Q_{\text{loss, slow}} \cdot \Omega_{\text{slow}} + \eta_{\text{prop}} \cdot Q_{\text{fast}} \cdot \Omega_{\text{fast}}. \]

This relation for the loss of power differs from the specifications. A good description gives: \( Q_{\text{loss}} = 45.1 \text{ kNm} \) and \( \eta_{\text{prop}} = 0.048 \), see Figure 15.

\[^8\] \( \frac{dQ}{d\Omega} = \frac{(12000 \text{ kW} / 1.01 - 6000 \text{ kW})}{(14.0 \text{ rpm} \cdot \pi / 30 \cdot 92.873) / (0.01 \cdot 14.0 \text{ rpm})} = 308.5 \text{ kNm} / \text{rpm}. \)
4.3 Generator curve

Combining the properties of the synchronous generator mentioned in chapter 4.1 with the available power as provided by the stationary rotor characteristics (chapter 2.4) yields the torque versus rotor speed curve (Q-Ω) given in Table 5 and shown in Figure 16. The 10% slip between the optimum lambda range and rated is the same as for the NM3600 turbine. The slip at low wind speeds between cut in value and the point where optimum lambda starts is 5%. Optimum lambda here begins at approximately $V = 6.7$ m/s wind speed. The effect of the amount of slip towards optimum lambda on the energy capture is small.
Table 5: Q-Ω relation DOWEC 6 MW pre-design

<table>
<thead>
<tr>
<th>rotor speed [rpm]</th>
<th>generator torque [Nm]</th>
<th>rotor speed [rpm]</th>
<th>generator torque [Nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.531</td>
<td>0.00</td>
<td>11.800</td>
<td>49954.28</td>
</tr>
<tr>
<td>7.537</td>
<td>536.04</td>
<td>11.844</td>
<td>52087.07</td>
</tr>
<tr>
<td>7.557</td>
<td>2419.49</td>
<td>12.000</td>
<td>51410.33</td>
</tr>
<tr>
<td>7.591</td>
<td>5643.82</td>
<td>12.400</td>
<td>49751.93</td>
</tr>
<tr>
<td>7.681</td>
<td>14316.27</td>
<td>12.800</td>
<td>48197.18</td>
</tr>
<tr>
<td>8.032</td>
<td>15757.52</td>
<td>13.400</td>
<td>46039.10</td>
</tr>
<tr>
<td>9.179</td>
<td>20896.50</td>
<td>13.500</td>
<td>45698.07</td>
</tr>
<tr>
<td>10.327</td>
<td>26806.91</td>
<td>13.998</td>
<td>44073.68</td>
</tr>
<tr>
<td>11.359</td>
<td>32720.10</td>
<td>14.138</td>
<td>87274.61</td>
</tr>
<tr>
<td>11.500</td>
<td>35441.78</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 17: net power as function of wind speed (stationary calculations).

Figure 18: power coefficient and pitch angle versus wind speed (stationary calculations).
It is assumed here that pitch control is applied between cut in and the wind speed above which optimum lambda can be applied. Pitch to vane already begins somewhat below rated wind speed. This increases rated wind speed but reduces the maximum axial thrust. This kind of blade pitch control to make the tower top loading more benign is called “thrust clipping”. The dotted line in Figure 19 qualitatively shows the axial thrust if the pitch angle in partial load would be kept constant up to rated wind speed.

4.4 Turbine geometry

<table>
<thead>
<tr>
<th>Table 6: turbine geometry</th>
</tr>
</thead>
<tbody>
<tr>
<td>cone angle</td>
</tr>
<tr>
<td>rotor tilt angle</td>
</tr>
<tr>
<td>rotor centre w.r.t. tower centre, horizontal</td>
</tr>
<tr>
<td>rotor centre w.r.t. tower centre, vertical</td>
</tr>
<tr>
<td>nacelle height</td>
</tr>
<tr>
<td>nacelle side area for aerodynamic drag</td>
</tr>
<tr>
<td>nacelle front area for aerodynamic drag</td>
</tr>
<tr>
<td>radial distance pitch bearing</td>
</tr>
<tr>
<td>radial distance blade - hub connection</td>
</tr>
<tr>
<td>position main bearing aft of rotor centre</td>
</tr>
<tr>
<td>yaw bearing above tower top</td>
</tr>
</tbody>
</table>

4.5 Turbine properties

<table>
<thead>
<tr>
<th>Table 7: turbine mass properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>hub mass (including pitch bearings)</td>
</tr>
<tr>
<td>hub rotating moment of inertia</td>
</tr>
<tr>
<td>nacelle mass</td>
</tr>
<tr>
<td>(incl. generator, without rotor and hub)</td>
</tr>
</tbody>
</table>
Table 8: drive train properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Rotating moment of inertia</th>
<th>Torsional stiffness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low speed shaft</td>
<td>5495 kgm²</td>
<td>3.29 · 10⁶ Nm / rad</td>
</tr>
<tr>
<td>Transmission</td>
<td>6 kgm²</td>
<td>8.12 · 10⁶ Nm / rad</td>
</tr>
<tr>
<td>Brake</td>
<td>19 kgm²</td>
<td>~</td>
</tr>
<tr>
<td>High speed shaft</td>
<td>24 kgm²</td>
<td>2.7 · 10⁶ Nm / rad</td>
</tr>
<tr>
<td>Drive train transmission ratio</td>
<td>92.873</td>
<td></td>
</tr>
</tbody>
</table>

Shaft torsion flexibility is currently not modelled as a degree of freedom. The asymmetric in plane or collective mode is discussed in chapter 6.2, page 36.

4.6 Yaw control

Yaw control is not modelled as a degree of freedom for the first design calculations. The parameters are as follows:

- Net yaw power: 6 electrically powered units of each 2 kW = 12 kW
- Synchronous rotor speed: 1500 rpm
- Nominal slip: 0.067
- Nominal yaw speed: 1400 rpm / 0.05 rpm = 28000
- Ratio yaw ring: 1/12
- Motor 'kip torque': ~ 2.5 times nominal torque.
- Yaw motor inertia: \( I_p \) (yaw motor) = 0.005 kgm²
- Effective yaw inertia: 6 · 0.005 kgm² · (28000)² = 23.52 · 10⁶ kgm²
- First yaw misalignment trigger: \( \Delta \varphi_{yaw1} \) (30 seconds average) > 10°
- Second yaw misalignment trigger: \( \Delta \varphi_{yaw2} \) (2 seconds average) > 30°
- Stop criterion yaw action: \( \Delta \varphi_{yaw} \) (2 seconds average) < 0.5°
- Efficiency yaw motors: 0.90
- Efficiency total transmission: 0.92 (similar to NM3600 turbine)
- Nominal yaw torque:
  \[
  P_{nom} / \omega_{low} \cdot i \cdot \eta = 12 \cdot 10^3 / (1400 \cdot \pi / 30) \cdot 28000 \cdot 0.88 = 2017 \text{kNm}
  \]

An active brake is envisaged using 9 (maybe one or two more) brake systems of around 200 kN net braking force located along the yaw bearing perimeter. The brake is released when yaw action takes place. The braking torque is estimated at:

\[
M_{brake} = 9 \cdot 200 \text{kN} \cdot (\text{yaw bearing radius}) = 9 \cdot 200 \text{kN} \cdot 1.9 \text{m} = 3.42 \text{MNm}
\]

The number of brake systems is limited by the available space.

The electric mechanical braking torque of the yaw motors will be determined after parameter sensitivity studies with PHATAS. A value of around 2 times nominal value is expected.
Objective in this respect is to virtually prevent yaw motion due to strong asymmetric wind loading at normal production at 16 m/s by using the reaction force from both the electrical mechanical brake and the separate brakes. On the whole yawing action should take place at about 10% of the operational time or less.

The yaw misalignment is determined on the basis of a running average calculation. Damping and bearing friction of the yaw system are preliminary neglected.

4.7 Rotor-speed control

By application of power electronics the 6 MW turbine has a stationary constant power characteristic above the rated rotor speed of 11.844 rpm. Therefore, the destabilising effect of the generator torque at full load operation has to be stabilised by rotor speed feedback control, which sets the pitch angle to such a value that the rotor speed is kept within a small range.

Below rated speed, the pitch angle is kept almost constant (value of maximum power) and the generator torque is set to a value which corresponds with the design tip speed ratio in partial load.

A more detailed description of the rotor-speed controller is given by E.L. van der Hooft in ref. [12]. File references for the control algorithm are given in Appendix B.

4.7.1 Pitch control

In the DOWEC project, a pitch control algorithm has been developed to keep the rotor speed between its rated value and the maximum allowable generator speed of 14 rpm during full load operation. To achieve this the low pass filtered rotor speed is fed back to a PD-controller that determines a set point value of the pitching speed that stabilises the rotor speed.

Due to the heavily non-linear character of the (turbine) aerodynamics some non-linear extensions are added to meet the satisfied performance:

- linear controller gains have been scheduled dependent on the current blade angle, with gain scheduling the linear controller is adapted to the whole operational envelope of the wind turbine;
- an (scheduled) inactivity zone with hysteresis, to avoid undesired pitch angle adjustments due to small controller corrections caused by noise, tower shadow, rotational sampling effects, etc.;
- rotor speed set point adaptation at a higher wind speed level, which supports the linear controller to avoid energy loss in case of a sudden drop in wind speed (flywheel effect);
- rotor speed limitation, which forces the pitch angle quickly to vane (+4 °/s) - be it for a short period of time as possible - as soon as the current rotor speed exceeds a certain safety level (here 13.5 rpm) and is still accelerating;
- pitching bounds are incorporated to limit the calculated pitch speed values;
- non-linear compensation on the calculated pitching speed to compensate amplification due to 'dynamic inflow effects' (rotor wake effects).

Additional optimisation is incorporated by a mechanism called 'estimated wind speed feed forward' which adds a non-linear control action to the pitching speed set point based on the reconstructed value of the rotor effective wind speed. This reconstruction is based on measured values of rotor speed, pitch angle, electric power and theoretical aerodynamic properties of the rotor. 'Estimated wind speed feed forward' results in a higher energy yield, less rotor speed fluctuations and smaller pitch actions.

A switching hysteresis is implemented to limit the number of transitions between full load operation and partial load operation.
4.7.2 Torque control

Reduction of rotor speed variations is attractive to limit the electrical dimensions of the converter. Knowing that both converter and generator are able to stand overloading as long as their thermal limitations are not exceeded, the electrical system is used to react quickly on rotor speed variations. Therefore the generator torque set point consists of a stationary value to produce rated power and an additive dynamic set point to reduce rotor speed variations. To ensure excessive power or torque values the fluctuations are limited to 125% of their rated values. Using dynamic torque control (together with pitch control) leads to rotor speed variations of approximately +/- 0.5 rpm.

4.8 Turbine control

The following preliminary specifications apply for the general control of the turbine. More information on the turbine control characteristics will be included in a load set description document.

- pitch rate for start up: -2 °/s
- delay time for pitch brake: 0.3 s to activate, 0.6 s to reach maximum brake torque
- brake torque fast shaft: 34 kNm
- generator cut off dependent on rotor speed:
  i.) if \( \Omega < \Omega_{\text{start}} / 1.04 = 7.54 \text{ rpm} / 1.04 = 7.25 \text{ rpm} \).
  ii.) if \( \Omega > 13.5 \text{ rpm} \) (upper limit for blade pitch control, inverter boundary is 14.0 rpm, see chapter 4.1).

---

9 Specifications from NEG Micon Holland are an exponential increase in brake torque with a period of 1 second to reach full brake torque. PHATAS models a linear increase in brake torque.
5. SUPPORT STRUCTURE

This chapter presents the characteristics of the pre-design of the support structure, consisting of an 80 m high tower (NEG Micon) and 75 m long mono pile foundation (Ballast Nedam). For the clamping stiffness at the seabed 4 different levels of the sea bottom were investigated varying between 21 m and 36 m w.r.t. mean sea level (MSL). For the baseline study scour protection is envisaged and only the 21 m MSL option is used for preliminary load calculations.

5.1 Tower design

NEG Micon Holland defined the tower geometry. The tubular shaped tower is assumed to have a linear diameter and thickness distribution. This is a necessary simplification because more detailed (non-linear) specifications of the eventual structure are not available yet. Also no flanges were modelled\(^{10}\). The base diameter of 6 m is similar to the diameter of the connecting mono pile foundation.

<table>
<thead>
<tr>
<th>Table 9: tower properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>tower height</td>
</tr>
<tr>
<td>tower base outer diameter</td>
</tr>
<tr>
<td>tower base wall thickness</td>
</tr>
<tr>
<td>tower top outer diameter</td>
</tr>
<tr>
<td>tower top wall thickness</td>
</tr>
<tr>
<td>tower mass</td>
</tr>
<tr>
<td>steel density (effective)</td>
</tr>
<tr>
<td>Young's modulus (E)</td>
</tr>
<tr>
<td>shear modulus (G)</td>
</tr>
</tbody>
</table>

5.2 Foundation design

<table>
<thead>
<tr>
<th>Table 10: mono pile properties (Ballast Nedam)</th>
</tr>
</thead>
<tbody>
<tr>
<td>pile length</td>
</tr>
<tr>
<td>pile diameter</td>
</tr>
<tr>
<td>pile wall thickness</td>
</tr>
<tr>
<td>mono pile weight</td>
</tr>
<tr>
<td>additional water depths investigated</td>
</tr>
<tr>
<td>distance in seabed investigated</td>
</tr>
</tbody>
</table>

For the DOWEC design scour protection is considered. Ballast Nedam assumed for their analysis of the prismatic mono pile foundation that there would be no scour protection, ref. [22]. Consequently the inclination stiffness was investigated for different water depths: 21 m w.r.t. MSL (no erosion), 26 m water depth, 31 m, and 36 m water depth (i.e. 15 m erosion of the sea bottom surrounding the pile). The total length of the foundation is for all cases 75 m and the corresponding depth of the pile in the seabed varies between 30 m (minimum) and 45 m (baseline). The dynamic behaviour for the different states of seabed erosion is here reported.

\(^{10}\) Because tower flanges and miscellaneous components are not modelled a tower material density of 8500 kg/m\(^3\) is chosen instead of a density of 7850 kg/m\(^3\) which is the value for steel. The effect of this extra mass on derived natural frequencies is approximately 1%.
However, only the 21 m MSL option is researched for the first design load calculations because the other options have a structure bending frequency that is expected to be unallowably close to the rotor passing frequency, see Figure 20 on page 34.

For the structural modelling of the foundation the same material properties were used as for the tower, except for the density, here 7850 kg/m$^3$. No additional mass was used to account for the situations where a scour hole around the tower would occur and the inside of the tower would be filled with sand and/or water. Also the influence of a transition piece on top of the pile was neglected in the analysis, as no information was available to date. The foundation plus tower properties are tabulated in Table 10.

'Location 1' near NESS grid point NL7 was used as a reference for the geo-technical information. Data for the optional NL3 grid point (Location 2) were far too poor according to Ballast Nedam. It is noticed that the mean sea level of 21 m used by Ballast Nedam for location 1 differs from the value of 27.5 m given in the terms of reference, ref. [10].

For the water depth of 21 m the foundation protrudes 9 m above mean sea level (CD+10). The minimum distance between the rotor tip and MSL is then:

\[
\text{b.t.c. (vertical)} = 9 \, \text{m} + (\text{tower height}) + (\text{hub height}) - (\text{rotor radius}) = 9 \, \text{m} + 80 \, \text{m} + 2.4 \, \text{m} - 64.5 \, \text{m} = 26.9 \, \text{m}.
\]

It must be checked that the minimum tip height is not over-conservative but at least sufficient to prevent intolerable erosion due to extreme high rising splash zones.

The flexibility at the intersection of the mono pile and the seabed level is derived from ref. [22, appendix F]. NEG Micon Holland provided the design loading conditions at seabed level to Ballast Nedam. From this a design graph was made with a linear relation between (CD-25, M=24.6 MNm) and (CD+20, M=0). The design conditions ('service limit state') for the investigation of soil flexibility were then defined at CD+10 (pile top level) as M = 54.7 MNm and F = 5470 kN (independent of height). The following relation applies to the PHATAS foundation model:

\[
\begin{bmatrix}
\varphi_{\text{base}} \\
F_{\text{base}}
\end{bmatrix} = \begin{bmatrix}
\frac{1}{k_u} & \frac{f}{k_p} \\
\frac{f}{k_p} & \frac{1}{k_u}
\end{bmatrix} \begin{bmatrix}
M_{\text{base}} \\
F_{\text{base}}
\end{bmatrix}
\]

It is here assumed that the influence of the shear force ($F_{\text{base}}$) is negligible. Table 11 summarises the design conditions at different seabed levels corresponding to an increase in water depth between 0 m and 15 m which may be due to e.g. scour or sand waves (troughs) around the pile. Depending on the sea bottom level there are 4 different input files for the support structure using different foundation flexibilities ($k_u$) and (f). Preliminary structure bending frequencies calculated with PHATAS have been verified by the project partners TUD and Stentec. Based upon the result of this short study reported in ref. 15, it was decided to use the clamping properties given in bold print in Table 11 as reference.

<table>
<thead>
<tr>
<th>decrease in seabed [m]</th>
<th>pile length above seabed [m]</th>
<th>$f$ [m/rad]</th>
<th>$k_u$ [Nm/rad]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>30</td>
<td>0.0</td>
<td>3.705E+10</td>
</tr>
<tr>
<td>0</td>
<td>30</td>
<td>14.58</td>
<td>3.282E+10</td>
</tr>
<tr>
<td>5</td>
<td>35</td>
<td>14.13</td>
<td>3.509E+10</td>
</tr>
<tr>
<td>10</td>
<td>40</td>
<td>12.76</td>
<td>3.799E+10</td>
</tr>
<tr>
<td>15</td>
<td>45</td>
<td>10.41</td>
<td>4.424E+10</td>
</tr>
</tbody>
</table>
The input specification for the structural properties of tower and foundation are given in Table 12. The horizontal line in the table indicates the intersection between the mono pile and the tower. The last line in the table is used to describe the mass of the yaw bearing on top of the tower.

<table>
<thead>
<tr>
<th>height above sea bottom [m]</th>
<th>diameter section base [m]</th>
<th>diameter section top [m]</th>
<th>thickness [m]</th>
<th>tensile stiffness [N/m²]</th>
<th>shear stiffness [N/m²]</th>
<th>density [kg/m³]</th>
<th>chart datum</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.00</td>
<td>6.000</td>
<td>6.000</td>
<td>0.060</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>7850</td>
<td>-20.0</td>
</tr>
<tr>
<td>6.00</td>
<td>6.000</td>
<td>6.000</td>
<td>0.060</td>
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<td>8.08E+10</td>
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<td>-17.0</td>
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</tr>
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<td>6.000</td>
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<td>8.08E+10</td>
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<td>-11.0</td>
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<td>6.000</td>
<td>0.060</td>
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<td>8.08E+10</td>
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<td>-8.0</td>
</tr>
<tr>
<td>18.00</td>
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<td>0.060</td>
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<td>-5.0</td>
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<tr>
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<td>0.060</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>7850</td>
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</tr>
<tr>
<td>34.00</td>
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<td>5.894</td>
<td>0.0268</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>14.0</td>
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<tr>
<td>38.00</td>
<td>5.894</td>
<td>5.787</td>
<td>0.0264</td>
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<td>8.08E+10</td>
<td>8500</td>
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<tr>
<td>42.00</td>
<td>5.787</td>
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<td>8500</td>
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<td>0.0256</td>
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<tr>
<td>50.00</td>
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<td>5.468</td>
<td>0.0252</td>
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<td>8500</td>
<td>30.0</td>
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<td>5.361</td>
<td>0.0247</td>
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<td>58.00</td>
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<td>0.0243</td>
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<td>8.08E+10</td>
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<td>38.0</td>
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<tr>
<td>62.00</td>
<td>5.255</td>
<td>5.148</td>
<td>0.0239</td>
<td>2.10E+11</td>
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<tr>
<td>66.00</td>
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<td>5.042</td>
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<td>4.935</td>
<td>0.0231</td>
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<td>8.08E+10</td>
<td>8500</td>
<td>50.0</td>
</tr>
<tr>
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<td>4.935</td>
<td>4.829</td>
<td>0.0227</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>54.0</td>
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<tr>
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<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>58.0</td>
</tr>
<tr>
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<td>4.722</td>
<td>4.616</td>
<td>0.0219</td>
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<td>8.08E+10</td>
<td>8500</td>
<td>62.0</td>
</tr>
<tr>
<td>86.00</td>
<td>4.616</td>
<td>4.509</td>
<td>0.0215</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>66.0</td>
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<tr>
<td>90.00</td>
<td>4.509</td>
<td>4.403</td>
<td>0.0211</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>70.0</td>
</tr>
<tr>
<td>94.00</td>
<td>4.403</td>
<td>4.296</td>
<td>0.0206</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>74.0</td>
</tr>
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<td>4.296</td>
<td>4.190</td>
<td>0.0202</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>78.0</td>
</tr>
<tr>
<td>102.00</td>
<td>4.190</td>
<td>4.083</td>
<td>0.0198</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>82.0</td>
</tr>
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<td>4.083</td>
<td>3.977</td>
<td>0.0194</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>86.0</td>
</tr>
<tr>
<td>110.00</td>
<td>3.977</td>
<td>3.870</td>
<td>0.0190</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>8500</td>
<td>90.0</td>
</tr>
<tr>
<td>110.20</td>
<td>3.870</td>
<td>3.870</td>
<td>0.42</td>
<td>2.10E+11</td>
<td>8.08E+10</td>
<td>7862</td>
<td>90.2</td>
</tr>
</tbody>
</table>

Table 12: PHATAS input format for support structure, here for MSL = 21 m.

5.3 Dynamic characteristics of turbine including support structure

Table 13 contains the calculated natural frequencies of the entire construction at different states of seabed erosion. The margin between the calculated first structure bending frequency and the rotor passing frequency at nominal speed seems sufficient: 0.242 Hz / (11.844 rpm / 60 s) = 1.23P. The structure bending frequency also lies above the maximum rotor speed in full load of 13.5 rpm (0.225 Hz).

Resonance in deeper water is however possible, as is shown in Figure 20. It must be noticed that of the coupled modes having side ways bending and torsion, the first mode (predominantly bending) is marginally damped by rotor aerodynamics compared to the fore aft motion. This makes it for stability analysis more important than the fore aft mode.
Table 13: support natural frequencies (values derived with PHATAS)

<table>
<thead>
<tr>
<th>decrease in seabed level [m]</th>
<th>ν [Hz] fore aft 1\textsuperscript{st}</th>
<th>ν [Hz] fore aft 2\textsuperscript{nd}</th>
<th>ν [Hz] coupled, idling\textsuperscript{11}</th>
<th>ν [Hz] coupled, idling 2\textsuperscript{nd} (tortion + side)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.242</td>
<td>1.429</td>
<td>0.244</td>
<td>1.384</td>
</tr>
<tr>
<td>5</td>
<td>0.231</td>
<td>1.288</td>
<td>0.233</td>
<td>1.376</td>
</tr>
<tr>
<td>10</td>
<td>0.221</td>
<td>1.156</td>
<td>0.223</td>
<td>1.369</td>
</tr>
<tr>
<td>15</td>
<td>0.211</td>
<td>1.036</td>
<td>0.212</td>
<td>1.361</td>
</tr>
</tbody>
</table>

**DOWEC 6 MW: variation in lowest natural bending frequency**

![Graph showing the variation in lowest natural bending frequency as a function of water depth.](image)

Figure 20: 1\textsuperscript{st} sideways bending frequency as function of water depth surrounding the pile.

\textsuperscript{11} The natural frequencies are determined with PHATAS using the idling state of the rotor because this represents the dynamic condition of a variable speed machine better than a 'fixed rotor state'.
6. TURBINE FREQUENCY ANALYSIS

This chapter presents an investigation of the different natural frequencies of the turbine and support structure.

6.1 Campbell diagram

Figure 21 shows a selection of natural frequencies as function of rotor speed. The sideways bending frequencies of the construction depend on the water depth, see Table 13 at page 34 for the exact values. In fact these are coupled modes with predominantly bending for the first mode and predominantly torsion for the second mode. The fore-aft bending modes have virtually the same bending frequencies but are less important because they are aerodynamically well damped.

Figure 21: 'Campbell diagram' with significant natural frequencies.
Intersections of frequencies in the Campbell diagram worth notifying are:

~ 1.11 Hz: Reaction-less edgewise frequency and 6P.
This deserves attention because the edgewise mode is marginally
aerodynamically damped, especially around 11 rpm, just below nominal speed.
At this point pitch angles are limited and variations in rotor speed are small
because of the steep generator torque characteristics.
To a lesser amount the crossing of this frequency with 5P and 7P also
contributes to the fatigue loading intensity.

~ 1.25 Hz: Collective edgewise frequency and 6P.
As mentioned above the edgewise mode is in principle marginally
aerodynamically damped. Moreover, in this case above rated wind speed the
negative slope of the Q-Ω curve implies negative damping from the drive train
on the in-plane collective rotor blade motions.

~ 0.22 Hz Tower bending and 1P. According to the current specifications (with scour
protection) the margin between the calculated first structure bending frequency
and the rotor passing frequency is sufficient. However:
- If scour protection is omitted - as in the assessment by Ballast Nedam -
  variations in sea bottom level of more than 10 m need to be accounted for.
  This reduces the effective stiffness of the support structure significantly,
  leading to resonance for water depths of more than 30 m. The same
  accounts for larger water depths due to variations in water level in the wind
  farm area or due to sand waves.
- The properties that determine the structure bending frequency such as the
  static moment of the turbine, soil properties, scour, sand waves, and water depth,
  need to be kept in check as the risk of resonance with the nominal
  rotor passing frequency looms if the bending frequency should decrease.

6.2 Collective edgewise bending mode

The collective edgewise bending mode of the rotor blades couples with the shaft and drive train.
Its value is calculated with the tool BLADMODE of ECN as 1.254 Hz (6.4P). Torsion
flexibility of the shaft is hereby taken into account. When excluding axis torsion the frequency
is approximately 2.0 Hz. In these calculations are the inertia of the generator rotor, fast shaft
brake and fast shaft related to the slow shaft. This is permissible because the slow shaft is
relatively more flexible than the fast shaft. This is explained in the following equation where
\( \sqrt{\frac{k}{I}} \) is proportional to the torsion frequency, \( k \) represents torsion stiffness, \( i \) is the gearbox
ratio, and \( J \) stands for inertia.

\[
\sqrt{\frac{k_{\text{slow shaft}}}{J_{\text{slow}} + J_{\text{gear box}} + (J_{\text{brake}} + J_{\text{HSS}} + J_{\text{generator rotor}}) \cdot i^2}} < \sqrt{\frac{k_{\text{fast shaft}}}{J_{\text{brake}} + J_{\text{HSS}} + J_{\text{generator rotor}}}}
\]

The moments of inertia of the drive train are given in Table 8. The denominator in the left-handed equation above is 5.025 \( \cdot 10^6 \) kgm

For comparison; the rotor rotational inertia is 36.34 \( \cdot 10^6 \) kgm

Note: Rotor torque variations from the rotor, e.g. due to the collective edgewise bending, have
a more or less asynchronous behaviour depending on the time lag for the control of
generator torque (\( \tau \)). Above rated wind speed where \( dQ/d\Omega \) is negative, the delay from
the generator torque induces damping, increasing with (\( \tau \)). NEG Micon Holland
estimated \( \tau = 0.01 \) s. This damping is however not accounted for in PHATAS.
7. MODELLING OF EXTERNAL LOADING

This chapter describes the modelling of wind and wave loading used for the computer calculations. Input files for PHATAS for these two sources of (stochastic) external loading are generated with the ECN programs SWIFT and ROWS respectively.

7.1 Modelling of wind conditions

The wind loading for the calculations is generated with the ECN program SWIFT (Simulation of Wind Fields in Time). The 3 component turbulent wind is described for a grid with:
- 15 points in radial direction (equal to number of blade elements);
- 64 points in circumferential direction;
- 8192 points in longitudinal (time-) direction.

The circles of the grid match with the blade element centres. The inner circle coincides with the 2nd blade elements and the most outside ring of the wind field lies half a blade element length away from the circle described by the blade tip. This outer circle is used for interpolation in the SWIFT wind grid data in case of (strong) sideways tower deformation.

With 64 points in circumferential direction, the 'cut-off frequency’ for turbulence crossing is 32P. For the interpolation in circumferential direction used in PHATAS it has been shown that dynamics up to 60% of this cut-off frequency (19.2P) are represented for at least 90%. Hence the third flapwise frequency (20.0P) and the second symmetric lead lag frequency (20.2P) are excited marginally. The asymmetric lead lag frequency (10.3P) is excited well.

For the 0.10 s time-interval of the stochastic wind files the cut-off frequency in 'time direction’ is 5.0 Hz (25.2P) while the total length is 819.2 s.

In 'time direction’ PHATAS uses the same interpolation as in circumferential direction, which means that frequencies up to 60% of the cut off frequency (15.1P) are represented for at least 90%. This is sufficient for excitation of the 2nd flapwise mode (9.6P) and the reaction-less edgewise bending mode, but not entirely for the 2nd reaction-less edgewise mode (20.2P), which is however acceptable.

The characteristic turbulence level as function of wind speed has been discussed by J. Peeringa (ECN, internal note, March 2002). This information will be included in an update of ref. [3].

7.2 Modelling of wave conditions

The time-dependent hydrodynamic forces acting on the structure are obtained through the use of linear wave theory, Morison’s equation and constant drag and inertia coefficients.

Wave kinematics are simulated with the ECN program ROWS (Random Ocean Wave Simulator), which is based on Airy theory. The 2-parameter JONSWAP spectrum with as input parameters significant wave height $H_s$ and the mean zero up-crossing period $T_c$, describes the sea state.
Hc is defined as the average of the highest 1/3-fraction of the waves and Tz is defined as

\[ T_z = 2\pi \sqrt{\frac{m_0}{m_2}} \cdot \frac{1}{m_2} \left[ \frac{\int_{0}^{\infty} S(\omega) d\omega}{\int_{0}^{\infty} \omega^2 S(\omega) d\omega} \right]^{0.5} \]

The three parameters (γ, σa, and σb) of the so-called peak enhancement factor are assumed to be constant and set to their average values, i.e. γ = 3.3, σa = 0.07 and σb = 0.09.

Possible combinations of Hc and Tz are taken from a 3D-scatter diagram (Hc, Tz, V) for the NL3 location in the North Sea, ref. [3]. The water depth (d) at this location is 21 m. Note that this data applies to one specific water depth. Moreover, the investigations by Ballast Nedam Engineering on different water depths were done for the NL7 location. The significant wave height Hc and mean zero crossing period Tz vary between 0.25 m and 4.75 m and 2.75 s and 6.75 s respectively.

To account for scour, calculations will also be performed for inclination stiffness of the foundation corresponding to scour holes of 5 m, 10 m and 15 m, using the same water depth and scatter diagram. Linear wave theory is not suitable for the entire range of values of Hc, Tz and d and the use of third order Stream function theory should be considered for part of the calculations. It should be noticed however that the use of a deterministic wave train for fatigue load calculations of a wind turbine needs yet to be verified.

Ideally the simulation length should be 3 hours because wave statistics are based on 3-hour averages. But practical considerations limit the simulation length to about 30 minutes, using the same time step as in PHATAS, i.e. 0.02 seconds. The less than ideal simulation period will give approximately 20 percent underestimation of maximum wave heights. A sensitivity study must show whether this leads to a significant underestimation of fatigue loads.

The use of the Morison equation imposes limitations on the allowable spectral content of the wave time series. As a rule of thumb the smallest Fourier component must have a wavelength of at least five tower diameters. If a shorter wavelength, i.e. a higher cut-off frequency is specified then the MacCamy-Fuchs correction should be applied. To introduce a safety margin the cut-off frequency will typically be set to 0.6 Hz, (approximately 3P) without the correction mentioned above.

For the actual load computation the Morison equation is incorporated in PHATAS. Constant drag and added mass coefficients are used over the entire frequency range considered. The values are preliminary set to 1.4 for C_d and 1.0 for C_a.12 The coefficients need to be finalised in a yet to be prepared load set description document.

Because linear wave theory only specifies water particle velocity and acceleration up to the mean still water level, Wheeler stretching is used to account for the wave loading in the wave top (above MSL).

Furthermore, current is not taken into account and water density is set at 1025 kg/m³.

Aspects like long-crestedness, i.e. uniformity of the lateral wave field, and non-alignment of wind and waves are addressed later in the load set specification work.

---

12 The drag and inertia coefficient (cm = c_m + 1.0) depend on the smoothness of the cylinder and wave and water characteristics. The here assumed values are based on a typical "marine growth" as explained in the rules from DNV for fixed offshore structures, ref. [4], part 3 chapter 1 section 5, also similarly formulated in chapter 6.1.6 of the DNV classification notes, ref. [5].

For water velocities larger than 0.13 m/s the criterion that Re > 5 · 10³ applies. The Keulegan-Carpenter number (Ke) may be assumed as Ke < 10, hence c_d = 1.4 (see figure 1 with relative surface roughness k/D = 0.01, in ref. [4]). From figure 2 in the DNV rules it can be derived that 1.8 < c_m < 2.0.
8. CONCLUSIONS

Conclusions are here limited to the main characteristics of the DOWEC 6 MW pre-design. Evidently, the design is subject to change during the course of the project.

1. The aerodynamic performance of the turbine is very good (section 2.4).
2. Three possible resonance - rotor speed frequencies are enlightened in section 6.1:
   i.) reaction less or symmetric edgewise frequency = 6P (approximately 11.0 rpm),
   ii.) collective edgewise frequency = 6P (approximately 12.6 rpm), and
   iii.) tower bending = 1P (approximately 13.7 rpm).
   Further study should determine the danger of these resonance frequencies.
3. The modelled rotor mass imbalance is almost 6 times smaller than what is prescribed by Germanischer Lloyd (section 3.2).
9. REFERENCES

5. DNV; Classification Notes No. 30.5 Environmental conditions and environmental loads. March 2000, Høvik, Norway.
6. ESDU; No. 79026; Mean fluid forces and moments on cylindrical structures: (...). Manchester, March 1980.
14. Kooij, J.; e-mail correspondence, LM Glasfiber Holland, 12 November 2001. Files received:
   - "table.frb" (FAROB output file with structural and geometrical properties)
   - "OD.01.067" (word document with description of aerodynamic design)
   - "aero data LMH64-5" (airfoil coefficients (not used))
   - "LMH64-5 geometry" (EXCEL file with pre bend, blade geometry, etc.).
APPENDIX A: AIRFOIL CHARACTERISTICS LMH64-5

An explanation to the following graphs is given in chapter 1, page 11. The dotted lines represent the values from StC calculations added to the measured data (continuous lines).
NA64
aspect ratio 17

angle of attack [deg]
APPENDIX B: SOFTWARE

The following software has been used for the analysis of the DOWEC 6 MW pre-design so far:

- PHATAS-IV: /home/viper1/koert/bin/alpha/phatas6MW, date 25 January 2002
  /home/klavecimbel2/peerings/RECOFF/phat_source/phatas6MW: turbine specific PHATAS executable, under development at the time of writing.
- SWIFT: /home/klavecimbel1/danny/bin/SWIFTi2, date 17 January 2002
- ROWS: /home/klavecimbel1/danny/bin/rows, date 8 January 2002
- Blade pitch control: `parNM104.fi`, `ecnNM104.fi` and `conNM104.fi`, developed by Eric van der Hooft, date 19 March 2002

APPENDIX C: MONO PILE CLAMPING STIFFNESS

In addition to the information given in 5.2, page 31, the following tables provide more elaborate data on the derived clamping stiffnesses.

Explanation: $\Delta u$ horizontal displacement at seabed level
$\Delta \phi$ angle of rotation at seabed level, $\Delta \phi = \Delta u/\Delta z$
Mz corresponding bending moment at seabed level
Fz shear force at seabed level (5470 kN, constant).
f coupling between rotation and translation,
\[ f = k_\phi \cdot \left( \frac{u_{\text{base}}}{M_{\text{base}}} \right) = \frac{u_{\text{base}}}{\Delta \phi_{\text{base}}} \]
$k_\phi$ rotation stiffness, $k_\phi = M / \Delta \phi$

<table>
<thead>
<tr>
<th>decrease in seabed</th>
<th>pile length above seabed</th>
<th>$u_{\text{average}}$</th>
<th>$\Delta u$</th>
<th>$\Delta z$</th>
<th>Mz</th>
<th>$f$</th>
<th>$k_\phi$</th>
</tr>
</thead>
<tbody>
<tr>
<td>[m]</td>
<td>[m]</td>
<td>[m]</td>
<td>[m]</td>
<td>[m]</td>
<td>[kNm]</td>
<td>[m/rad]</td>
<td>[Nm/rad]</td>
</tr>
<tr>
<td>0</td>
<td>30</td>
<td>0.0972</td>
<td>0.0040</td>
<td>0.60</td>
<td>2.188E+5</td>
<td>14.58</td>
<td>3.282E+10</td>
</tr>
<tr>
<td>5</td>
<td>35</td>
<td>0.0992</td>
<td>0.0047</td>
<td>0.67</td>
<td>2.462E+5</td>
<td>14.13</td>
<td>3.509E+10</td>
</tr>
<tr>
<td>10</td>
<td>40</td>
<td>0.0919</td>
<td>0.0036</td>
<td>0.50</td>
<td>2.735E+5</td>
<td>12.76</td>
<td>3.799E+10</td>
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<tr>
<td>15</td>
<td>45</td>
<td>0.0708</td>
<td>0.0034</td>
<td>0.50</td>
<td>3.009E+5</td>
<td>10.41</td>
<td>4.424E+10</td>
</tr>
</tbody>
</table>

In Table 15 are flexibilities that were derived using the deflections and loading given by Ballast Nedam at the baseline sea bottom level of 21 m below MSL.

<table>
<thead>
<tr>
<th>scour depth</th>
<th>$u_{\text{average}}$</th>
<th>$\Delta u$</th>
<th>$\Delta z$</th>
<th>Mz</th>
<th>$f$</th>
<th>$k_\phi$</th>
</tr>
</thead>
<tbody>
<tr>
<td>[m]</td>
<td>[m]</td>
<td>[m]</td>
<td>[kNm]</td>
<td>[m/rad]</td>
<td>[Nm/rad]</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>0.0972</td>
<td>0.0040</td>
<td>0.60</td>
<td>2.188E+5</td>
<td>14.58</td>
<td>3.282E+10</td>
</tr>
<tr>
<td>5</td>
<td>0.1380</td>
<td>0.0041</td>
<td>0.50</td>
<td>2.188E+5</td>
<td>16.82</td>
<td>2.668E+10</td>
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<tr>
<td>10</td>
<td>0.1760</td>
<td>0.0064</td>
<td>0.67</td>
<td>2.188E+5</td>
<td>18.43</td>
<td>2.291E+10</td>
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<tr>
<td>15</td>
<td>0.2037</td>
<td>0.0073</td>
<td>0.70</td>
<td>2.166E+5</td>
<td>19.53</td>
<td>2.077E+10</td>
</tr>
</tbody>
</table>