International Wind Engineering Conference

Support Structures and Electrical Systems

3 + 4 September 2014, Hannover, Germany

Proceedings

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- **Hans-Dietrich Haasis, ISL, Bremen, Germany**
- **Michael Muskulus, NTNU, Trondheim, Norway**
- **Freede Blaabjerg, AAL, Aalborg, Denmark**

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### Industry Session at Test Center for Support Structures

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Wind Energy R&D strategy
a review

Jos Beurskens
SET Analysis
(Former ECN)

IWEC2014
Hannover
3 September 2014
Present R&D strategies (offshore)

• Focus almost completely on Cost Reduction (LCoE)

• Examples: GB, D, NL, DK, EU

• Why not maximising: \((\text{Value} \uparrow – \text{Cost} \downarrow)\)
  (= cost effectiveness)

• Value enhancement is difficult to quantify by KPI’s !!
Cost structure (offshore)

Break down of LCOE for a typical offshore wind farm
Source: BVG Consulting 2014, for KIC

- **CAPEX**: 66%.
  - Big chunks:
    - Turbine (**),
    - Foundations
    - Installation of foundation

- **OPEX**: 33%
  - Big chunks:
    - O&M
    - Grid charges

(**) Land based wind energy benefits more from cost reduction oriented wind turbine innovations than offshore wind does
Cost reduction potential (ex. D, GB)

GB
Offshore cost reduction.
Pathways Study, The Crowne Estate 2012

D
Cost reduction Potentials of Offshore Wind Power in Germany, St. Offshore Windenergie, Prognos and Fichtner, 2013

Note: Base cases are different!
Cost reduction potential (ex. NL)

NL

![Cost reduction potential graph](image-url)
Base cases for Cost reduction (ex. D, GB, NL)

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<th>Country</th>
<th>Cost reduction</th>
<th>Period</th>
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<td>United Kingdom</td>
<td>18-37%</td>
<td>2011-2020</td>
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<tr>
<td>Germany</td>
<td>32-39%</td>
<td>2013-2023</td>
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<tr>
<td>The Netherlands</td>
<td>40% (target)</td>
<td>2012-2020</td>
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Cost reduction targets are extremely ambitious. Including value enhancement in national targets leads to more realistic targets.

Analysis: by NWEA, Voormolen
Cost reduction (LCoE)
Mechanisms for cost reduction

- **Innovation** *(Incremental improvements: better modeling, improved aerodynamic models, better power electronics, improved testing and operational verification)* *(New advanced dedicated offshore concepts: integrated installation & foundation, down wind, passive yaw, etc.)*
- Up-scaling of operations
- Wind farm efficiency improvement
- Decrease & mitigate risks associated with uncertainties, inaccuracies, reliability
- Utilise synergies of integration in design, grid integration, multipurpose applications, etc.
The trap of new concepts

Different learning curves

- Incremental improvements / Debugging / advanced components
- Innovation oriented / new concepts

- Concept 1
- Concept 2

Service life time

Maintenance cost
Incremental improvement  
some examples

To expand the European drive train technology mix by including superconducting and magnetic pseudo direct drive generators in order to improve the reliability, to reduce the tower top mass of large offshore turbines at $P = 10-20$ MW and to reduce the influence of a heavily fluctuating world market of the Rare Earth metals.

Source: InnWind.EU  
See also: 2013 JRC Status Report
Neodymium for PM generators

China has production monopoly of appr. 97% and restricted export in 2010.

Nd used in electronic devices (grams), Electric vehicles (kg’s) and wind turbine generators (tonnes)

Source: www.peakresources.com.au
Incremental improvement
some examples

InnWind: Light weight rotors for 8 +MW turbines

Innovative airfoils
Winglets
Advanced models: EU AVATAR project

Blade Root Spoilers
Incremental improvement
some examples

Winglets for reducing tip losses and noise emission

Foto: Spitzner Engineers.
Up scaling; distributed blade control reduces fatigue loads
Distributed blade control
Reduces fatigue loads

Comparison with Individual Pitch Control:
• 15-25% reduction of fatigue damage equivalent load, depending on load case
• Can add up to 30%

Wind tunnel tests on a scaled smart rotor – flapwise root bending moment with and without flap activation
Incremental improvement
some examples

Distributed blade control reduces fatigue loads

BMU Foerderprojekt Smart Blades. DLR, ForWind, FhG-IWES

20-40% reduction in blade- en tower fatigue load

“Smart” material variable trailing edge flap

Aktive Rotorblätter passen sich der Windstärke an

UpWind
Rasmussen. DTU-Risø
Advanced concepts

InnWind: Innovations for 8+MW turbines

Tower top weight is critical for cost reductions

![Graph showing nacelle weight vs. rated power for differentMW turbines. The R&D goal is indicated by an arrow.]
Advanced concepts

Multi-rotor turbine?
Advanced concepts

Two bladers?
Wind farm efficiency

Source: ECN
Wind farm efficiency

FarmFlow simulations

Wd = 82°     U = 8 m/s     TI = 7%
Wind farm efficiency

Active Wake Control

▲ Software analysis on variety of wind farms
▲ Tests on 2MW Nordex turbines and scaled wind farm at ECN test site

Total farm efficiency production increases from 0.5 to 3% AND load reduction
Completing range of test facilities

- Specific wind energy facilities exist in the fields of:
  - Blade testing
  - (Blade) material testing
  - Drive train testing
  - Mechanical transmission testing
  - Electric conversion system testing
  - Climatic testing of drive trains/nacelles
  - Complete wind turbines onshore (also in extreme climate zones) & offshore

- And since today: (offshore) foundation testing

Also relevant for intertidal and cold climate sites!!
Cold climate foundations are more complex to design than offshore ones.
O&M cost are 25% - 30% of kWh cost (LCoE)
Revenue losses approximately 50% of O&M cost
O&M Overall strategy

Preventive Maintenance (PM) → Condition Based Maintenance (CBM)
Corrective Maintenance (CM)
O&M Overall strategy

Ultimate goal: Asset Life Extension

Compared O&M process in aerospace and wind energy technology with a doctor’s practice:
* Examine
* Diagnose
* Prognose
* Fix

Do we manage the entire process in WE engineering?

There is still a long way to go from sensor to analysis!

Source: ALEX project. M. Martinez (TU Delft)
O&M Overall strategy

Physics needed!!

Material properties
(Initial) Structural condition
Usage (monitoring)
Load spectrum (monitoring)
Environmental factors (monitoring)

Damage degradation modes for metals and composites (hidden damages due to manufacturing & inherent in materials)

Water and salt ingress
Freezing and thaw cycles
Wear and Fretting
Long term effects of UV Light exposure
Heat damage
Lightning Strike
Bird Impact
Erosion (blade leading edges)

Based on: Martinez, TUD
Some (arbitrary) striking examples of R&D for cost reduction

but ....

• Significant part of initial cost is financing
• These cost are associated with risk
• Which risk elements are endogenous?
• … and technology related?
Value enhancement

(Value↑ – Cost↓)
Value enhancement

If value is reflected in selling price of wind electricity the following topics gain relevance:

- Power output forecasting
- Power quality of wind farms
- Maximising energy production per kW installed power (maximising capacity factor)
- Recycling / end of life solutions
Value enhancement power quality (output variations)

Wind farm lay-out, effect of wind direction variations

DO NOT PLACE TURBINES IN STRAIGHT ROWS!

1% ambient t.int

10% ambient t.int

Torben J. Larsen, DTU-Risø
Value enhancement

Power rating is under estimated and poorly understood issue

- In the past: only maximising output [kWh] and minimising cost [€/kWh] and maximising MW’s
- Importance of capacity factor underestimated
- Urgent shift towards focus on balancing, cost of grid integration & electrical infrastructure
Value enhancement

Wind turbine power rating

\[ p = \frac{P_R}{A_{\text{rotor}}} \quad [\text{W/m}^2] \]

Source: J.P. Molly, DEWI
Value enhancement

Wind turbine power rating and capacity factor
Trend turbine rating: Low specific power $p$ in high winds

Source: J.P. Molly, DEWI

100 m hub height, Weibull $k=2.92$, $V_{ave} = 7.3$ m/s)
Value enhancement

Cost of de-rating wind turbines

Source: J.P. Molly, DEWI
Value enhancement

Wind turbine power rating and capacity factor

Source: J.P. Molly, DEWI
Value enhancement

Cost reduction of cables & transformer & generator & convertor as these cost are proportional to rated power and not to the amount of energy generated, converted and transported. This leads to lower generation cost of WE (and less government subsidies).

Low specific power in high winds:

Higher capacity factor (= equivalent full load hours) leads to higher penetration degree of wind energy systems, better predictability, lower storage capacity. This leads to higher value of WE.
Value enhancement

Wind turbines need to be re-designed

- Optimised low $p$, high $V$ rotor concepts
- Design control and safety strategies for lower load spectrum
- Incorporate these findings in integrated wind farm design
- ........
Implementation issues
Implementation issues

• Socialising grid leads to cost reductions for project owner, but there is no incentive for optimising for grid integration unless grid codes provide obligations, e.g. in setting a maximum value for specific power \([W/m^2]\) at a certain wind speed or requiring minimum values of capacity factor.

• Cost reduction and Value enhancement topics form the pieces of a large and complex jigsaw puzzle. Each piece has its own actor. Each actor has its own responsibility within a subsystem. In order to come close to the cost and value objectives a steady and long term government policy is needed and instruments which stimulate the desired effects and avoid the wrong ones.
Implementation issues

• Setting national targets of Wind Energy in MW’s is wrong and stimulates low capacity factor wind plants.

• Measuring progress of market growth and contribution of WE to national energy supply in MW’s is wrong.

It is like measuring the water temperature with a ruler.
We should not only know the **cost** of WE, but also the **value**!!!
Computer-aided optimization of support structures

Michael Muskulus

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International Wind Engineering Conference
September 3–4, 2014
”The ethics of our profession today does not allow any design for a structure without optimization.”

(Mungan, 2001)
Example: powerline transmission tower

M. Muskulus (michael.muskulus@ntnu.no)
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Computer-aided optimization of support structures

International Wind Engineering Conference, September 3–4, 2014
A case for structural optimization 1/3

The basic strategy of structural design without resort to optimization algorithms:

"Find the best design iteratively modifying the design variables by trial-and-error process based on the experience of the designer, under requirements on the structural responses and mechanical properties, e.g., the stresses and displacements against static/dynamic design loads, eigenvalues of free vibration, linear/nonlinear buckling loads, etc."

(Elishakoff & Ohsaki, 2010)
A case for structural optimization 2/3

- Structural optimization can answer the following two questions:
  - How can we define the best design?
  - What is the best strategy to modify the design variables if the current design is not the best?
• Structural optimization provides us with the following benefits:
  • A design satisfying all the constraints (feasible design) can be found automatically, and efficiently, while simultaneously minimizing the objective function, such as the total structural volume or weight.
  • Optimization is very helpful for designing complex structures, for which even an experienced designer cannot easily find a feasible design.
  • Even if the optimal solution cannot be used directly in design practice, the optimal design gives insight into a better design.
  • If we start with a feasible and nearly optimal design found by an expert, the design cannot be worse after optimization, and usually a better design can be found.
  • The trade-off relation between the structural cost and responses can be made clear, if optimization is carried out several times by modifying the input parameters such as the cost coefficients and upper bounds of responses.
• Gailelo Galilei wrote was is probably the first paper on structural optimization when he discussed the optimal shape of beams in 1638 AD.

• Nowadays, computer-aided structural optimization using modern gradient-based algorithms has become routine for structures with more than 1 Mill. finite elements...

• ...when a few static loadcases are considered
Challenges for wind turbine support structures

- Nonlinearities
- Complex environment
- Fatigue as design-driver
- Tightly coupled and strongly interrelated systems
- Specialized analysis software
- Many design variables and constraints

Optimization with numerical gradients
Total wind turbine optimization

(Ashuri et al, 2014)
Total wind turbine optimization

- Analysis based on 11 FLS and one ULS loadcase
- $12 \times 6 \times 24 = 17280$ simulations each iteration (10 min loadcases)
- **Four** iterations using convex linearization / Lagrange multiplier method
- Optimized turbine 4 percent more expensive, but 5 percent higher AEP
- Taking everything into account, LCoE reduced by 2.3 percent
Optimization with genetic algorithms

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Computer-aided optimization of support structures
International Wind Engineering Conference, September 3–4, 2014
A simple genetic algorithm for the OC4 jacket

(Pasamontes et al, 2014)

- Design variables coded as binary strings, subject to mutation and crossover (combination of two successful designs)
A simple genetic algorithm for the OC4 jacket

(Pasamontes et al, 2014)

- Stochastic algorithm (variable results)
- Typically **150-200 iterations** needed
- Limitations: Small population size, only one FLS loadcase analyzed
Results

- Highest damage utilization at the top
- Almost all range constraints close to active (except brace thickness)
- Leg parameters similar for all sections
- Brace parameters increase with depth
- Somewhat faster convergence with N=30 designs (nominal)
- More efficient with N=15 designs

- Further improvements possible
  - Re-analysis of previous designs during time domain simulations
  - Heuristic bounds for when analysis is needed
  - Possible to analyze 2N designs in the same time
• **Straightforward** extension

• Node positions: 14 bits ($\pm 8192$ mm)

• Number of parameters: 178 bits (19 variables) $\approx 10^{53}$ designs
Results

- Internal bay heights remain more or less the same
- Optimum design has increased height of top bay
Results

- Internal bay heights remain more or less the same
- Optimum design has increased height of top bay
Statistical modeling of fatigue damage
Fatigue damage under power production

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Computer-aided optimization of support structures

International Wind Engineering Conference, September 3–4, 2014
Linear statistical model

(Zwick & Muskulus, *submitted*)

- Example: damage estimated for each hotspot ($i = 1, 2, \ldots$) in terms of damage at three wind speeds.

$$D^{(i)} = d_0^{(i)} + d_9^{(i)} D_9^{(i)} + d_{14}^{(i)} D_{14}^{(i)} + d_{19}^{(i)} D_{19}^{(i)} + \epsilon$$

- Trained with three models: original (M100), scaled (M110) and random (R1)

- Fatigue damage estimates accurate within 6.5 percent for all joints

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Computer-aided optimization of support structures

International Wind Engineering Conference, September 3–4, 2014
Damage distribution scaling
Motivation

(Sandal, 2014)
Motivation

- Observation: the normalized damage accumulation from different wind speeds kept a more or less constant distribution, even with large modifications of the design
- Therefore one load case can represent all wind speeds (after calibration)
- A simulation length of 10 min gives acceptable accuracy
Local optimization

- Optimization based on a domain-decomposition principle
- Approximate problem solved using two locality assumptions:
  - The loads on a member are not changed when changing the geometry of the member
  - The loads on a member are not changed when changing the geometry of other members in the structure

(Zwick & Muskulus, 2012)
Local optimization

- Simplification of stress concentration factors (Sandal, 2014):
  \[
  \sigma_{\text{Leg}} \propto t^{0.9} \frac{1}{T^{2.4}D^{0.5}}
  \]
  \[
  \sigma_{\text{Brace}} \propto \frac{1}{t^2}
  \]
  where \( D, T \) and \( d, t \) are diameter and thickness for legs and for braces, respectively.
- Fixing the \( D/T \) ratio, the following parameter update during an iteration aims for full damage utilization \( U^j \to 1 \):
  \[
  \Delta t = t_j \left( (U^j_{\text{Brace}})^{\frac{1}{10}} - 1 \right)
  \]
  \[
  \Delta T = T_j \left( (U^j_{\text{Leg}})^{\frac{1}{14.5}} \left( \frac{t_j + \delta t}{t_j} \right)^{\frac{4.5}{14.5}} - 1 \right)
  \]
Results

- The algorithm finished in ca. 10 iterations.
- The final design seems to be independent of initial design.

(Sandal, 2014)
The weights of the two concepts are approximately the same, not including the transition piece.

(Sandal, 2014)
Optimization with design sensitivity analysis
Design sensitivity analysis

- Equation of motion for a linearly damped finite-dimensional structure subject to time-varying loads $P(t)$:
  $$M\ddot{U}(t) + C\dot{U}(t) + KU(t) = P(t)$$
- After time-domain analysis the displacements $U(t)$, velocities $\dot{U}(t)$ and accelerations $\ddot{U}(t)$ are known.
- Differentiating both sides with respect to a design parameter $A_i$ one obtains
  $$M\frac{\partial \ddot{U}}{\partial A_i} + C\frac{\partial \dot{U}}{\partial A_i} + K\frac{\partial U}{\partial A_i} = Q$$
  where
  $$Q = \frac{\partial P}{\partial A_i} - \frac{\partial M}{\partial A_i} \ddot{U} - \frac{\partial C}{\partial A_i} \dot{U} - \frac{\partial K}{\partial A_i} U.$$
- Assuming $\frac{\partial U}{\partial A_i} = 0$ and $\frac{\partial \dot{U}}{\partial A_i} = 0$ at $t = 0$ this can be solved by another time-domain integration.
- Tested for offshore wind turbine jackets by (Jørgensen & Nissen, 2013)
"New methods for treating uncertainty will become important in virtually all branches of mechanics."

(Oden, Belytschko, Babuska and Hughes, 2003)
• During the 50th anniversary on May 24, 1987, authorities expected 50000 people to show up and walk over the bridge.
• Instead of this, **800000 people** came to walk over the bridge.
A near disaster?

- The live load on the bridge was about two times the maximum load possible with cars, and **within 6 percent** of the maximum design load.
- However, due to uncertainties in calculation methods in the 1930s, a **safety factor** of another 150 percent applied and no danger for structural failure existed.
Over-conservatism or over-optimization?

- Typically, optimal solutions are not resilient against even small violations of the assumptions used in their optimization.
- These can be as simple as manufacturing tolerances...
- A well-known example is the instability for buckling when small imperfections are present ("Dangers of structural optimization", Thomson & Hunt, 1974)
- Safety factors can guard against such uncertain influences, to some extent.
- This incurs a cost, obviously, and in a marginal business this is of concern...
- Moreover, safety factors are a mere practical tool for producing a qualified product, not a substitute for calculating probabilities of failure (structural reliability).
- Ideally, we have to quantify all uncertainties involved and optimize the structure for a target reliability level (e.g. probability of failure $p < 10^{-4}$)
Robust optimization

- There is a need for **taking uncertainties** explicitly into account when optimizing structures ("robust optimization")
- Two methodologies distinguish themselves by their targets:
  - Robust-design optimization (RDO): minimizes some outcome measure (e.g. weight) and its variance
  - Reliability-based design optimization (RBDO): minimizes probability of failure, in addition to outcome measures
ABYSS: Advancing BeYond Shallow waterS

• ABYSS: Optimal design of offshore wind turbine support structures is a four year research project funded by the Danish Council for Strategic Research.

• ABYSS develops novel mathematical models, reliable numerical optimization techniques and software for optimal design of cost effective bottom-fixed offshore wind turbine support structures for all relevant water depths including deep waters in excess of 50m.

• ABYSS includes a PhD project on "Design optimization under uncertainties"

• More information at: www.abyss.dk

M. Muskulus (michael.muskulus@ntnu.no)
For exploring future challenges and shifting boundaries in science and technology, the members of EAWE have defined ten ambitious R&D focus areas for the period 2014 - 2025.

**Topic 7: Design methods**

http://www.eawe.eu/longtermresearch/

M. Muskulus (michael.muskulus@ntnu.no)  
Norwegian University of Science and Technology  
Computer-aided optimization of support structures  
International Wind Engineering Conference, September 3–4, 2014
Computer-aided optimization of support structures is a challenging field

Starting to see first useable methods for deterministic, global design

Strong need for robust and probabilistic approaches

Still need to bring the available solutions to market...

"Optimal design became a rich and rewarding field of research"

(Niordson, 2001)
References 1/2


Future trends in the electrical drive system for wind turbines

Dr. Frede Blaabjerg
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Aalborg University
Department of Energy Technology
Aalborg, Denmark

CORPE
Center of Reliable Power Electronics
www.corpe.et.aau.dk
Future trends in the electrical drive system for wind turbines

- Demands to wind turbines
- Power Converters for Wind Turbines
- Power Device Technology
- Press-pack versus module
- Mission profiles analysis and translation
- Reliability tests & monitoring based on IGBT modules
- Conclusions
Demands to wind turbines
Needs for Lower Cost of Wind Power

Different trends
But the Cost of Energy will be reduced
Requirements for Wind Turbine Systems

General Requirements & Specific Requirements

1. Controllable I
2. Variable freq & U

1. Energy balance/storage
2. High power density
3. Strong cooling
4. Reliable

1. Fast/long P response
2. Controllable/large Q
3. Freq & U stabilization
4. Low Voltage Ride Through

Generator side

Wind Power Conversion System

Grid side
Grid Codes for Wind Turbines

**Conventional power plants** provide active and reactive power, inertia response, synchronizing power, oscillation damping, short-circuit capability and voltage backup during faults.

**Wind turbine technology** differs from conventional power plants regarding the converter-based grid interface and asynchronous operation.

**Grid code requirements today**
- Active power control
- Reactive power control
- Frequency control
- Steady-state operating range
- Fault ride-through capability

**Wind turbines are active power plants.**
Frequency control through active power regulation.

Reactive power control according to active power generation.

Voltage support through reactive power control.
Requirements during grid faults

- Withstand extreme grid voltage dips.
- Contribute to grid recovery by injecting $I_q$.
- Higher power controllability of converter.
More Power Electronics

Example of the power electronics technology in wind turbine applications.

- **Power electronics technology** will be more active into the grid.
- **Advanced power electronics systems** in the future RESs to enable a better and flexible integration.

Global installed wind capacity (until 2013): +320 GW, 2013: +50 GW
Power has to be controlled by means of the aerodynamic system and has to react based on a set-point given by a dispatched center or locally with the goal to maximize the power production based on the available wind power.
Power Converters for wind turbines
Wind Turbine Concepts

- **Wound-rotor induction generator**
- **Variable pitch – variable speed**
- **±30% slip variation around synchronous speed**
- **Power converter** (back to back/direct AC/AC) in rotor circuit

- **Variable pitch – variable speed**
- **With/without gearbox**
- **Generator**
  - Synchronous generator
  - Permanent magnet generator
  - Squirrel-cage induction generator
- **Power converter**
  - Diode rectifier + boost DC/DC + inverter
  - Back-to-back converter
  - Direct AC/AC (e.g. matrix, cycloconverters)
Back-to-back two-level voltage source converter

- Proven technology
- Standard power devices (integrated)
- Decoupling between grid and generator (compensation for non-symmetry and other power quality issues)

- Need for major energy-storage in DC-link (reduced life-time and increased expenses)
- Power losses (switching and conduction losses) $\Rightarrow$ low efficiency
- Scalable
Multi-level converter technology to handle with higher power

- More output voltage levels $\rightarrow$ Smaller filter
- Higher voltage, and larger output power
- Unequal losses on the inner and outer power devices $\rightarrow$ derated converter power capacity

Multi-cell converter technology is adopted to solve this issue, e.g. converters introduced by Gamesa and Siemens.
Future Generation of HV Wind Turbines

• **Features:**
  – Transformer-less operation
  – Low cooling system requirements
  – Less copper required

• **Challenges:**
  – Reliable insulation systems
  – Short circuit protection
  – Safety

---

**LV wind turbine (0.69 kV AC)**

**MV wind turbine (3.3 kV AC)**

**HV wind turbine (12-66 kV AC)**
3L-NPC vs. MMC solutions

• HV converter candidates:
  – NPC (Neutral Point Clamped), adapted from medium voltage drives
  – MMC (Modular Multilevel Converter), adapted from HVDC systems.

- high switching frequencies
- required series-connected IGBTs.
- moderate filtering requirements.
- integrated design
- high dv/dt stress

- low switching frequencies
- no series-connected IGBTs.
- low filtering requirements.
- modular design
- low dv/dt stress
Power Device Technology
Power Devices Applications

- **High-End Solutions**
  - 200 V: GaN
  - 600 V: Super Junction MOSFET
  - +1200 V: SiC

- **Middle-End Solutions**
  - 200 V: GaN / Super Junction MOSFET
  - 600 V: Silicon IGBT / Super Junction MOSFET
  - +1200 V: SiC

- **Low-End Solutions**
  - 200 V: Silicon IGBT / Super Junction MOSFET
  - 600 V: Silicon IGBT

Source: Yole Developpement. Status of the power electronics industry. 2012
## Power Devices for Wind Power Applications

<table>
<thead>
<tr>
<th></th>
<th>IGBT module</th>
<th>IGBT Press-pack</th>
<th>IGCT Press-pack</th>
<th>SiC MOSFET module</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Power Density</strong></td>
<td>Low</td>
<td>High</td>
<td>High</td>
<td>Low</td>
</tr>
<tr>
<td><strong>Reliability</strong></td>
<td>Moderate</td>
<td>High</td>
<td>High</td>
<td>Unknown</td>
</tr>
<tr>
<td><strong>Cost</strong></td>
<td>Moderate</td>
<td>High</td>
<td>High</td>
<td>High</td>
</tr>
<tr>
<td><strong>Failure mode</strong></td>
<td>Open circuit</td>
<td>Short circuit</td>
<td>Short circuit</td>
<td>Open circuit</td>
</tr>
<tr>
<td><strong>Easy maintenance</strong></td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td><strong>Insulation of heat sink</strong></td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td><strong>Snubber requirement</strong></td>
<td>-</td>
<td>-</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td><strong>Thermal resistance</strong></td>
<td>Large</td>
<td>Small</td>
<td>Small</td>
<td>Moderate</td>
</tr>
<tr>
<td><strong>Switching loss</strong></td>
<td>Low</td>
<td>Moderate</td>
<td>Moderate</td>
<td>Low</td>
</tr>
<tr>
<td><strong>Conduction loss</strong></td>
<td>Moderate</td>
<td>Moderate</td>
<td>Moderate</td>
<td>Large</td>
</tr>
<tr>
<td><strong>Gate driver</strong></td>
<td>Moderate</td>
<td>Moderate</td>
<td>Large</td>
<td>Small</td>
</tr>
<tr>
<td><strong>Major manufacturers</strong></td>
<td>Infineon, Semikron, Mitsubishi, ABB, Fuji</td>
<td>Westcode, ABB</td>
<td>ABB</td>
<td>Cree, Rohm, Mitsubishi</td>
</tr>
<tr>
<td><strong>Medium voltage ratings</strong></td>
<td>3.3 kV / 4.5 kV / 6.5 kV</td>
<td>2.5 kV / 4.5 kV</td>
<td>4.5 kV / 6.5 kV / 10 kV</td>
<td>1.2 kV / 10 kV</td>
</tr>
<tr>
<td><strong>Max. current ratings</strong></td>
<td>1.5 kV / 1.2 kA / 750 A</td>
<td>2.3 kA / 2.4 kA</td>
<td>3.6 kA / 3.8 kA / 2 kA</td>
<td>180 A / 120 A</td>
</tr>
</tbody>
</table>
PressPack Technology

✓ Press-pack devices vs. IGBT module
✓ 3L-NPC medium voltage solution
✓ MMC high voltage solution

By Cristian Busca, Michal Sztykiel
PhD fellows, Aalborg University
cbu@et.aau.dk, msz@et.aau.dk
Press-pack IGBTs vs. Module IGBTs

In Al wire-bonded IGBTs the interconnections are realized with solder/bonding. In press-pack IGBTs the interconnections are realized with pressure contact.

Source: F. Wakeman, et al

Source: I. F. Kovacevic, et al

Al wire-bonded IGBTs
Reliability improvements

<table>
<thead>
<tr>
<th>No.</th>
<th>Wire-bonded IGBT modules</th>
<th>Press-pack IGBT modules</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Bond wire lift-off</td>
<td>Fretting damage</td>
</tr>
<tr>
<td>2</td>
<td>Solder joint fatigue</td>
<td>Spring fatigue</td>
</tr>
<tr>
<td>3</td>
<td>Bond wire heel cracking</td>
<td>Spring stress relaxation</td>
</tr>
<tr>
<td>4</td>
<td>Aluminium reconstruction</td>
<td>Cosmic ray induced failures</td>
</tr>
<tr>
<td>5</td>
<td>Cosmic ray induced failures</td>
<td></td>
</tr>
</tbody>
</table>

Each packaging technology has its own characteristic failure mechanisms

Source: IXYS Westcode UK

Press-pack IGBTs outperform wire-bonded IGBTs in terms of thermal cycling capability
Issues with press-pack IGBTs

Clamping force in press-pack IGBTs influences thermal contact resistance and electrical contact resistance.

The Finite Element Method (FEM) is used to model the clamping pressure distribution among the chips under various mechanical clamping conditions.
Non-uniform clamping pressure among chips

Ideal clamping

Thickness variation of chip assemblies

Thickness variation + Misalignment in clamp
Performances with non-uniform pressure

Press-pack IGBT stack
(2 IGBTs and 3 cooling plates)

Chip-level clamping pressure on collector side Mo plates

Current distribution at non-uniform pressure
## A 3L-NPC setup based on PP IGBT

### Setup parameters

<table>
<thead>
<tr>
<th>Power Circuitry</th>
<th>Phase-to-neutral output voltage</th>
<th>1.7kV&lt;sub&gt;rms&lt;/sub&gt; (50Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated apparent power, S</td>
<td>2MVA</td>
<td></td>
</tr>
<tr>
<td>DC bus voltage, V&lt;sub&gt;DC1&lt;/sub&gt;, V&lt;sub&gt;DC2&lt;/sub&gt;</td>
<td>2500V</td>
<td></td>
</tr>
<tr>
<td>Inductance, L</td>
<td>450µH (10%)</td>
<td></td>
</tr>
<tr>
<td>Inductor resistance, r&lt;sub&gt;L&lt;/sub&gt;</td>
<td>1.5mΩ</td>
<td></td>
</tr>
<tr>
<td>Transformer turns ratio, N</td>
<td>1:1</td>
<td></td>
</tr>
<tr>
<td>Transformer leakage inductance, L</td>
<td>325µH (7.2%)</td>
<td></td>
</tr>
<tr>
<td>DC bus resistance, r&lt;sub&gt;DC&lt;/sub&gt;</td>
<td>3.2kΩ</td>
<td></td>
</tr>
<tr>
<td>Capacitance, C</td>
<td>1.1mF</td>
<td></td>
</tr>
<tr>
<td>Capacitor (Westcode, 3600V-220µF)</td>
<td>E51.S30-224R20</td>
<td></td>
</tr>
<tr>
<td>IGBT-diode pair (Westcode)</td>
<td>T1800GB45A</td>
<td></td>
</tr>
<tr>
<td>Gate resistance, R&lt;sub&gt;Gon&lt;/sub&gt; &amp; R&lt;sub&gt;Goff&lt;/sub&gt;</td>
<td>2.5Ω &amp; 3.8Ω</td>
<td></td>
</tr>
<tr>
<td>PWM frequency, f&lt;sub&gt;PWM&lt;/sub&gt;</td>
<td>1050Hz</td>
<td></td>
</tr>
<tr>
<td>Sampling time, T&lt;sub&gt;S&lt;/sub&gt; (double-update)</td>
<td>476.2µs</td>
<td></td>
</tr>
<tr>
<td>Dead time</td>
<td>10µs</td>
<td></td>
</tr>
<tr>
<td>Cooling System</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooling plate (Westcode, AlN ceramic)</td>
<td>XW180GN25A</td>
<td></td>
</tr>
<tr>
<td>Water flow rate through cooling plates, ν</td>
<td>5l/min</td>
<td></td>
</tr>
<tr>
<td>Ambient temperature, T&lt;sub&gt;amb&lt;/sub&gt;</td>
<td>15-20°C</td>
<td></td>
</tr>
<tr>
<td>Main cooling water temperature, T&lt;sub&gt;cw&lt;/sub&gt;</td>
<td>10-15°C</td>
<td></td>
</tr>
</tbody>
</table>
Converter performances – 3L-NPC

Outputs – steady state

Output transition from 0.5MW to 1MW

Thermal transition between 0.5MW and 1MW

\[ V_{DC} = 2500 \text{V}, \ P = 1.25 \text{MW}, \ \text{PF} = 1, \ \text{and} \ v_{bn} = 1700 \text{V}_{\text{rms}} \]
Mission profiles analysis and translation

- Mission profiles of wind power converter
- Mission profile vs. reliability

By Ke Ma
Postdoc, Aalborg University
kema@et.aau.dk
Mission profiles of WTC

Harsh environment

Limited space

Grid faults

Variable wind and temp.

All have impacts to component stress and reliability!

Q support

1. Controllable
2. Variable freq & U
1. Energy balance/storage
2. High power density
3. Cost effective
4. Reliable

1. Fast/long \( P \) response
2. Controllable/large \( Q \)
3. freq & \( U \) stabilization
4. Low Voltage Ride Through

P/Prated (p.u.)
Q/Prated (p.u.)

0.2
0.4
0.6
0.8
1.0

-0.3
0
0.4

-0.3
0
0.4

-0.3
0
0.4

-0.3
0
0.4

Underexcited
Overexcited

Keep connected above the curves

P
Q
P
Q

Generator side
Wind Power Converter System
Grid side

Danmark
Spain
Germany
US

0
25
75
90
100
150 500 750 1000 1500

Voltage(%)
Time (ms)
Wind speeds & grid codes to thermal stress

Mission profiles

Wind speed variations

Thermal loading of power devices

Tj Tc vs. time

LVRT and Q injections

Tj Tc vs. grid voltage dips
Mission profiles translations to reliability

Mission Profile

Converter Design

Stress Analysis
- Mission analysis
- Load estimation
- ...

Loading profile

Strength Modeling
- Failure mechanism
- Accelerating test
- Field feedback
- ...

Lifetime

Variation & Statistics
- System combination
- Stress/devices variation
- Production robustness
- ...

Reliability information

Robustness margins

Mean cumulative failure rate (MCF) Curve

Lifetime

0%
10%
5%
0% 1000 2000 2000

 Zuckerberg
E.g. 1 year wind speeds to lifetime

Frame work for mission profiles translations.


Consumed lifetime vs. different wind speeds.
Reliability tests & monitoring based on IGBT modules

- Failures mechanisms of IGBT module
- Vce monitoring of IGBT modules
- Lifetime cycling test

By Pramod Ghimire
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Failure Mechanisms of IGBT modules

Package related failures
- Bond fire fatigue
- Solder fatigue
- DCB failures
- Partial discharge

Stressors:
\[ T, \Delta T, T_m, dT/dt, \Delta H, J \]

Chip related failures
- Time dependent dielectric breakdown
- Electro-migration
- Transient electrical stresses
- Cosmic-rays

Bond wire heel cracking
Latch up
Bond wire lift off
Crack in chip

Motivations and approaches

Online Monitoring:
- Real time monitoring of electrical parameters in converter operation
- Estimation of junction temperature online
- Knowing wear-out/ ageing of the device
- Practical implementations in real field

Lifetime tests:
- A test set up for wind power converter
- Power converter operates at normal PWM switching operation
- Sinusoidal loading of current for the device
$V_{ce}$ monitoring of IGBT modules

$V_{ce}$ monitoring
- 1mV accuracy of measurement
- Easy to implement in gate drivers
- No additional effects to converter operation and device failures

Online and offline Vce / Vf monitoring

Online on-state voltage monitoring

- a. Load current, fundamental freq. 6Hz
- b. On-state voltage drop for low side IGBT and diode
- c. On-state voltage drop for high side IGBT and diode


Offline on-state voltage monitoring

- Fails after 5.1 M cycles at 6 Hz.
- Max. 7 mV increment
- Increment started after 4.5Mcycles
Summary  Wind turbine technology

- Cost of Energy is the main driver (and added value)
- Power Electronics is the enabling technology for superior generator and grid control
- Many topologies still on the table (DFIG, full scale, PM-less solutions)
- Medium voltage not very much used yet
- New power devices can be introduced
- More intelligence put into power electronics
- Design for reliability in wind turbine drive trains
- Park-scale system configurations are not discussed here – like dc-grid
Key References

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The Early-Age Cycling and its Influence on the Properties of Hardened Grout Material

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Keywords: Offshore Wind Turbine, Grouted Joint, Grout, Early Age Cycling Effects

1 INTRODUCTION

The objective of the investigations was to point out the influence of early-age movements on the structural behavior of grouted connections. Another objective of the investigation was researching to evaluate different grout materials with the developed test facility. Grouted joints are used for the supporting structures of offshore wind turbines. They connect the substructures to the piles or the piles to the transition piece depending on the kind of supporting structure. The different kinds of supporting structures and the position of grouted joints are shown in Figure 1. The gap between pile and sleeve, is later filled with grout, which is typically a high-performance mortar or cement paste. High demands are placed on the materials of grouted joints with regard to the strength of the concrete in the early age, the processing of concrete, the flowing property, and the sedimentation stability [4,7].

Figure 1: Supporting structures, from left to right: monopile, tripile, tripod, jacket
A further basic requirement is a low risk of cracking [1,4,5]. Further, there are additional requirements regarding to the early age strength of the grout as a result of the small time slot or installation and the low temperatures in the depths of the sea [22]. The maritime environmental conditions also place high demands on the building materials with regard to the durability properties. Offshore platforms are exposed to an aggressive and constantly high level of a mixture of salt water and air, especially in the splash zone [1]. Requirements for the grout used, construction, structural design, and manufacturing process for offshore wind turbines can be found in standard regulations [3,8,12,14]. The main problem during the installation process is the application of the grout under offshore conditions [21]. Grouted joint connections for offshore wind turbines are already subjected to dynamic loadings during the manufacturing process. This is caused by wind, sea state and the manufacturing process itself [24]. Wind and waves are dominant factors relating to the load-carrying capacity [1].

The result caused by these loadings could be a relative movement between pile and sleeve, which depends on the type of structure and its stiffness. The grout material in the early age is very susceptible to deformations, which reduce the load-carrying capacities [1]. This leads to a disturbance in the structure of the hardened mortar or cement paste [9,16,17]. For this reason the guideline [9] states, the capacity of the grout matrix, considering of an early-age cycling reduction factor, may be taken as:

\[
\tau_{kg} = \kappa \cdot f_{ck}^{0.7} \cdot (1 - e^{-2L/R_p})
\]

where:

- \(\tau_{kg}\) = characteristic shear strength of the grout
- \(f_{ck}\) = characteristic compressive cube strength of the grout
- \(L\) = effective grouted connection length
- \(R_p\) = pile outer radius
- \(\kappa\) = early age cycling reduction factor

\[
\kappa = \begin{cases} 
1 - 3 \frac{\Delta}{R_p} & \text{for } s/\sqrt{R_p t_p} < 3 \\
1 & \text{for } s/\sqrt{R_p t_p} \geq 3
\end{cases}
\]

- \(\Delta\) = early-age cycling movement
- \(t_p\) = wall thickness of pile
- \(s\) = shear key spacing.

The calculated value (1.1) with the reduction factor corresponds to the values with the results out of the investigations. However, the new version of this Guideline [10] sets standards that the movement shall be limited to maximum 1 mm during the 24-hour period. Thus, there is a significant discrepancy between these two versions of guidelines. Unquestionably, these movements occur, though their range is unknown [2,15,17,19]. Figure 2 shows a schematic and simplified drawing of a grouted joint and the possible deformation of the gap between pile and pin.
In principle, these movements could lead to three adverse effects on the strength of the grout and on the load-bearing capacity of the grouted joint (Figure 3). First of all, the loads caused by movement could lead to a sedimentation and segregation process of the self-compacting grout. Another effect which could occur is structural damage of the grout matrix. The third effect which should be considered is plastic deformation of the hardening grout in the gap between pile and sleeve or pile and pin. In the following, the reported effects are described in more detail.

2 EARLY AGE CYCLING EFFECTS ON THE PROPERTIES OF HARDENED GROUT MATERIAL

It is possible that the aggregates in the grout sediment and segregate due to the movement of the connections. This process is known in the application of self-compacting concrete [2]. The German DAfStb Guideline [6] on self-compacting concrete states that applying an additional compacting energy on the self-compacting concrete is not allowed. In view of today’s applications of grout materials for offshore wind turbines, a specialized, high-strength, and self-compacting material is used [1, 21]. The movement could lead to a progressive accumulation of the larger particles in the lower area, while the cement paste and the fine grained particles remain in the upper part. This could have an impact on the hydration temperature and the autogenous shrinkage in the upper part of the grouted joint [23], leading to a higher cracking tendency. Another effect which can occur is structural damage in the cement paste matrix of the hardening grout material during the hydration process. This could have an influence on the mechanical and durability properties, especially with regard to the fatigue strength of the grout because of micro-cracking [12]. Further, plastic deformations might occur due to the loadings, thereby creating cavities around the shear keys [13, 18]. This could undermine the mobilization of the maximum load-carrying capacity of the whole structure. The possible adverse effects are described in [19] in more detail. In the following, the reported effects are summarized in Figure 3.
Based on these considerations, a test facility was developed that enables the simulation of the influences of early age cycling effects on the properties of hardened grout material. The following chapter will explain the developed test facility.

3 Test System to Simulate the Influence of Early Age Cycling

The whole construction of the test apparatus is based on a steel framework with wooden formwork panels. The construction thus forms two containers for the grout material. The wooden board in the middle of the containers is equipped with two joints in the bottom of the construction. The dimension of the base area of the grout containers is 12 cm x 80 cm and the height is 120 cm, with a total capacity for each container of approximately 115 dm³. The torque (25 Nm) of the engine used will be transmitted to the board in the middle of the two containers. In this way, the movement between pile and sleeve can be simulated under laboratory conditions on a small scale. More specific information of the test data is given in [19]. The test system is shown in Figure 4.

Figure 3: Early-age cycling effects [19]

Figure 4: Schematic drawing of the test facility [19]
4 EXPERIMENTS

The test series consisted of three different grout materials. The first test was performed with a high-performance mortar grout with a maximum grain size of 5 mm (material “A”). The grout mixture was prepared with a water-solid ratio of 7% and a slump flow of 830 mm. The second test was carried out with a high-performance mortar grout, too (material “B”). Also with a maximum grain size of 5 mm, a water-solid ratio of 8.5% and a slump flow of 830 mm. The third test was performed with an ordinary portland cement (material “C”), so called OPC-grout, prepared with a water-cement ratio of 0.45, with the result of a slump flow of 830 mm. Apart from the cyclic test (“cyclic test”) with the test system, a reference test (“static test”) without movements was carried out to compare the two tests from a static and a dynamic point of view. Samples were prepared to prove the compressive strength. Therefore, prisms with an edge length of 40 mm x 40 mm x 160 mm were produced from the hardening grout objects. In addition, formwork casted prisms (“reference test”) with the same edge length were also prepared as a further comparison (according to [11]).

![Figure 5: Front view of the grout object (left side) and (right side) the numbering of the samples to prove the compressive strength (white rectangle) and the plates to carry out the counting (white strips)](image)

The diagram in Figure 6 shows the compressive strength of the cyclic test with the test facility, the static test with a static wall and the mean of the values of the reference test with formwork casted prisms (broken black line).

![Figure 6: Compressive strength of material A, B and C](image)
It is clearly visible that the samples from the upper part have a reduced compressive strength in contrast to the samples from the lower part, of each material. Furthermore, the diagrams show a higher scattering of the values of the dynamic test series in comparison to the static test. In addition, the values of the static test show similar results as the reference test values. In terms of comparing the different grouts, significant differences are recognizable. Considering first material “A”, the results of the samples from the upper part show a reduction of 20% in contrast to the mean of the values of the reference test. The situation is different to material “B” (-38%) and material “C” (-44%). Thereby, material “A” shows a reduced compressive strength with more than 85 N/mm² still a high level.

Further, a simple count was carried out in order to verify the sedimentation process. Plates were resected out of the object of the cyclic and the static test. In total, 20 plates were resected out of the test objects to carry out the counting (white strips in Figure 5). All particles with a minimum size of 2 mm were counted. Figure 7 shows the results of the counting. The number of the particles (>2 mm) per 100 cm² counted is plotted on the abscissa axis and the designation of the plates is plotted on the ordinate axe, from 1 (upper part) to 20 (lower part). The count showed that a sedimentation process took place, for each material (material “A” and “B”). The cyclic load leads to a progressive accumulation of the coarse aggregates in the lower part. It can be seen that the upper part exhibits a reduced proportion of larger particles.

![Figure 7: Number of particles (>2 mm) per 100 cm² in different plates of material “A” and “B”](image)

The upper part of material “A” (relative viscosity of 2.8 kNs m²) shows a reduced proportion of larger particles with a difference of 65%. The discrepancy of larger aggregates of material “B” (relative viscosity of 1.8 kNs m²) is with 85% even greater. Based on the area between the static and the dynamic curve the sedimentation tendency can be derived. The larger the area, the greater is the sedimentation tendency, expressed in the formula below.

\[ A_1 + A_2 < B_1 + B_2 \]  \hspace{1cm} (3)

The last effect which can be determined is plastic deformation. The hardened grout object in the test facility shows the effect of plastic deformations, as shown in Figure 8. A gap was formed between the grout and the middle wall caused by movement and the toughening grout...
over time. In this example the gap has a size of about 1 cm. The shear keys are not enclosed by grout, especially in the upper area.

Figure 8: Example of the gap between the grout object and the middle wall [19]

### 5 Evaluation

The reduction of the compressive strength could be an indication of structural damage in the cement paste matrix of the hardening grout material during the hydration process, caused by dynamic loads. Dynamic loads could lead to a higher macro and micro-cracking in the structure of the grout matrix. Considering the results of the static test, the measured values underline the impression. The compressive strength of the samples from the upper part of the cyclic test is noticeably lower than the results of the samples from the static test. Dependent of the horizontal position, the results show significant differences. This is also proven by the results of material “C”. The accumulation of the cement paste leads to a higher cracking tendency caused by a higher hydration temperature and higher autogenous shrinkage in the upper part. This could have an influence on the mechanical and especially on the durability properties and, furthermore, on the fatigue strength of the grout material. The results of the test with material “A” and “B” show another reason to explain the reduction of the compressive strength of the samples. In addition to structural damage, a decisive reason for differences in the values of the compressive strength of the cyclic test, dependent of the position, could also be a sedimentation process. It is obvious that the sedimentation process is responsible for the lower values. The sedimentation process leads to a non-homogeneity in the structure, which could have an impact on the mechanical properties. This could have a major influence on the mechanical properties of the grout. Last, the gap between the grout object and the middle board leads to a negative force-fit and form-fit connection. Due to advancing hardening, from a certain point on, the elastic deformations can no longer be restored. The gap could undermine the mobilization of the maximum load-carrying capacity of the whole structure. In a comparison of the three materials, the investigation points to a higher resistance to early age movements of material “A”, because of the lower difference in the compressive strength between the samples from the upper part and the lower part in comparison to the samples of the cyclic and the reference tests. The greater the fresh properties, the lower the influence of early age movements on the properties of the hardened grout, and thus, the influence on the mechanical and durability properties of the grouted joint. It has to be pointed out that the mentioned effects are influenced through several additional circumstances as
temperature conditions, type and stiffness of the structure, damping of the sealing, and of course by the intensity of wave-impact.

6 SUMMARY

The objective of the investigations was to point out the influence of early age movements on the structural behavior of grouted connections. Another objective of the investigation was researching to evaluate different grout materials with the developed test system. The test series consisted of three grout materials. A cyclic test was carried out with the developed test facility. Furthermore, a static test and a reference test with formwork casted prisms were carried out to compare the results of each investigation. It has been shown that the part of the grout exposed to movement exhibits a reduced compressive strength in comparison to the lower part and to the static and the reference test. Furthermore, sedimentation of the larger particles (material “A” and “B”) could reduce the compressive strength of the material. Plastic deformations have also been observed. The investigation showed a different behavior between the different materials and a different influence of dynamic loads. The high-performance mortar grout (material “A”) showed a higher resistance to early age movements than material “B” and the OPC-grout (material “C”). To evaluate the impact of the mentioned effects on the load-carrying capacity of the structure, additional influences as type and stiffness of the structure, geometrical proportions and temperature during the grouting process must be considered. The impact of early age cycling effects depends on the fresh properties of the grout. For example grout materials with a low viscosity showed an influence of the effects of early age movements for example the sedimentation and segregation process. In this way it could lead to an impairment of the load-carrying capacity of the structure. However, the investigations also represent a grout material with a good performance and thereby only a minor influence of early age movements.

7 REFERENCES

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NEW STANDARD AND GUIDELINES FOR GROUTED CONNECTIONS

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Summary: The newly released Technical Note and offshore standard from the original legacies of Germanischer Lloyd (GL) and Det Norske Veritas (DNV) on the topic of grouted connections for offshore wind turbine foundations are presented. Additionally, the future of said publications within the newly merged legacy DNV GL is outlined.

1 INTRODUCTION

DNV GL, Renewables Certification is a newly merged division combining the global expertise of DNV KEMA and GL Renewables Certification. We are actively involved in driving the industry forward through development of standards and guidelines to ensure that the latest knowledge and best practice can be applied around the world.

Following the 2010 genesis of axial slippage of plain monopile grouted connection for offshore wind turbines, the industry has been working towards not only a remedy for the connection already in operation but also to establish new design guidance and standards rectifying the problem associated in particular with the large diameter plain piped grouted connections used in monopile foundations.

As a consequence existing standards and guidelines such as GL-IV-2 [2], DNV-OS-J101 [3] and NORSOK N-004 [4] have been reviewed and existing design approaches and calculation tools have been modified.

At the time GL was developing their own guidelines focusing on connections with shear keys as an alternative to the then common plain pipe connection.

DNV having at the time the J101 standard used in the design of existing plain pipe connections, promptly initiated an internal investigation into the background for the at the time valid industry standard on grouted connections in general, and followed suit with two Joint Industry Projects conducting large scale experiments on first plain pipe connections (JIP I), next on shear key connections (JIP II) from which new design standards have been developed.

DNV continually revised the J101 standard as knowledge from the JIPs materialized, ending up in the present newly revised May 2014 edition of J101, which introduces analytical design formulas not only for plain but also for shear key connections.
GL issued a new Technical Note for the Certification of Grouted Connections for Offshore Wind Turbines in December 2013 that utilizes a numerical approach to the design of grouted connections with shear keys.

Albeit not directly pertinent to grouted connections, present day status is however that the two companies DNV and GL have now merged into a new single company DNV GL, and that both of the original companies had on or about the same time issued new guidance on the design of grouted connections.

The present paper will in this light elaborate not only on the new design guidance and requirements set forth in each of the two publications, but also on the currently ongoing efforts of aligning and harmonizing the present GL and DNV design requirements into a unified set of DNV GL Standards and Recommended Practices for offshore applications of grouted connections.

2 BACKGROUND

The GL Technical Note for the Certification of Grouted Connections for Offshore Wind Turbines [1] and the DNV Offshore Standard DNV-OS-J101 [3], stems from different backgrounds and at the time different legacies, why they are addressed separately by various authors in the following.

a. GL Technical Note

The origin of the design recommendations for grouted connections given in oil and gas standards ISO 19902 and API is based on research work done in the 1970 to 1990s, e.g. Billington et al.[5].

Due to observed vertical settlements in grouted connections in offshore wind turbine support structures it became aware that existing knowledge in design of grouted connections does not explain the actual load bearing behavior under ultimate and fatigue loading conditions sufficiently.

Within the last years, additional research was initiated in order to investigate the particularities induced by the offshore wind turbines.

From 2006 to 2010, the GROW project at the Institute for Steel Construction, Leibniz University Hannover (LUH) in cooperation with GL, SIAG Anlagenbau Finsterwalde and Heijmans Oevermann investigated improving the strength of hybrid connections by applying shear keys and additionally compiling the relevant phenomena, which influence the bearing capacity of a grouted connection with shear key reinforcement under predominant bending in a scientific manner. More background information can be found in Schaumann et al.[6].

Germanischer Lloyd (GL) has been involved with the task to transform the scientific outcome into applicable requirements and rules for application by the offshore wind turbine industry. The project team reacted on the reported incidents with modifications on the test scope in order to avoid further failures. Furthermore experiences made during ongoing offshore projects are incorporated in this process, indicating if requirements are suitable to design process.
As result the Technical Note describes the current state-of-the-art for the application of grouted connections for offshore wind turbine support structures. With this Technical Note, GL wants to provide a contribution to summarize the existing knowledge which can be used as a basis for design and construction of grouted connections. It includes guidance on design calculation and structural details as well as requirements for the manufacturing, quality control, installation and monitoring.


For the designer, it should be clear that uncertainties exist for the time being which shall be covered by assumptions considered to lead to a safe design.

b. DNV Offshore Standard

As mentioned, the first reports of axial slippage in grouted connections occurred in early 2010 where 2009 autumn/winter inspection of the Egmond aan Zee wind farm in the Dutch part of the North sea revealed systemic subsidence of the transition piece relative to the monopile.

In response to this, DNV undertook on its own an initial in-depth review of the background (tests, standards, etc.) for the DNV-OS-J101 offshore standard for wind turbine foundations, which had been the governing standard for the Egmond aan Zee wind farm, following which DNV issued an amendment to the J101 that passive axial capacity of grouted connection where not to be relied on in larger diameter grouted connections for offshore wind turbine foundations. Further, DNV initiated a Joint Industry Project (JIP) where it and the industry came together and performed a test program specifically aimed a larger diameter grouted connections without shear keys for wind turbine foundations.

The outcome of this first JIP manifested itself in the 2011 Edition of the J101 with design requirements of axial capacity to be established without the use of any passive shear capacity existing between grout and steel in any plain pipe grouted connection. In lieu, parametric design formulas developed from the JIP where the introduction of a conical plain grouted connection was introduced as a possibility for ensuring axial capacity. Alternatively, elastomeric bearings were suggested as a method for cylindrical connections.

The use of traditional shear keys common in small diameter grouted connection within the oil and gas industry was in the 2011 Edition of J101 still cautioned against for large diameter monopile connections carrying offshore wind turbines, as the findings of the first JIP left concerns as to the validity of existing standards for fatigue in particular of large diameter grouted connection driven primarily by large dynamic moments as opposed to the traditional oil and gas application of small diameter connection where the dynamic fatigue load is characterized by being primarily axially and typically with a large overburden eliminating the possibility of load reversal in the grouted connection.

For precisely these reasons, a second JIP was undertaken as shear key connections was acknowledged as a potential competitor to the conical design. The aim of said second JIP was to establish design requirements and formulas for the use of shear keys in large diameter
monopiles grouted connections. The JIP was completed by the end of 2013, and its findings incorporated into the presently newest May 2014 Edition of the J101 standard.

As with the first JIP, the second JIP takes one more step forward in establishing an all-encompassing design standard for the use of grouted connection in foundation structures for offshore wind turbines. However, as was also the case with the previous JIP’s, limitations have been identified and highlighted for further assessment. In particular, the industry’s need to move into deeper waters making the special case of the jacket stabbed directly into pre-driven piles a strong contender to the traditional monopile, whereby the grouted connection is once again move back to relatively small diameter connections between typically four piles and the corresponding for jacket legs. But now with the new twist of pre-driven piles necessitating thick grout annulus uncommon to traditional oil and gas pile-sleeve connections, and the additional change from a top heavy, axially dominated fatigue loading with marginal bending and no load reversal to an offshore wind turbine loading condition that likely will provide both axial load reversals and significant local bending in the fatigue limit state.

Towards this issue DNV GL is presently seeking active participation in a JIP specifically for pre-piled jacket grouted connections, again with the aim of future inclusion in our design standards for offshore wind turbine foundations. The principal focus here is the implication of large nominal grout annulus to accommodate a preferable large installation tolerance.

3 NEW STANDARDS

c. GL Technical Note

The Technical Note is intended as helpful manual for the designer in order to simplify the design process of cylindrical grouted joints reinforced by shear keys as long as the topic grouted joints are subject of research. It illustrates the attitudes of loads and load distribution, and defines requirements for materials (steel and grout), the structural design parameters in general, manufacturing quality control, production and installation. Moreover a concept for evaluation of the bearing capacity is given for ULS and FLS loads.

Due to the small number of large-scale tests performed, the Technical Note is focusing on the applicability of numerical methods (FEA) for different types of grouted connections, in order to evaluate the bearing capacity of a grouted connection for ULS loads and FLS loads. Since the error rate might be high for a FEA with high non-linear features the Technical Note is intended to establish numerical methods as suitable design instrument. Therefore all non-linear effects and requirements for numerical simulation are introduced.

In order to keep the error susceptibility low the numerical analysis has to be combined with an analytical check. Also recommendations for FE-modelling and post-processing procedures are provided.

A special focus is set on the fatigue verification of the grout material. The long term fatigue behavior of high-strength grout material ($f_{ck} = 80 \text{ MPa} - 200 \text{ MPa}$) in a multi-axial stress state is still subject of research [7]. As a consequence safety factors have to be introduced to cover all uncertainties. Recent experimental investigations on the fatigue behavior of grout material [8] as well as of the grouted connection [9] in wet environment show a significantly lower fatigue resistance. For that reason the influence of water on the grout properties has to
be considered for grouted connections under water or in the splash zone. Based on the currently published test results, the Technical Note recommends determining the accordant fatigue strength of the grout by reducing the number of allowable load cycles by a factor of 10.

Special effects as crossing compression struts and contact respectively penetration of water are covered by safety factors referring to uniaxial behavior.

In order to keep the effort at a level which is assumed to be reasonable for the designer, different methods within a range from simplified to complex can be applied for fatigue verification. As usual, simplifications have to be paid by an increase of safety, but might make sense if the dimension of the grouted connection is not design driving. Otherwise, if certain parameters should be optimized, also a time consuming complex method might be of interest.

d. DNV Offshore Standard

As described in Sec. 2.b the DNV-OS-J101 offshore standard Sec. 9 on the design and construction of grouted connection for the use in foundation for offshore wind turbines have evolved over the last 4 years to encompass both plain and shear key designs for large diameter monopile foundations.

The latter, being provisions for the design of cylindrical large diameter connections with shear keys, being the principal update and expansion of the newly released May 2014 edition of the DNV-OS-J101 offshore standard. Additionally, specific design provisions for pile-sleeve grouted connections traditionally used with jacket structures and for stabbed jacket legs in pre-driven piles have been included in the 2014 Edition of J101, together with minor updates to the design provisions for plain conical connections.

Addressing the updates to the plain conical grouted connections first, operational experiences gained over the last couple of years with said conical connections has prompted an increase in the lower bound coefficient of friction between the steel and grout to now 0.7 together with an increased maximum allowable nominal interface contact pressure of now 1.5 MPa between the steel and the grout.

As a prelude to the addition of new design provision for the shear key connections – not only for large diameter monopiles, but also for traditional small diameter jacket-sleeve connections – the classification of ‘normal’ and ‘thick’ grout annulus has been changed from 100 to 150 mm based on the findings in the latest JIP. Additionally, a safety class for a reinforced grout annulus has been introduced, expanding the general applicability of the J101 to grouted connections insofar as handing design issues pertaining to large tensile stresses at grout body extremities and confinement / grout body integrity associated herewith.

The principle update is nevertheless the inclusion of parametric design formulas for cylindrical shear key connections based on the findings of the latest conducted JIP. Contrary to the design methodology for the plain connections being formulated in stresses, the shear key design provisions are formulated in forces to circumvent the complexity of in particular the grout stress locally at the shear keys.
Aspects of the stress approach is however maintained in the form of a maximum allowable (radial) contact pressure due to bending and transvers shear loading of 1.5 MPa directly akin to what is required for plain connections. This withstanding, the overall design of a shear key connection is in the new J101 attainable using formulas for ultimate capacity and a ‘S-N’ or rather ‘F-N’ fatigue damage curve formulated in force per shear key due to axial and bending loading.

As stated, the new J101 covers both the large diameter grouted connections typically associated with monopile foundations, as well as both the classical pile-sleeve connection for a jacket structure and the leg stabbed in pre-driven piles variety for jacket structures. This however within certain stated limitations.

Firstly, there are for all cases general limitation on the geometry of the connection, defining validity ranges for e.g. pipe slenderness, minima for shear key size and spacing, connection overlap length etc. Hardly being a novelty these will not be addressed in greater detail.

Instead, a couple of additional restrictions on the application of the 2014 Edition of J101 specific to shear key designs will be highlighted.

An important distinction is made pending on the connections exposure to bending loads. Here, knowledge gained for the JIP’s conducted on large diameter monopile foundations have led to the conclusion that for connections exposed to significant dynamic bending loads, it is unfavorable to have shear keys distributed over the entire height of the connection. This because of the relative sliding between the grout and the steel innate to the moment transfer from one tubular to the other in the plain condition would, with the introduction of shear keys throughout the height of the connection, cause a inevitably excessive focusing of loads on the first set of shear keys at top and bottom of the connection, which is considered detrimental to a robust design philosophy for a monopile design.

Keeping in mind that for a monopile design the primary use of the grouted connection is to transfer the bending load from transition piece to monopile, and that the requirements to axial capacity comparatively are small, for these types of connections the new J101 standard limits the placement of shear keys to the middle half height of the grouted connection.

For the traditional pile-sleeve jacket connection this requirement is waivered as these a predominantly axially loaded why said concern pertinent to large bending is generally insignificant. The same holds true for the design provisions formulated for the jacket leg stabbed into pre-driven piles. However, with the moderation that for this case, potentially – pending the actual design – bending loads may be significant, in which case caution should be exercised. Not only in the general use of the provided formulas, but also in the layout of shear keys within the connections.

Additionally for the stabbed leg configuration, should this be designed with an large allowance for installation tolerance in relation to the concentric placement of jacket leg and pre-driven pile, the implications of not only the associated very thick nominal grout annulus, but also the even larger min/max thickness ratio associated locally with large installation tolerance, will warrant careful consideration. This is in part the motivation for the previously announces third JIP initiative (GOAL), why pending its conduction and implementation in the
DNV GL design standards, it is advised to be cautious in the straight forward application of the design provisions of the present 2014 Edition of the J101.

4 WAY FORWARD TO DNV GL HARMONIZED STANDARD

The DNV GL merger prompted an immediate internal assessment of not only the how the GL Technical Note and DNV offshore standard related/compared to each other on insofar as the design approach for grouted connection where concerned, but also an internal assessment on how the standards, rules, and guidelines of the individual original legacies compared both to each other and the IEC 6400 requirements.

Through this process it has been determined that while promoting principally different design assessment methodologies – a primarily design formula based J101 versus a finite element analysis method – rather than being at odds the two standards complement each other while maintaining the same overall principal target reliability complying with that of IEC 6400.

In relation to the above it should be emphasized that although the J101 standard does provide parametric design formulas, it does not exclude the use of finite element assessment as a design tool, but actually recommends its use e.g. in the assessment of stabbed jacket leg in pre-driven piles with large installation tolerances.

Nevertheless, the J101 standard has been and remains very general in its recommendation for such analysis, why the new GL Technical Note aiming at being far more informative and detailed in its guidance on specifically these types of finite element assessments may been seen and used as an excellent source of guidance for design of grouted connections using numerical analysis.

In summary therefore, the present state of standard and guidance offered by the newly merged DNV GL is concluded to be a substantial step forward on the topic of grouted connection in offshore environments, with offerings ranging from simple parametric design formulas to advanced numerical analysis methodologies free to be applied either individually each on their own or as a complementary unified design approach e.g. for specialized optimized cases.

Within this realization lies also the clue to the future DNV GL harmonized standards.

It is the declared goal of DNV GL to have available to the industry a complete collection of unified standards and supporting recommended practices that by the end of 2015 will void any further use of the presently existing standards, rules, and guidelines from the old legacies.

In this process, the present aim of DNV GL is to maintain the present openness in design methodologies, i.e. keeping both a formula and numerical analysis approach, while keeping track and updating the standards in accordance with gained industry learnings on all relevant types of grouted connections for offshore applications.

Pertaining to the latter, DNV GL is currently actively participating in two research projects (GOAL and GROWUP) relating to the jacket stabbed into pre-driven piles. One with emphasis of fatigue of ordinary Portland cements (OPC’s) type grout, and one focusing also on OPC’s, but aiming more widely at establishing a complete design guideline for pre-piled jacket structures with large annulus of the grouted connection.
5 REFERENCES


ENERGY

New Standard and Guidelines for Grouted Connections

International Wind Engineering Conference
IWEC 2014 Leibniz Universität Hannover Germany

Lars P. Nielsen, Marc Mittelstaedt, Christian Ertel
2014-09-03
Outline

- Introduction – Company Merger
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  - State of Knowledge
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- Conclusion
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  - Future Harmonization
Renewables Certification – Newly Merged Division
DNV GL Group - Operating from September 12th 2013

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Mayfair
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Arnhem, Netherlands
Appr. 3000 employees

Business Assurance
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Appr. 2000 employees

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Offshore Wind – Combining DNV GL Competences

30 years of Type Certification of wind turbines

45+ years of Offshore Oil & Gas experience

Global leader in Project Certification of offshore wind projects
Grouted Connection – Offshore Applications

Transition piece
Brackets for temporary support before grouting
Grout
Monopile
Gaps and stresses caused by deformation in the transition piece and monopile*

*Please note that deformation and gap sizes shown have been significantly enlarged for illustration purposes only and are not true to scale to actual events.
Grouted Connection – Offshore Applications

- Grouted Connections used for wind turbine foundations within various application
- Fatigue loads as a design driver
- Monopile: predominant bending loading
- Large diameter piles
Grouted Connection – Offshore Applications

- More than 2000 offshore wind turbines so far
- More than 75% of them with monopiles
- Even more to be built in the next few years (22 GW under construction, or in planning)
- Slippage in monopiles first spotted in 2009

Grouted Connection – Available Design Standards

- GL Technical Note on Grouted Connections (2013)
- Annex K of NORSOK N004 for jacket structures (2013)
- ISO 19902, Fixed steel offshore structures (2007 + 2013 amendment)
**Grouted Connection – Design Parameters**

- Cross sectional geometry
- Grout thickness
- Pile thickness
- TP thickness
- Curvature, Diameters
- Grout compressive strength
- Shear key height and spacing or Cone angle
- Overlap Length
- Manufacturing imperfections
- S-N curve for grout material
- Load (combination, level)
- Early age movement
Grouted Connection – State of Knowledge

- Grouted Connection is still subject of research (GROWup, JIP-GOAL)
- Uncertainties are recognized to be:
  - Influence of large scale dimensions on bearing capacity
  - Determination of limits for ultimate and fatigue loads
  - Limited database with tests or measurements is available
  - Influence of surface conditions of tubular surfaces and shrinkage of grout
  - Effect of reversed loading
  - Effect of submergence underwater
  - Effect of combined axial and bending load
  - Effect of load level on FLS capacity
  - Effect of load history on FLS capacity
- Uncertainties are covered by sufficient high safety factors
Grouted Connection – Analytical Approach (DNV)

- Analytical approach based on test results, DNV-OS-J101 (May 2014)
- Extended to shear key connections
- Design limit on steel-grout contact pressure
- Geometric validity ranges imposed
Grouted Connection – Analytical Approach (DNV)

- Fatigue curve for shear key connection in a strut Force-Cycles formulation
Grouted Connection – Numerical Approach (GL)

- Approach GL TN Grouted Connections 2013
- Numerical approach based on Finite Element Analysis
- Only covering cylindrical type with shear keys
Grouted Connection – Numerical Approach (GL)

- Benefits of the FEA approach:
  - No limitation of application by ratio of geometric parameters
  - Cost efficient assessment of design variations (scale effects are circumvented)
  - Combined response delivers potential for optimization
  - Acknowledgement of state of the art of similar problems
    e.g. hot-spot interpolation according to IIW fatigue recommendation
    (IIW-1823-07:2008)

- Drawbacks of the FEA approach:
  - Special knowledge required in computational engineering
  - Time consuming application / computational intensive
  - Uncertainties due to hot spots and material behavior have to be evaluated by the engineer
Grouted Connection – Present Status and Future Harmonization

- Present Status of DNV GL Standards & Guidance
  - Original design standards exist independently from both legacies.
  - Both methods of design verification according to DNV or GL consider the same safety level and comply with safety level of IEC 61400 (principal target reliability).
  - A combination of both approaches is feasible.

- Future Harmonization of the Standards
  - The future aim of DNV GL is to make available to the industry a complete collection of unified standards and supporting recommended practices.
  - This is presently slated to be achieved by end of 2015.
  - New harmonized standard will remain in compliance with the IEC 61400 requirements.
  - Specifically for grouted connection – industry learnings will be continuously incorporated as it materializes from e.g. JIP’s and operational experience.
EXPERIMENTAL FATIGUE TESTS ON AXIALLY LOADED GROUTED JOINTS

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Keywords: Grouted Joint, Jacket, Support Structure, Fatigue, Submerged

Summary: Grouted Joints are a typical connection detail of substructures for offshore wind turbines. Nevertheless, at present merely little knowledge is available for the fatigue behaviour of this connection regarding a large grout annulus and submerged ambient conditions. This paper presents results from small and large scale fatigue tests of grouted joints. The small scale tests reveal a significant impact of the surrounding water. The large scale tests indicate that a larger grout annulus results in a different damage pattern.

1 INTRODUCTION

By the year 2030, 15 GW of Germany’s electric energy demand shall be covered by offshore wind farms according to the goals of the German government [1]. To achieve these goals about 30 to 40 wind farms with an overall amount of up to 3000 offshore wind turbines (OWT) need to be erected within the next 15 years. These offshore wind farms are mainly located in the German Exclusive Economic Zone (EEZ) of the North Sea. As the Maritime Spatial Plan for the North Sea [2] shows, most of these areas have water depths of 30 m and deeper. For these large water depths the preferred solution is currently a lattice substructure [2], the jacket (cf. Figure 1).

Figure 1: Grouted joints in a jacket substructure for OWTs and detail of grouted joint
Peter Schaumann, Alexander Raba and Anne Bechtel

A group of steel piles, which are driven into the seabed, forms the foundation for lattice substructures. Piles and substructure are connected via grouted joints, a connection well known from the oil and gas industry. The connection consists of an inner steel tube (pile) and an outer steel tube (sleeve) creating an annulus, which is filled with high performance grout (cf. Figure 1). For a reliable force transmission between steel and grout, the facing steel surfaces are equipped with shear keys made of weld beads.

Compared to grouted connections for Monopiles, the grout layer thickness in jackets is much larger and the connection is prevailingly axially loaded. Alternating loads from wind and waves are the decisive actions. The high alternating loads, the geometric variations and high strength filling materials differentiate the connection for OWTs from the ones utilized in oil and gas platforms. As a result, use of knowledge and experience from oil and gas for OWT grouted joints is limited and has to be verified. In addition, latest experiments and research investigations focusing on grouted connections for OWTs did not consider the effect of large grout annulus and submerged conditions. Hence, current research investigations concentrate on the fatigue behaviour of grouted joints with large grout annuli.

Therefore, the research project ‘GROWup – grouted joints for Offshore Wind Energy Converters under reversed axial loadings and up scaled thicknesses’ (funding sign: 0325290) funded by the German Federal Ministry for Economic Affairs and Energy (BMWi) deals with the fatigue behaviour and execution perspectives of grouted joints in jackets. The joint project is conducted at the Institutes for Steel Construction and Building Materials Science at the Leibniz University in Hannover, Germany. The Institute for Steel Construction investigates the fatigue behaviour of small and large scale grouted joint specimens. In this paper results of the conducted tests are presented.

2 SMALL SCALE TESTS

To investigate a broad range of different filling materials and the influence of various loading parameters at an economic level a small scale grouted joint specimen (cf. Figure 3, left) was developed [4]. The specimen is equipped with machined shear keys and comparatively stiff steel tubes. As a result, the decisive damage mode is a failure of the grout matrix in the occurring compression struts. Moreover, the specimen is not really to scale of grouted joints in real substructures. Nevertheless, the grout’s stress state in the small scale specimen is comparable to the multiaxial stress state in real connections.

The test procedure for small scale specimens is composed by two major parts. The first part is an Ultimate Limit State (ULS) test in which the specimen’s quasi static resistance $F_{ULS}$ is determined. The specimen is compressed in a displacement controlled test rig with a load application speed of 0.2 mm/min (cf. Figure 2, right) beyond its maximum resistance. The average $F_{ULS}$ of three specimens is chosen to be representative for a batch of specimens produced at once with the same filling material.

Figure 3, top left, shows the force displacement behaviour of test specimens under quasi static compression. The behaviour can be separated into three stages. Up to about 50 \% $F_{ULS}$ (cf. P1) the connection shows a linear elastic behaviour. Then the first cracks due to tensile stresses transverse to the compression struts occur and cause a stiffness reduction in stage 2. Finally, the static resistance $F_{ULS}$ is reached at P3 and after that the displacement increases while the bearable load reduces in stage 3.
Within the second part of the test setup specimens are tested regarding their fatigue behaviour. Within these tests the ambient condition (AC) is varied between dry and wet. Where wet means, that the specimen is loaded while being fully submerged in a water basin. Also, the maximum load level \( F_{\text{max}} \) is varied between 50% and 25% of the static resistance \( F_{\text{ULS}} \). And finally, the loading frequency \( f \) is varied between 5 Hz, 1 Hz and 0.3 Hz. Since real support structures are mainly loaded at their first eigenfrequency of about 0.3 Hz [5], the influence of the loading frequency on the connection’s fatigue behaviour is investigated additionally. All tests are conducted until \( N = 2 \) m load cycles or failure of the specimen.

Figure 3: Force displacement behaviour of three specimen under quasi static compression (top left), load cycle dependent displacement \( u \) of small scale specimens for different ambient conditions (AC) (top right), different load levels \( F_{\text{max}} \) (bottom, left) and different loading frequencies \( f \) (bottom, right)
Peter Schaumann, Alexander Raba and Anne Bechtel

Results from several fatigue tests are presented in Figure 3. The major influence of the varied parameters described before can be attributed to the ambient condition. As Figure 3, top right shows, testing under water reduces the number of endurable load cycles to $N \approx 35,000$, while the specimen in dry ACs passes the load cycle limit at $N = 2 \text{ m}$. (cf. Figure 3, top right). A halving of $F_{\text{max}}$ (cf. Figure 3, bottom left) leads to an increase of endurable load cycles from $N \approx 100,000$ to $N \approx 1 \text{ m}$. The influence of $F_{\text{max}}$ can be reduced if the number of endurable load cycles is defined as the first point of a major stiffness decrease. For the results presented, this can be defined at $N \approx 200,000$. A reduction of the loading frequency $f$ also leads to an increase of endurable load cycles from $N \approx 35,000$ at $5 \text{ Hz}$, $N \approx 100,000$ at $1 \text{ Hz}$ to $N \approx 150,000$ at $0.3 \text{ Hz}$ (cf. Figure 3, bottom right).

During the submerged tests processes of grout material flushing were observed (cf. Figure 4, left). As a result the volume of the grout section reduces and the stiffness of the connection decreases like it is visible in the displacement plots (cf. Figure 3). After the specimens were tested, they were opened to investigate the grout section. Figure 4, right, shows the grout section of a specimen tested in dry and one tested in wet ambient conditions. Both specimens were loaded identically. While in dry ambient conditions no effects of attrition are visible, the submerged specimen shows significant cracks due to transverse tensile stresses in the lower part, as well as vertical cracks in the upper part at the pile surface.

On basis of the presented results, three different processes can be addressed as influence of the water to the connection’s fatigue behaviour. At first, deformations of the loaded specimen lead to an opening of the contact interface between steel and grout. As a result water invades the contact interface and as the first degradation process hydro lubrication reduces the friction between steel and grout [6]. Due to high compressive stresses caused by the local load application of the shear keys, grout material in front of the shear keys crushes and resides as loose material in the connection. In the second process this loosened grout particles get washed out by the invading water [7]. And finally, as a third process, high water overpressure and flow speeds caused by pumping effects in the interface due to the cyclic loading, lead to cracking and wear of the grout material. This leads to a significant stiffness reduction and a decrease of the number of endurable load cycles of the specimen.
3 LARGE SCALE TESTS

In order to quantify the effect of large grout thicknesses to the fatigue behaviour of axially loaded grouted joints large scale fatigue tests were conducted. With reference to real jacket and tripod dimensions, two test specimens were developed, cf. Table 1. The grout layer thickness of test specimen no 1 and 2 are varied between ~82 mm and ~184 mm. Both specimens were equipped with five shear keys on pile and sleeve. For the filling material a high-performance grout with a nominal compressive strength of 140 MPa was chosen.

Both specimens were tested in a 10 MN large servo-hydraulic testing machine (cf. Figure 5). The major testing was performed by applying six different load levels, each for 100,000 load cycles. In the first three load levels an alternating load \((R = -1)\) was applied, which was followed by three load steps of pulsating-compression forces \((R = \infty)\). As the first specimen did not show significant displacement changes or stiffness reduction after completion of the major test program, two additional load steps were applied: load step no 7 and 8. These load steps were used to analyse the fatigue and deformation behaviour of preloaded and predamaged specimens. Therefore it was sufficient to apply 15,000 load cycles. A test frequency of 1-2 Hz could be realized in all load steps, which are summarized in Table 2.

Table 1: Geometric dimensions of large scale grouted joint test specimens

<table>
<thead>
<tr>
<th>Test specimen no</th>
<th>Scale (D_p/t_p \text{[-]})</th>
<th>Pile (D_s/t_s \text{[-]})</th>
<th>Sleeve (D_s/t_s \text{[-]})</th>
<th>Grout (D_s/t_s \text{[-]})</th>
<th>Overlap length (L_g \text{[mm]})</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>~1:4</td>
<td>~24</td>
<td>~41</td>
<td>~10</td>
<td>1240</td>
</tr>
<tr>
<td>2</td>
<td>~1:2</td>
<td>~16</td>
<td>~41</td>
<td>~4</td>
<td></td>
</tr>
</tbody>
</table>

For predominantly axially loaded grouted joints different failure modes are known. Substantially, it can be distinguished between the two material relating modes for steel and for grout. Shear key failure and local plasticization of the steel shell at the shear keys are typical steel failure modes that are covered by steel design approaches.

Table 2: Load level and related maximum and minimum forces of applied test program

<table>
<thead>
<tr>
<th>Load level</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>(F_{max} \text{[MN]})</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>3</td>
</tr>
<tr>
<td>(F_{min} \text{[MN]})</td>
<td>-1</td>
<td>-2</td>
<td>-3</td>
<td>-4</td>
<td>-5</td>
<td>-6</td>
<td>-8</td>
<td>-3</td>
</tr>
</tbody>
</table>

In contrast, the grout failure modes being more sensitive to fatigue loading, especially regarding tension forces, display more complex multiaxial failure behaviour. An initial state
of grout cracking (cf. Figure 6a)), reflects the first stage of failure considering a shear keyed grouted connection. The grout matrix failure in the local region around the shear key is influenced by arising stress concentrations and exceedance of the grout tensile strength. Rising stresses lead to compression strut failure (cf. Figure 6b)), on the force flux between facing shear keys. This failure mode is considered by design formula within current design standards, e.g. ISO 19902 [8]. Failure mode c) (cf. Figure 6) reflects a shear failure which is influenced significantly by the shear key spacing s.

With regard to experimental tests and results presented in literature, the major influencing parameters are the shear key height h, the shear key spacing s and the uniaxial compressive strength $f_{cu}$ of the grout. These parameters are considered by empirical equations for the Ultimate Limit State design according to ISO 19902 [8], cf. Eq. 1 and 2.

$$f_{g,shear} = 0.75 + 1.4 \left( \frac{h}{s} \right) f_{cu}^{0.5}$$

$$f_{g,sliding} = C_p \cdot 2 \cdot 140 \left( \frac{h}{s} \right) ^{0.8} \cdot f_{cu}^{0.3} \cdot K$$

with

$$K = \left( \frac{D_p + D_s}{t_p + t_s} \right) ^{-1} + \frac{E_g}{E_s} \left( \frac{D_s}{t_g} \right) ^{-1}$$

In these equations, h is the shear key height, s the shear key distance, $f_{cu}$ the compressive strength of the grout material, D the diameter, t the thickness, and E the young’s modulus of grout ($E_g$) and steel ($E_s$). Subscript s represents the sleeve meaning the outer tube, p the pile as the inner tube, and g the grout layer. The first equation considers a grout matrix failure reflected by the grout cracking and compression strut failure depicted in Figure 5. The shear failure also named as sliding failure being depending on the radial connection stiffness is reflected by the second equation. Further developments to these equations for the Ultimate limit State design are presented in [9].

According to these design equations for the tested specimens a compression strut failure might occur due to the utilised geometric dimensions.

This initial indication of failure mode correlates to the cracks inside the tested specimens. Figure 7 shows the opened test specimens no 1 and 2 after completion of the testing procedure. Both damaged specimens indicate crossing compression strut cracks caused by tension and compression loads. In addition to these cracks a wedge of crushed grout appears on the load averted side of the pile shear keys. This grout wedge, primarily described by Krahl & Karsan [10], has an average height correlating to the shear key height and a length which is four-times larger than the shear key height. Comparing the different damaged grout cross-sections of test specimen no 1 and 2 depicts that the large
grout thickness indicates a different damage pattern than the smaller grout layer, even though for both specimens crossing compression struts appear. For test specimen no 2 larger compression struts occur by skipping shear keys on the opposing side. The larger grout annulus and the smaller pile diameter influenced not only the damage pattern but also the applicable loads. Contrary to the test specimen no 1, it was not possible to apply the loads of load step 7 and 8 to test specimen no 2, after application of the major test steps 1 - 6. This implies that the test specimen no 2 has a reduced fatigue capacity compared to test specimen no 1. Further analyses of measured displacements and strains in combination with numerical simulation will improve the knowledge about the failure procedure and the fatigue behaviour of grouted joints with large grout annuli.

4 CONCLUSIONS

In this paper, results of small and large scale specimen tests of axially loaded grouted joints are presented. The small scale specimen fatigue tests show a significant impact of water on the connection’s fatigue behaviour. As mentioned, the small scale specimens are not real to scale and therefore, the presented results cannot be directly transferred to real connections. To overcome this uncertainty, submerged fatigue tests with large scale specimens are planned for the future.

Large scale fatigue tests have shown that differing grout thicknesses and pile diameter influence the damage pattern and fatigue capacity. The crack paths of the opened specimens have shown that compression struts for the large grout thickness skip opposing shear keys. Both test specimens have shown crossing compression struts which are induced by tension and compression loading. Investigations of measured data and numerical simulations are planned to comprehend the failure procedure and crack mechanism.

Figure 7: Damage pattern of opened tested large scale specimen no 1 (left) and no 2 (right)
ACKNOWLEDGEMENT

The presented experimental tests and results are achieved within the research project ‘GROWup - Grouted Joints for Offshore Wind Energy Converters under reversed axial loadings and up scaled thicknesses’ (funding sign: 0325290) funded by BMWi. The authors thank the BMWi for financing and all accompanying industry project partners (Fraunhofer IWES, GL, Senvion, RWE Innogy, Strabag) for their support. A special thanks goes to the Institute for Building Materials Science, in particular to its head Prof. Ludger Lohaus, for the fruitful cooperation within the project especially with regard to the use of the testing facility for the large scale test specimen.

REFERENCES


EXPERIMENTAL TEST PROCEDURES TO SIMULATE THE IN SITU ASSEMBLY OF GROUTED JOINTS

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Keywords: Grouted Joint, Grout Material, in situ Material Properties, Mock-up Tests, Offshore Wind Turbines, Test Facilities

Summary: Large-scale test facilities for the simulation of offshore filling processes for the in situ assembly of grouted joints is presented. Furthermore, results of the first filling test in the Large-Scale Test Facility were presented and discussed. This filling test shows the possibility of an in situ assembly without negative influences on the macroscopic structure of grout in grouted joints. It also shows that the compressive strength of the test wall was not entirely homogeneous. Minor inhomogeneities of the grain size distribution around the shear keys and high temperature inside the wall caused by the hydration process were observed. In addition, the new Underwater Test Facility for filling tests, which is under construction, is presented. This will be the most realistic filling test in a research facility possible up to now.

1 INTRODUCTION

The foundations of offshore wind turbines are typically anchored into the seabed with driven piles. The upper part – the substructure – is connected to the piles by using grouted joints. The ability to compensate imperfections of the pile driving process is the benefit of this kind of connection.

Grouted joints are tube-in-tube connections with tubes of different diameters. The gap between outer (sleeve) and inner tube (pile) is later filled with a specialized high-performance concrete, mortar or ordinary Portland cement (OPC-grout) – the grout. In general, high-strength grouts are used for grouted joints of offshore wind turbines [1][2] in Germany. The loads from the tower are transferred by compression struts from the sleeve to the pile, as supposed in [3].

The compressive strength of the grout has an important influence on the load-bearing behavior of the connection. It is known from small-scale tests of grouted joints that the compressive strength has a significant influence on the load-bearing capacity of grouted connections [4]; for example, a 40% reduction of the grout compressive strength can lead to a drop of the ultimate load of more than 50% [5].

The in situ assembly of grouted joints is challenging due to the harsh offshore conditions and because of the exposure of grouted joints [6]. It is nearly impossible to check the material properties of grout inside a grouted joint after the filling process. Thereby, grouted joints
represent a “black-box” which does not include any information about the *in situ* material properties of the grout.

Depending on the kind of substructure, further circumstances must be considered which could have a negative influence of the material behavior of the grout.

![Jacket substructure (left side); sketch of a jacket substructure and grouting line to the grouted joint (right side)](image)

Figure 1 shows a jacket substructure as an example of a substructure. The challenges of the filling process in this example are long grout lines, high pumping pressures and the exposure of the grouted joint underwater under low ambient temperatures.

It is necessary to verify the assumption that the complex *in situ* assembly, including the offshore conditions, mixing and filling process, has a minor influence on the material properties of the grout.

In addition to the uncertainties caused by the *in situ* assembly offshore mentioned, further risks exist which could have an negative influence on the material properties of the grout. Examples of these risks are the segregation and sedimentation of the highly flowable and self-compacting material because of the height. Gravity could lead to a progressive accumulation of the larger particles in the lower part, while the cement paste remains in the upper part of the
construction. This could have an impact on the hydration temperature and the autogenous shrinkage in the upper part of the construction, leading to a higher tendency of cracking.[7]

Research projects such as “Überwiegend axial wechselbeanspruchte Grout-Verbindungen in Tragstrukturen von OWEA (GROWup)” (BMWI, Ref. No. 0325290) and “Probabilistic Safety Assessment of Offshore Wind Turbines (PSA)” (MWK, Support Code GZZM2547) deal with the influence of the in situ assembly and the materials properties of grout in grouted joints.[8]

Certification bodies, e.g. DNV-GL, also expect a negative influence of the in situ assembly of the grout. The new technical note [9] reduces the compressive strength of the grout material and sets an additional including allowance (\(\gamma_{\text{offshore}}\)) to take the influence of the in situ assembly into account.

Test facilities are needed for the evaluation of these uncertainties. A test facility for filling tests on a large scale is presented below.

2 LARGE-SCALE TEST FACILITY

Based on a downscaled test facility (see [10][11]), the Large-Scale Test Facility was developed within the research project GROWup. This test facility consists of a watertight transparent formwork, a tilting device for secure sampling of test specimens, a pumping unit, and a mixing plant similar to mixers used offshore. The total height is 4.9 m, with the height of the formwork being 3.75 m. The width is 3.3 m and the size of the gap is flexible, from 5 cm up to 20 cm.

![Filling test in the Large-Scale Test Facility](image)

Figure 2: Filling test in the Large-Scale Test Facility
Temperature sensors, pressure gauges and a thermographic camera collect data of the filling and hardening process. The formwork can be tilted horizontally to dismantle the transparent front panels, take test specimens and cut slices of the wall to determine inhomogeneities and segregations over the full height of the wall. Nearly realistic filling tests can be conducted in this test facility due to the dimensions and the mixing and pumping equipment. Detailed information is given in [12]. Figure 2 shows the test facility during a filling test.

3 RESULTS OF THE FIRST FILLING TEST

A typical grout material for offshore use was used for the filling test in the Large-Scale Test Facility. The gap between the front panels and the back panels was adjusted to approximately 10 cm and the formwork was filled with water. The length of the grout hoses was 13 m with a diameter of 2 inches. The inlet was placed in the middle of the formwork at a height of 10 cm from the bottom. A high-strength lubrication mix was used prior to grouting. Table 1 shows the fresh grout properties and the air temperature.

Table 1: Fresh grout properties and air temperature

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air temperature:</td>
<td>25 – 28°C</td>
</tr>
<tr>
<td>Fresh grout temperature:</td>
<td>33 – 35°C</td>
</tr>
<tr>
<td>Slump-flow on basis of:</td>
<td>770 – 850 mm</td>
</tr>
<tr>
<td>Air content:</td>
<td>2.9 – 3.5%</td>
</tr>
<tr>
<td>Density:</td>
<td>2380 kg/m³</td>
</tr>
</tbody>
</table>

The grout material shows almost self-leveling characteristics over the whole period of filling, which is shown in Figure 2. A maximum pump pressure of about 7 bar was reached and a maximum pressure of 1.9 bar inside the formwork during the filling process. Temperatures during the hydration reached a maximum of 57°C in the test wall. After dismantling the front panels, the observation of the surface showed no big faults, no cracks and no obvious defects in the structure, as shown in Figure 3 on the right side. The formwork was turned horizontally by the tilting device to drill cores from the test wall after two days of hardening. These cores were tested after seven days to determine the compressive strength.

Figure 3 shows the compressive strength after seven days (on the left side) as a percent of the reference cubes. The reference for the compressive strength is a set of three 75 mm cubes with a compressive strength of 117 N/mm² after seven days. The compressive strength of the test wall is invariably higher than the reference test specimens. In this case, the results vary from 112% up to 134%.
The sawn profiles of the test wall show minor inhomogeneities in the distribution of the larger grain size. No significant sedimentations or segregations were observed over the height of the wall. The shear keys were predominantly fully enclosed by grout. Inhomogeneities in the grain distribution were observed around the shear keys. Figure 4 shows exemplarily on the left side, a profile with an accumulation of large grains around the shear key. The profiles on the right side show, contrarily, an increased content of fine material around the shear key.

Figure 3: Results of the filling test – compressive strength compared to reference cubes in percent (left side), dismantled test wall (right side)

Figure 4: Sawn profiles of the test wall in the area of the shear keys
4 CONCLUSIONS

The new Large-Scale Test Facility shows its suitability for filling tests with grout materials.

The first filling test (reference test) in the Large-Scale Test Facility shows an obviously homogeneous test wall without cracks. This illustrates that the undisturbed in situ filling process of the formwork generally has no negative influences on the macroscopic structure of the test wall. No large faults or wide and large cracks were observed by using the material tested.

All cores drilled had a higher compressive strength compared to the reference cubes, but the compressive strength is not entirely homogeneous. There is a variation in the compressive strength of 26 N/mm². The high temperature inside the wall has to be taken into account for the evaluation of the compressive strength after seven days. This may lead to the higher compressive strength of the wall compared to the reference cubes, where this influence is not given. However, the high temperature has to be taken into account for the design of the grouted joint. High temperature while the grout is hardening could lead to cracks. Moreover, high temperatures while hardening could also have a negative influence on the durability of the grout. A secondary formation of ettringite causing an expansion could damage the structure of the grout.

The sawn profiles of the test wall show no significant inhomogeneities outside the areas of the shear keys. A partially non-homogeneous grout structure was observed within this highly stressed area. An accumulation of fine materials, which was observed around the shear keys, could change the material properties of the grout in this area. A lower elastic modulus, higher tendency to crack due to higher autogeneous shrinkage and different compressive strengths are possible. This new phenomenon will be investigated more deeply.

The pressure inside the formwork shows that a hydrostatic pressure on the grout seal is possible. Higher pressure on the grout seal is possible because of higher pumping pressure offshore combined with the high viscosities of the grout.

The results of the first test acts as a reference for following tests which will simulate disturbed filling processes.

Downscaled tests in the Laboratory Test Facility have shown significant reductions of the compressive strength simulating disturbed filling processes [13]. Inhomogeneities of the grout structure and cracks in the test specimens were also observed.

5 OUTLOOK

More information about the influence of in situ assembly on the material properties of grout is needed for an optimization of grouted joints and a possible reduction of safety factors in the design. Tests with different materials in the test facilities are ongoing within the research project GROWup. Furthermore, tests on the impact of disturbed filling processes will be conducted. The influence of a water-filled formwork is possible with the test facilities presented, and an Underwater Test Facility is under construction. Figure 5 shows a sketch of this facility. The flowability of the fresh grout could be simulated more realistically with the length of the formwork being about 7.20 m. Heights up to 8.1 m are also possible.
This formwork is especially suitable for filling tests with OPC-grout because of the steel formwork panels and the exposure under water for a realistic transport of the hydration energy. Downscaled tests with OPC-grouts have shown temperatures of more than 100°C due to the hydration of the grout. Large cracks in test specimens were also observed. If and how OPC-grouts are applicable for offshore wind turbines could be evaluated in the Underwater Testing Facility under almost realistic conditions.

6 REFERENCES


Ludger Lohaus, Michael Werner and Dario Cotardo


Optimized Soil-Model to Derive the Pile Driving Fatigue for Offshore Piles Using Drivability Predictions

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Keywords: Support Structures, Offshore foundation pile, Installation, Drivability prediction, Friction fatigue, Pile driving fatigue

Summary:
During the last years, the authors were involved in varying (commercial and research) projects concentrating on the described optimized soil-model to derive the pile driving fatigue for offshore piles under the use of drivability predictions. After a short introduction general background information and recommendations for pile driving prediction analysis of impact driven open-ended steel pipe piles are given. Main topics are the stress wave propagation through a pile, the amount of energy introduced into the pile by the hammer-anvil-cushion-pile setup and the change of the most important soil parameters during pile driving (skin friction reduction). Following a short description of the used simplified pile driving fatigue analysis approach used, a fictitious example of a numerical offshore pipe pile installation is shown for different pile driving simulation approaches. The result shows the sensitivity in changing different parameters with respect to the pile driving fatigue calculations.

1 Introduction
It is well known that in the next few decades a large number of offshore wind farms will be installed in the German North and Baltic Sea as well as in many other countries all over the world. To transfer the forces from the wind energy plant into the subsoil, large diameter open-ended steel pipe piles are used in general. The installation of the supporting piles is usually performed by a hydraulic hammer [10]. During impact driving, the pile is subject to high forces and stresses for each hammer blow, causing damages in the structure that have to be considered in terms of operating life (20 to 25 years) calculations. Commonly, the pile driving fatigue calculations are conducted by use of a pile driving prediction (PDP) software as for example Allwave-PDP [11] or GRLWEAP [13]. For every hammer impact, the software calculates the maximum tension and compression stresses at each pile element. Summing up the stresses for each blow and at each pile element, the pile driving fatigue can be calculated using e.g. the linear cumulative damage approach of Palmgren & Miner [12], as suggested for example in the GL Guidelines [6].
In the past, the soil model used for the drivability predictions or drivability back calculations was rather simple. For purposes of simplification it was generally assumed, that the resistance of the soil remains constant during pile driving, even though the skin friction resistance changes during impact pile driving (caused by cyclic shearing between pile wall and soil). To improve the quality of the drivability results and thus the pile driving fatigue results, on request of Fichtner Water & Wind GmbH (FWW), Hamburg, Pile Dynamics, Inc. (PDI) and Allnamics BV, The Netherlands, applied a new/updated approach into their software taking into account the changing soil resistances during pile installation (friction fatigue). Using this new approach, the soil resistance to driving (SRD) of each soil element will be recalculated / changed in accordance with the distance between the element and the pile toe (number of shear cycles). So, the skin friction is recalculated for every numerical simulation or time step. For the skin friction reduction a mathematically varying approach related to [1] or [7] can be used for example.

2 BACKGROUND INFORMATION AND RECOMMENDATIONS FOR PILE DRIVING PREDICTIONS

To calculate the pile driving fatigue by available simulation software and as accurate as possible, large experience in terms of the used software (numerical model), the geo-mechanical behavior of the soil under dynamic influence or the stress wave propagation through a pile is necessary.

In the following sections, important phenomena will be described occurring during pile driving, pile driving analysis and pile driving prediction with respect to the pile driving fatigue calculations.

a. Stress wave propagation through the pile

Figure 1 shows exemplarily the result from the simulation of a single hammer blow introduced into a tubular steel pile with a length of 45 m. Figure 1 a (left side) shows the tension stresses (dotted red line) and the compression stresses (black line) over the length of the pile (blue) at a pile penetration of 5.0 m. Figure 1 b (right side) shows the same result at a penetration depth of 30.0 m. For both results the same hammer energy (Eₗ) and the same efficiency hₜ was used.

Figure 1: Maximum compression and tension stresses over the pile length caused by a hammer impact.
As can be seen at the beginning of driving, the tension forces are very high, leading to a large stress range and therefore causing high pile damage (fatigue). With further pile penetration the tension forces and accordingly the stress ranges reduce significantly, causing less pile damage per single blow. The reason for the shown result is the damping of the stress wave by the surrounding soil. At the beginning of driving the downward directed stress wave (under compression) propagates almost without damping towards the pile toe. If the soil below the pile toe is rather loose or soft (free pile toe), the stress wave will be reflected mainly as tensile stress wave, causing the shown result. With further penetration and higher soil resistances, the majority of the stress wave will dissipate into the surrounded soil during propagation through the pile. Therefore, tension stresses from reflection will be a lot smaller. Additionally, when the soil below the pile toe is very dense or very stiff (fixed pile toe) the downward traveling stress wave will be reflected as stress wave under compression as well. Therefore, to avoid high pile damage during driving, the hammer energy should always be selected in accordance with the subsoil conditions and the penetration depth on site.

b. Energy Problem
The hammer energy introduced into the pile for every single blow is the most important value for pile driving fatigue calculations. In general, a pile driving prediction software does have an internal hammer database, where the most common hammer types can be selected. Besides the hammer's kinetic energy, for a precise pile driving fatigue calculation, the efficiency of the whole system (hammer – anvil/follower – cushion – pile) has to be known as accurate as possible. For example, manufacturers of 'new generation' hydrohammers confirm efficiencies of ≥ 95%. However, lower values were measured/calculated on site as well. Therefore, preferably the efficiency should be determined on site and for each pile driving system. To determine the hammer’s efficiency during impact pile driving two different energy values have to be known. At first, the hammer’s kinetic energy $E_i$. Using 'new generation' hydrohammers, the kinetic energy is monitored by an internal monitoring unit and available in real time. Secondly, the transferred energy $E_t$, as the energy introduced into the pile head has to be known. The transferred energy can be calculated when pile driving analysis (PDA) will be performed during pile installation (by the integral of the measured/calculated force ($F_t$) and velocity ($v_t$) over time). Using both energy values, the transferred efficiency $\eta_t$ as the energy loss between the kinetic energy and the transferred energy can be calculated using the following equation:

$$\eta_t = \frac{E_t}{E_i} \quad (1)$$

The described calculation of the transferred energy $E_t$ form the PDA results however is not always valid and correct, as the transferred energy will normally be calculated by the sum of the downward ($\downarrow$) and upward ($\uparrow$) traveling stress wave over time. For correct calculation of $E_t$ it has to be guaranteed that only the downward traveling particles (and only from the hammer impact) will be taken into consideration. Any reflected particles are already part of the energy calculations and should not be considered a second time (see Figure 2). Depending on the size of the reflected particles (stress wave), the length of the pile, the pile penetration, the piles cross-section, the sensor level below the pile head, the contact time of the hammer etc. the reflecting particles may have a significant influence on the calculated energy, leading to an under or over prediction of $E_t$. 
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Therefore, the authors strongly suggest calculating the transferred energy $E_t$ only by the downward traveling parts of the stress wave and at the beginning of impact pile driving. At the beginning of pile driving, generally the distance between sensor level and soil level is larger than half of the introduced stress wave length. Therefore, overlapping of downward and upward traveling parts is rather unlikely (constant pile cross-section). Further information about the described topic can be found for example in [3].

Additionaly, to simulate a pile installation as accurately as possible, detailed geotechnical knowledge about the mechanical behavior of the soil next to the pile is necessary during installation. This relates in particular to the unit skin friction as it changes significantly during pile installation. Therefore, in any case of "geo-mechanical correctness" during pile driving simulation, the shaft friction has to be recalculated at each penetration level during numerical simulation.

Figure 2: Calculation of the transferred energy

c. Friction Fatigue

Vesic [15] presented one of the first publications dealing with the friction fatigue phenomenon. Lehane [9], Fugro [4], White et al. [16] and Gavin et al. [5] performed comparative investigations. All of the investigations confirmed the Vesic findings which are that the shaft friction at a defined depth below ground reduces with increasing penetration. These studies showed that the unit shaft friction reaches a maximum or $SRD_{max}$ at the time the pile toe passes
the soil element under consideration. With further driving progress or advancement of the pile, the horizontal stresses acting on the pile wall reduce towards a minimum or residual shaft friction resistance value ($SRD_{\text{min}}$) and remain at this value until pile driving ends. Figure 3 (adopted from [14]) illustrates the propagation of the unit skin friction at a single element during pile installation.

3 PILE DRIVING FATIGUE CALCULATION

Nowadays offshore wind turbines and their foundations are designed for an operating life cycle of 20 to 25 years. During that time, wind-, wave- and turbine loads act on the whole structure, causing continuous bending fatigue. To ensure that the pile foundation can withstand the wind turbines’ operational life, the fatigue from pile installation has to be added up with the fatigue from the operational life (overall fatigue proof).

In general, the circumferential pile welds with or without a change in wall thickness are the most fragile steel pile details, determining the overall pile fatigue. According to S/N curves the reference stress is 90 for circumferential pile welds without grinding and 112 for circumferential pile welds, ground flush with the base metal. Following the GL Guidelines [6], the pile driving fatigue $D$ commonly is calculated using the simplified fatigue analysis suggested by Miner [12] (equation 2). Using the nominal stress concept, small inaccuracies in the weld geometry or small joint offsets were taken under consideration during the development of the reference S-N curves [8], [2].

$$D = \sum_{i=1}^{l} \left( \frac{n_i}{N_i} \right) \leq 1$$  \hspace{1cm} (2)

The expected total pile driving fatigue $D$ is calculated by the sum of the quotients of each number of stress cycles in an area of stress ranges $n_i$ and its number of tolerable stress cycles $N_i$. The number of stress cycles in an area of stress ranges $n_i$ is calculated by the PDP software for each pile segment (e.g. per meter) and for each hammer blow. The area of stress ranges $\Delta\sigma_i$ is generally $\leq 10$ MPa.

4 EXAMPLE

In the following example, the pile diving fatigue was calculated by pile driving simulation results and in accordance with section 3. The pile specifications are the same as used for Figure 1: Tubular steel pipe pile, total pile length 45.0 m, final penetration depth 30.0 m, constant wall thickness 40 mm, outer diameter 3.0 m. The soil was simulated as sand with a linear increase in total skin friction (inner and outer skin friction) from 0.0 kPa at the surface up to 350 kPa at a depth of 30.0 m below surface. A similar increase was defined for the toe resistance. Starting at 0 kPa at the surface towards 30000 kPa at a depth of 30.0 m. For the dynamic parameters (damping and quake) standard values were selected.

For the pile driving fatigue simulations, the following assumptions were simulated (following sections a, b and c):
i. Full soil resistance during installation (SRD\textsubscript{max} following Figure 3), max. hammer energy E\textsubscript{i}, transferred efficiency $\eta_t$ of 95%

ii. Fully reduced soil resistance during installation (SRD\textsubscript{min} following Figure 3) max. hammer energy E\textsubscript{i}, transferred efficiency $\eta_t$ of 95%

iii. Soil resistance including friction fatigue approach (SRD\textsubscript{max} towards SRD\textsubscript{min} following Figure 3) max. hammer energy E\textsubscript{i}, transferred efficiency $\eta_t$ of 95%

iv. Soil resistance including friction fatigue approach (SRD\textsubscript{max} towards SRD\textsubscript{min} following Figure 3) with a transferred efficiency $\eta_t$ of 95% and a specially adapted energy concept.

Table 1 shows the results from the pile driving simulations for the selected models. The total number of blows and the total pile resistance at the end of driving (EOD) are shown.

Table 1: Example of the construction of one table

<table>
<thead>
<tr>
<th>Simulation model</th>
<th>Total number of blows</th>
<th>Total pile resistance at EOD [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>i.</td>
<td>1539</td>
<td>62009</td>
</tr>
<tr>
<td>ii.</td>
<td>709</td>
<td>22428</td>
</tr>
<tr>
<td>iii.</td>
<td>1128</td>
<td>39367</td>
</tr>
<tr>
<td>iv.</td>
<td>3089</td>
<td>39367</td>
</tr>
</tbody>
</table>

Obviously, using the same hammer energy and efficiency (i, ii and iii), the highest soil resistance and accordingly the highest number of blows will result from simulation i (SRD\textsubscript{max} during the whole driving process). When only applying SRD\textsubscript{min} the lowest blow count and soil resistance will be calculated. Using the friction fatigue approach as described in section c, the soil changes in accordance with the distance between the soil element and the pile toe. Therefore, intermediate values for blow count and soil resistance are analyzed. For the simulation model iv the total pile resistance will be the same as the soil resistance changes the same way.

Figure 4: Reduction in unit shaft resistance at element under consideration during pile driving
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As the energy will be reduced especially at the beginning of driving, the blow counts are significantly higher. The pile driving fatigue calculation results are shown in Figure 4. The pile driving fatigue values from simulation iii (assumed as the 'correct' result) were defined as 100% (here: 100% means 'true value' - even though pile driving fatigue is far away from material failure).

Using 'false' soil models (soil resistance does not change during impact pile driving) will lead to a significant change in pile driving fatigue. Resistances that are too high will lead to an over prediction and resistances that are too low will lead to an under prediction of pile driving fatigue. Additionally, when adjusting the hammer energy $E_i$ during impact pile driving and in accordance with the subsoil conditions on site (under the use of the same friction fatigue model as in iii) a significant pile driving fatigue reduction of approx. 50% can be achieved.

5 CONCLUSIONS
For a proper and most accurate pile driving simulation large experience with pile driving simulation is strictly necessary. Detailed knowledge has to be available e.g. for the used numerical model, the stress wave propagation through a 'one-dimensional' system (pile) or the change of the soil parameters under a dynamic influence (impact pile driving). As shown in the presented example (section 4) small simplifications or misunderstandings in the selected model will already lead to a significant change in the pile driving prediction results. Consequently, the pile driving fatigue values will differ significantly as well.

During any pile driving fatigue calculation, particular attention has to be paid to topics such as for example the transferred energy (see section b), the friction fatigue during impact pile driving (see section c) or the general structure of the subsoil (see section a).

With many years of experience in pile driving analysis, dynamic load tests, signal matching, pile driving prediction etc. the Fichtner Water & Wind staff has great knowledge in terms of the most sensitive parameters required for pile driving predictions and pile driving fatigue calculations.

Therefore, besides most accurate pile driving prediction and pile driving fatigue calculations, an energy concept for each offshore wind farm location can be designed by us, to reduce the pile driving fatigue as significantly as possible.

6 REFERENCES
Fritsch, M., Fischer, J., Tworuschka, H.


Robust Installation Planning of Offshore Wind Farms

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Keywords: Installation, Offshore wind farm, Planning, ECN Install

Summary: The main risk in the installation of offshore wind farms is caused by unfavorable weather conditions, which can delay the entire installation plan up to several months and hence, increase the installation costs significantly. Currently, there is no study available to model the entire installation procedure of offshore wind farms, considering all the associated delays, costs and risks. In this paper, a generic model for the complete supply chain of offshore wind farms installation is discussed, which can be used for robust installation planning of support structures, turbines and cabling of offshore wind farms, considering delays caused by adverse weather conditions and weather limits of vessels and equipment in far offshore or at the harbour. With this model, it’s possible to calculate the costs and delays associated with offshore wind farm installation and thereby, arrive with solutions to mitigate the project risks and reduce the installation costs. This model is in the developing phase by the name of “ECN Install” as part of the Dutch FLOW program [1].

1 Introduction

Offshore wind energy is a key pillar of the European energy transition [2]. The European countries are aiming to cover 35% of their energy consumption from renewables by 2020, which sets a target of 40 GW installed offshore capacity by 2020 [3]. By the end of 2013, more than 2000 offshore wind turbines were installed in Europe, making a cumulative total of 6.5 GW in 69 offshore wind farms [4]. Therefore, to reach the ambitious European climate and energy targets, more than 30 GW offshore capacity should be installed by 2020. This means that the required effort for installation of offshore wind farms will be enormous in the coming years. Furthermore, because of environmental laws and limited space near shore, future wind farms will be located in further offshore and consequently in deeper water and harsher weather conditions. Hence, in order to reduce the installation time and costs it is essential to develop robust long term installation planning concepts that can be used by a variety of users such as installation contractors, wind farm developers, harbour authorities and OEMs.

There are several existing Installation models such as the Jobs and Economic Development Impact (JEDI) Model, DOWEC Cost Model, OWECOP II Model, Opti-OWES Model and the FLOW cost model [8,9,10,11]. However, the tools developed are
only cost models and don’t include the aspects of supply and logistics. Besides that, some contractors have their own logistic software in-house. But then, there is a lack of one software package which can be used by all the parties.

With the model proposed in this paper, the concerned parties involved in the installation of offshore wind farms can share project input and develop one software environment. Thereby giving an opportunity to model the entire wind farm installation with an integrated approach with inputs from all parties.

The planning and installation model being developed can be used for accurate prediction of associated costs, delays, effort incurred and risks while installing an offshore wind farm. In the consecutive sections, the general advancements and challenges of installation are discussed. The model methodology will be explained and sample results with tables and figures are demonstrated for a better understanding. The paper is concluded with a summary of capabilities of the tool and topics of further research for improvement of the working model.

2 INSTALLATION OF OFFSHORE WIND FARMS

The installation of offshore wind farms is influenced significantly by weather conditions. The initial planned timeline might not be valid while undertaking the real implementation of the installation steps. In general, it is observed that support structures can be installed in more adverse weather conditions compared to the blades and nacelle. The installation of the latter requires more reliability of the weather, logistics and working conditions. This implies that planning installation is quite critical for the overall success of the process. Moreover, an actual indication of the date of commissioning and associated installation cost is necessary. Another aspect critical for the successful realisation of the project is the spare part and resource management. For instance, if the capability of the vessel is to transport six piles, it is necessary that the same number are available at the harbour for loading. These small optimisation of resources can highly influence the installation costs and hence mitigate project risk. With larger and far-offshore wind farms planned in future, the efficiency and the reliability of the entire installation process is quite relevant.

2.1 Installation advancements

Installation costs are dominated by vessel and equipment hiring costs. To date, with existing installation of offshore farms, new installation practices are being explored which can impact these costs [8,9,10,11]. Some of these trends are:

1. Use of larger and faster vessels for installation, allowing more turbines to be transported and installed in one trip, thus reducing time spent transporting hardware from manufacturing facilities. An exciting method is the offshore transfer of components from support vessels to jack-up installation vessels, avoiding the
need for these high-cost vessels to return to port to collect another batch of turbines.
2. Pre-assembly of turbines completely on land and thereafter install turbines and foundations in one operation, potentially reducing dependence on expensive vessels. [6]
3. Use of multiple harbours to reduce the transport costs of arranging components and moreover also reduce the travel time from the nearest harbour to the wind farm offshore. [7]
4. To date, installation of subsea cables has been characterised by cost overruns and in a number of cases, cable damage. Subsea cables have been the largest source of insurance claims in offshore wind to date. There is certainly room for savings due to a more holistic consideration of cable design and installation requirements and methods.

2.2 Challenges in planning and Installation of offshore farms

A key concern for many offshore developers is the availability and suitability of installation vessels and the corresponding equipment. Choosing the right vessel is more important having a large vessel [5].

The other potential bottleneck are skills, project management and offshore technicians. It is recognized that different planning strategies can influence installation times and costs.

With the challenges above, an installation model is discussed in the next section to provide a solution for long term planning of installation of offshore wind farms in a thorough way applicable for different installation strategies.

3 GENERIC MODEL DESCRIPTION

3.1 Concept

The objective of a planning system for the installation of offshore wind farms is to reduce the construction time and the project risk, which is proportional to the installation costs. With each additional day, the leasing of vessels, equipment and crew will exceed the initial planned project time and estimated costs. Therefore, to support the process, ECN by using its in-depth knowledge in modeling operation and maintenance of offshore wind farms [12] started the development of the ‘ECN Install’, a software tool applicable for robust installation planning of offshore wind farms. As indicated above, the model can be applied differently by separate users.

The Installation model is a time-domain simulation program being developed in MATLAB and Visual Studio and is designed to assist OEMs and developers of wind farms to determine the optimal installation strategy during the planning phase of installation.
3.2 Properties of Installation Model

The Installation model of the ECN Install provides the user to simulate the proposed plan of installation in the form of steps. In order to have a generic model only three types of steps are considered:

1. Loading step, which describes the set of activities to load the components from the harbours to the vessel;
2. Traveling step, which describes the traveling of the vessels between the harbours and the farm;
3. Installation step, which is used to describe all installation activities being performed with vessels and equipment; e.g. piling or vessel positioning.

Besides the above set of steps defined as an initial plan by the user, general inputs of the wind farm installation like working patterns of the crew, climate information, vessels and equipment used, harbours and turbine data, etc. are also obtained. A screenshot of the interface with vessel inputs is shown in Figure 1.

![Figure 1: Graphical Interface sample for ECN Install](image)

While defining each of the steps, the user can select the corresponding vessel, equipment, weather thresholds and the components to be installed. The steps are initially processed to compile without any delays. Additionally, the weather information provided by the user is used to create accessibility vectors for performing each particular step. By using a Monte Carlo simulation in the time domain, costs and delays caused by working hours and vessel weather limits are calculated. A working example of the implementation of such steps is described in Figure 2.
In the following figure, an activity termed as pre-piling is performed. In this example, three lines of parallel activities are executed, with each box representing one step. The step could either be loading, traveling or installation.

<table>
<thead>
<tr>
<th>1.1</th>
<th>1.2</th>
<th>1.3</th>
<th>1.4</th>
<th>1.5</th>
<th>1.6</th>
<th>1.7</th>
<th>1.8</th>
<th>…</th>
<th>1.n</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>2.1</td>
<td>2.2</td>
<td>2.3</td>
<td>2.4</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>3.1</td>
<td>3.2</td>
<td>3.3</td>
<td>3.4</td>
<td>3.5</td>
<td>3.6</td>
<td></td>
<td>…</td>
<td>3.n</td>
</tr>
</tbody>
</table>

Figure 2: Sample Installation of Piles

For instance, step 1.1 could be loading a component to the jack-up barge at the harbour, step 1.2 could be defined as traveling to the farm, and steps 1.3 to the end of the sequence could be vessel positioning, vessel jacking up, placing the template, placing the pile and then hammering. The second line could be a parallel activity like feeding new piles to the jack-up barge with a support vessel and the third line could be an independent parallel activity like installation of balance of plant.

Additionally, it’s possible to cluster a few steps together to use the same weather window. For instance, steps for placing the template and pile could be clustered together to ensure that, during installation the necessary weather window to perform both activities exists. In order to simplify the definition of installation activities it is also possible to repeat several activities, like repeating all piling steps for the number of wind turbines in the farm.

4 RESULTS

The installation model provides an overview of the estimated delays, the time for the completion of each step and the associated costs to complete the installation. The following results and outputs are obtained:

1. Simulation of the installation logistics of an entire offshore wind power plant
2. An overview of:
   i. Breakdown of installation costs
   ii. Breakdown of installation times (planning) and expected date of commissioning
   iii. Breakdown of installation delays
   iv. Breakdown of installation hours for which an equipment is used
   v. Levelized Cost of Energy (LCOE) of installation
   vi. Lost revenues of energy production due to delays
3. Find the best installation solution by optimizing outputs with input parameters
4. Spare part and resource management during the installation process.

A sample output with the breakdown of delays for five steps is given in Table 1. These steps are a combination of loading, travelling and installation steps.
Table 1: Sample Output Table with Weather and Shift Delay

<table>
<thead>
<tr>
<th>Step No.</th>
<th>Step Code</th>
<th>Step Weather Duration</th>
<th>Step Duration</th>
<th>Step Starting Time</th>
<th>Shift Delay</th>
<th>Weather Delay</th>
<th>Harbour Delay</th>
<th>Step End Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.1</td>
<td>4</td>
<td>2</td>
<td>50</td>
<td>0</td>
<td>65</td>
<td>0</td>
<td>117</td>
</tr>
<tr>
<td>2</td>
<td>1.2</td>
<td>2</td>
<td>2</td>
<td>117</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>119</td>
</tr>
<tr>
<td>3</td>
<td>1.3</td>
<td>12</td>
<td>12</td>
<td>119</td>
<td>7</td>
<td>336</td>
<td>0</td>
<td>474</td>
</tr>
<tr>
<td>4</td>
<td>1.4</td>
<td>6</td>
<td>6</td>
<td>474</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>480</td>
</tr>
<tr>
<td>5</td>
<td>1.5</td>
<td>10</td>
<td>6</td>
<td>480</td>
<td>0</td>
<td>18</td>
<td>0</td>
<td>504</td>
</tr>
</tbody>
</table>

It can be easily identified that for each step there is a specific code. For each step with its particular working pattern and weather limits a number of simulations is performed to calculate the delays and associated costs. The corresponding delays, viz. shift delay (due to non-working hours of the crew), weather delay (due to non-accessibility of the wind farm) and harbour delay (due to harbour locks) are obtained. Corresponding to these values, the cost calculations and the effort by the crew are computed and finalized in the form of tables and graphs.

5 CONCLUSION

The model discussed above gives a clear understanding of the installation delays and the relevant costs. Currently, there is no tool addressing the complete long term installation planning. With this model, different users can use this application for various purposes. Some of them are discussed below:

- Developers, investors and financial institutions need to know the costs of installation, date of commissioning and involved risks. Therefore, they are mainly interested in the total installation CAPEX, LCOE and expected date of commissioning. Their risk analysis will be based on the probabilities given to outputs (e.g. 90% chance of value x between x1 and x2). The analysis will take place during the planning phase of the farm.
- Vessel designers analyse the business case of a potential design or make choices during the design process. Therefore, they are interested in the breakdowns of costs, hours, planning and delays. The focus will be on logistics and specifications of equipment. The installation procedures will be of great importance. The analysis can take place during a market study or design process.
- Installation contractors execute the installations and make a plan to estimate the expected costs. Therefore, they are interested in the breakdown of costs, hours, planning and delay. The focus will be on logistics, specifications of equipment and planning. The installation procedures will be of great importance. The analysis can take place during a request of quotation and the planning phase.
Suppliers deliver the components to the assembly sites or ports. Therefore, they are mainly interested in the required stock over time, besides when and where the component should be made available.

- Harbour authorities want to organise the site to meet the requirements for installations. They are interested in the required stock and cranes over time. The analysis can be done to investigate large investments to further develop the infrastructure of the harbour or to determine the logistics and requirements of the site for a specific project.

With the above list of potential users, the model has great capabilities in bringing the entire wind industry together following an integrated approach. Overall, there are definitely prospective areas which can be combined with this tool to make it really useful for the industry.

6 Future Works

In the above sections, the advantage of having a generic long term planning installation model and tool for the growing offshore wind farm industry are illustrated. However, installation is only one component of the entire life cycle cost estimation of the offshore wind farm. Hence, it will be interesting to assess the possibility of integrating the model with wind farm planning and life cycle cost assessment tools. Furthermore, short term decision support models for the daily installation process of offshore wind farms can also be explored.

7 Acknowledgements

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8 References


Installation Planning of Offshore Wind Farms

Masoud Asgarpour
IWEC Conference
03-09-2014

Introduction

- Why model the installation?
- The model
- The tool “ECN Install”
- Case study
- Conclusion
- Future work
- Acknowledgment
Why Model the Installation?

• Reduce cost
  – Total installation costs
  – Cost of technicians
  – Cost of equipment
  – Cost of vessels
  – Cost of harbour(s)

Why Model the Installation?

• Reduce delays and risk → again cost
  – Starting date
  – Operation time
  – Weather delays
  – Shift delays
  – Harbour lock
  – Commissioning date
Why Model the Installation?

- Optimized resource management \(\rightarrow\) and again cost
  - Number of technicians
  - Working shifts
  - Harbour(s)
  - Equipment
  - Vessels

Why Model the Installation?

- Commercial proof of new concepts
  - Innovative installation methods
  - Innovative foundations
  - Innovative wind turbines
  - New equipment
  - New offshore vessels
The Model

• **Inputs**
  - Wind turbine(s)
  - Components
  - Climate data
  - Operation bases
  - Equipment
  - Vessel
  - Working shifts
  - Fixed costs

• **Planning**
  - Operation lines
  - Operation steps
    - Loading
    - Sailing
    - Installation

• **Output**
  - Time
  - Cost
  - Resource

---

**Step Duration**: 6 hours (Including Harbour Duration)
**Step Weather Duration**: 10 hours (Including Harbour Duration)
**Working shift**: Single working shift (Regular)
**Step Type**: Traveling (ONE-Go step)

**Shift Delay**: 20.5 hours
**Weather Delay**: 3.5 hours
**Step Duration**: 5 hours
**Harbour Delay**: 1 hour
**Completion Time**: 30 hours
The Model

Installation Tool “ECN Install”
Case Study

- ENECO 210 MW ‘Q4 West’ offshore wind farm [V112 3.0 MW]
- Part I: installation of 5 wind turbines with monopiles
- Selected harbour: Ijmuiden [26km to the site]
- Water depth: on average 22m

Case Study

- Installation steps:
  - 98 steps, ranging from 1h to 20h, in total 468h or 20 days
  - Inspection, piling, TP, grouting, tower, nacelle, blades, substation & inter-array cables

- Possible delays:
  - Weather delay
  - Working shifts delay
  - Harbour delay

- Weather data:
  - 10 years [1995-2004]
  - Average WH: 1.44m
  - Average WS: 7.47m/s
Case Study

- Different starting time, working shift and operation type
  - Scenario I: single working shift, unsplittable operations
  - Scenario II: single working shift, splittable operations
  - Scenario III: 24/7 working shift, unsplittable operations
  - Scenario IV: 24/7 working shift, splittable operations

- Different vessels → different weather restrictions (WS-WH)
Case Study

(I) single shift, unsplittable

(IV) 24/7 shift, splittable
Case Study

- Different starting time, working shift and operation type
  - Scenario I: single working shift, unsplittable operations
  - Scenario II: single working shift, splittable operations
  - Scenario III: 24/7 working shift, unsplittable operations
  - Scenario IV: 24/7 working shift, splittable operations
- Different vessels → different weather restrictions (WS-WH)
Case Study

Different weather restrictions
(IV) 24/7 shift, splittable (Aug 2015)

Conclusion

- Better estimation (and optimization) of installation costs, time and resources for
  - WF developers & contractors
  - Investors & financial institutions
  - OEMs, suppliers and port authorities
- Commercial proof of new and innovative concepts
  - Installation methods
  - Support structures & wind turbines
  - Equipment and vessels
Future Work

- Uncertainty analysis
- More outputs: costs and resources
- More user friendly: repeat steps, cluster steps and parallel lines
- More restriction: wave direction, wave length and current
- Integration into LCCA models
- Short term decision support

Acknowledgment

- Part of Dutch FLOW “Industrialize Installation” project
- Project partners:
  - IHC Offshore & Marine B.V.
  - Van Oord
  - Ballast Nedam
  - Marine
  - TNO
Thanks for your attention

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Innovative solutions
to lower the cost of energy
DECENTRALIZED CONTROL DESIGN OF A NINE PHASE PERMANENT MAGNET GENERATOR

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Keywords: Permanent magnet generator, distributed control, wind turbine, wind emulator.

Summary: This work describes the decentralized control design of a direct driven triple three-phase Permanent Magnet Synchronous Machine (PMSM) for offshore wind turbine generation. The machine has three different stators embedded on the same yoke (nine-phase system), connected to the grid through three independent back-to-back converters. Normally, multiphase machines require centralized controllers in order to achieve a good control performance. However, employing the proposed control, the triple three-phase machine can be operated without communications between the different stator controls.

1 INTRODUCTION

In recent years, a number of offshore wind farms have been commissioned. Offshore locations show several advantages compared to onshore, namely: better wind resources, better availability of space, ease of transport of large turbine structures and reduced visual impact. However, offshore generation poses several technical challenges such as the complexity of transferring the produced energy to the main grid and installation and maintenance logistics and costs. Therefore, there is a need for highly reliable wind turbines with low maintenance requirements in order to minimize the operation and maintenance expenses. Specifically, the direct driven multipolar Permanent Magnet Synchronous Machine (PMSM) with multiple three-phase stators is one of the most interesting options due to the redundant structure which increase the operation possibilities [1, 2].

The aforementioned PMSM topology allows to operate each three-phase stator independently, setting different setpoints for each of the machine stators. Regarding the machine current regulators, its design is not straightforward and usually a centralized controller is required to ensure a proper operation of the machine without interferences caused by the cross-coupling between stators [3, 4]. In some implementations, this controller would not be convenient, as the
loss of communication between the controller boards of each individual converter would make the system unable to operate. Therefore, it is interesting to consider the machine current control from a decentralized point of view, designing individual current regulators for each of the machine stators, to increase the control reliability.

This work proposes a distributed control of a direct driven triple three-phase PMSG. The torque control is divided in three different controllers, each implemented in the converter connected to each stator. Each converter carries out its own control allowing the system to work with three different modes of operation, considering different setpoints for each of the controlled stators.

2 MACHINE MODELLING

This section shows the model derivation of a generic surface mounted permanent magnet triple three-phase machine in the rotating reference frame. The winding topology is not specified because the model is algebraically equivalent for the different possible types. It is assumed that the rotor permanent magnet flux is sinusoidally distributed along the air gap and linear reluctance behavior is considered for simplicity. Also, each stator has its own isolated neutral point.

Then, the machine voltage equations can be defined as [5]:

\[ v_s^{abc} = r_s^{abc} + \frac{d}{dt} i_s^{abc} \]

where \( v_s^{abc} \) is the vector of voltages applied to the machine, \( i_s^{abc} \) is the vector of currents flowing through the machine, \( \lambda_s^{abc} \) is the flux linkage vector of the machine and \( r_s \) is the diagonal matrix that represents the total resistance of the machine windings. Then, a triple park transformation oriented with the rotor angle (considering the shifting between stators), is applied to analyze the machine equations in the rotor reference frame [6]. Once the transformation is applied, the nine-phase machine model can be expressed as:

\[ v_s^{dq0} = \left( r_s + A_1 \right) i_s^{dq0} + A_2 \frac{d}{dt} i_s^{dq0} + v_m^{dq0}, \quad v_m^{dq0} = \lambda_m \omega (1 1 1 0 0 0)^T \]

where, \( v_s^{dq0}, v_m^{dq0} \) and \( i_s^{dq0} \) are the converter voltages, the machine voltages due to the magnets flux linkages and the current flowing through the machine in the new reference frame, respectively. The system matrices \( A_1 \) and \( A_2 \) show the relations between the machine variables in the rotating reference frame.

\[
A_1 = \omega \begin{pmatrix}
q_{s1} & q_{s2} & q_{s3} & d_{s1} & d_{s2} & d_{s3} \\
0 & -C & C & A & B & B \\
-C & C & 0 & B & A & A \\
-A & -B & -B & 0 & -C & C \\
-B & -A & -B & -A & -C & C \\
-B & -A & -B & C & 0 & -C \\
\end{pmatrix}
\]

\[
A_2 = \begin{pmatrix}
q_{s1} & q_{s2} & q_{s3} & d_{s1} & d_{s2} & d_{s3} \\
A & B & B & 0 & C & -C \\
B & A & B & -C & 0 & C \\
B & A & C & -C & 0 & C \\
0 & -C & C & A & B & d_{s1} \\
C & 0 & -C & B & A & d_{s2} \\
-C & 0 & B & A & A & d_{s3} \\
\end{pmatrix}
\]

where,

\[
A = L - M3, B = -\cos \frac{\pi}{9} M_4 - \cos \frac{5\pi}{9} M_2 + \cos \frac{2\pi}{9} M_1, C = -\cos \frac{7\pi}{18} M_4 + \cos \frac{\pi}{18} M_2 - \sin \frac{2\pi}{9} M_1
\]

Note that the number of equations has been reduced from nine to six because the machine windings have an isolated neutral point for each stator, thus the homopolar currents are zero. Regarding the machine decentralized control, the elements highlighted in red of the matrices
$A_1$ and $A_2$ show the couplings between the variables within the same stator. These elements are used to design the individual stator controllers neglecting other coupling components. Assuming this, the simplified equations for one stator are:

$$
\begin{bmatrix}
    v_{st}^q \\
    v_{st}^d
\end{bmatrix} = \begin{bmatrix}
    r & \omega(L - M_3) \\
    -\omega(L - M_3) & r
\end{bmatrix} \begin{bmatrix}
    i_{st}^q \\
    i_{st}^d
\end{bmatrix} + \begin{bmatrix}
    L - M_3 & 0 \\
    0 & L - M_3
\end{bmatrix} \frac{d}{dt} \begin{bmatrix}
    i_{st}^q \\
    i_{st}^d
\end{bmatrix} + \lambda_m \omega \begin{bmatrix}
    1 \\
    0
\end{bmatrix}
$$

These equations (4) are equivalent to a conventional three-phase machine. Therefore, it seems convenient to apply a classical vector control structure [7, 8] to regulate the current flowing through each stator. Based on this strategy, two conventional Proportional-Integral (PI) controllers are employed to regulate the current ($q$ and $d$ components), considering the closed loop time response $\tau$ as the control basic parameter design:

$$
K_c = \frac{K_p + K_i}{s}, K_p = \frac{1}{\tau} (L - M_3) \quad \text{and} \quad K_i = \frac{1}{\tau} (r)
$$

The selection of the time response $\tau$ will have an impact on the transient couplings between stators, because it is affecting the dynamics of the machine. Specifically, it should be selected to accomplish the control time response requirements while reducing the cross-coupling effects between the stators. Figure 1 shows the global machine current controller, based on the decentralized control proposal.

Figure 1 - Generator decentralized control scheme

3 Simulations

In this section, an example of decentralized control of a wind turbine including a triple three-phase generator is simulated using Matlab Simulink®. In order to evaluate the current controller performance, two different case studies are analyzed. The scenario, in both cases, consists on introducing an active current step change of the stator 1 ($i_{q1}$ from 0 to 500 A) while maintaining the other current references set to zero. The difference between both tests is the machine speed conditions defined for the simulation (rated speed and 20% of the rated speed), which allows to test the performance of the decentralized controller in a wide operating range. Besides, these simulations have been carried out considering four different time constants $\tau$ for the current control, specifically 10, 20, 50 and 100 ms, in order to show the effect of this parameter on the control performance.
Simulations in Figures 2 and 3 show the effects between the \( qd \) variables of the different stators for a current \( q \) reference step change of the stator 1. As the controllers have been designed based on the equations of the same stator, the current \( d \) of the stator 1 is not importantly affected during the current \( q \) transient due to the local decoupling included in the controller. Otherwise, the \( qd \) current components of the other stators are more affected in comparison with local \( d \) current, because the other controllers do not include any decoupling loop in this decentralized approach. Also, it can be seen that the effect of the couplings is highly dependent on the machine speed. Specifically, the cross-coupling between the variables is more important at high speeds.

Regarding the different time constants analyzed, it should be stated that the controller designed with a faster response \( r \), show smaller interaction effects between the \( qd \) stator components, which is desirable for the machine operation. Therefore, as faster local current regulators show better performance in terms of variables cross-coupling, a maximum time response for the local current loops can be defined, according to the maximum allowable error in the current regulators.
Once the methodology for designing the decentralized control has been tested in simulation, the control algorithms are validated in a scaled wind turbine test rig. This rig is composed by two 30 kW permanent magnet machines, one acting as a motor and the other as a generator, mechanically connected through their shafts. The motor is a conventional three-phase machine connected to a standard frequency converter. The generator is a nine-phase machine with three independent star-connected stators, connected to the grid by means of three independent back-to-back converters. Figure 4 shows the aforementioned machines, the cabinet where the frequency converter and the back-to-back converter for the multi-phase machine are enclosed and a complete conceptual diagram of the setup.

The test rig motor acts as a wind emulator, using the frequency converter to regulate the speed of its shaft. The generator regulates the machine torque by means of the specific designed decentralized controller. The grid side converters carry out a voltage control of the DC bus to inject the active power coming from the generator to the grid.

Regarding the generator control, the decentralized strategy implemented is able to regulate each stator active and reactive current. The controller design has been developed based on the parameters obtained from FEM simulation, Simulink simulations, together with experimental tests developed with the machine. Then, applying the methodology presented above, six PI current controllers (two for each stator) have been calculated and programmed to the DSP control board of the individual converters.
5 EXPERIMENTAL RESULTS

Two different experiments are performed to validate the behavior of the designed controller. The first test is performed to test the operation of the machine decentralized controller, setting the active current reference of the three stators to the same value. Then, a second experiment is developed to validate that the current regulator is able to operate the machine with different current references for each stator.

a. Test 1: Three stators regulating equal active current

Figure 5 shows the currents flowing through the phase ‘a’ of each stator while each converter is regulating the same amount of machine active current. As it is expected, the current values are equal. Also, it can be seen that the currents flowing through stators two and three are shifted 40º and 80º, from the current flowing through stator one, respectively (the stators are electrically shifted 40º).

Regarding the grid power integration, the grid side converters perform the DC bus voltage control of the three different back-to-back converters, injecting all the power that is being generated by the PMSG. Figure 6 shows that the DC bus voltage is maintained at a constant value, while the system is injecting power to the grid. This picture also shows that the three grid side converter phase ‘a’ currents are in phase, because these converters are connected to the same AC grid. These results validate the performance of the controller for a particular case in which it is regulating the same amount of active current flowing through the three stators.

b. Test 2: Three stators regulating different active current

Figure 7 shows the currents flowing through the phase ‘a’ of the machine. It can be observed that the controller is able to regulate different active current setpoints for each stator. Again, the 40º shift between the currents is present. This second test validates the designed controllers while regulating different levels of active current flowing through each stator.

Figure 51 - Currents flowing through the machine side converters
6 CONCLUSION

The decentralized control design of a nine phase wind turbine generator is presented. The control is performed by three back-to-back converters connected to the generator. Each converter carries out its own torque control without sharing information with the rest of the converters. The control design is based on the system equations obtained through theoretical developments and finite element models of the generator. The controller design methodology is tested in simulation and validated in a scaled wind turbine generator test rig. Different experiments are performed to test the behavior of the machine torque control, considering the operation with one, two or three active stators at the same time, also generating the same or a different amount of power. The performed tests have shown satisfactory results proving the concept viability. These results have also shown that this generator configuration along with the designed controllers could be an interesting alternative for offshore applications, showing additional capabilities in terms of redundancy and control possibilities.
7 ACKNOWLEDGEMENTS
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8 REFERENCES


GENERATOR EMULATION CONTROL FOR PROVIDING SYNTHETIC INERTIA IN VSC-HVDC SYSTEM

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Keywords: VSC-HVDC, Inertia emulation, Synchronous machine, Control strategy, Generator emulation control

Summary: This paper presents a new control method for providing synthetic inertia in VSC-HVDC system on the basis of the generator emulation concept. An extended control loop is added to emulate both the inertia and the droop characteristics of synchronous machine. By using the proposed method, the VSC-HVDC converter station can provide active power support to increase the inertia of power system and help the power system to restore from frequency deviations. The simulation results are provided to verify the proposed control method.

1 INTRODUCTION

The large inertia of traditional synchronous-machine (SM) based power system can be helpful to stabilize the frequency and increase the opportunity for control systems to act when a large power plant trips [1]. In contrast, renewable energy sources are typically based on power electronic converters and have very little or no inertia, thus they can cause a significant reduction of system inertia, which presents a greater risk of instability. Meanwhile, the growing number of HVDC installations decouples the kinetic inertia of different AC systems as well [2]. For these reasons, the level of inertia in future power system may decrease. To respond to this possible situation, new grid codes suggest that the power converters connected to the grid shall be capable of providing synthetic inertia [3-6].

Various strategies have been proposed to emulate the behavior of a synchronous machine to provide additional inertia by extended control of power electronic converters in wind farm or PV system, such as virtual synchronous machine [7, 8], generator emulation control [9] and frequency derivative method [10, 11]. An inertia emulation control has been proposed to use the energy of DC link capacitors in VSC-HVDC system to provide additional inertia [2].

This paper extends the concept of generator emulation control (GEC) to the VSC-HVDC system. Due to the large capacity and fast control ability of VSC-HVDC system, the stability of power system can be increased after incorporating the inertia and frequency support provided by GEC. A two terminal VSC-HVDC system with GEC has been simulated to verify the proposed control strategy.

This paper is organized as follows: Section II introduces the conventional control strategy of the VSC-HVDC system. The concept of generator emulation control is described briefly in
Section III, and the extended DC link controller is also introduced in this section. Section IV provides the simulation results for verification.

2 CONTROL OF VSC-HVDC SYSTEM

The VSC-HVDC system is widely considered as a promising technology for optimized integration of offshore windfarms, owing to its fast and flexible power control ability. Figure 1 shows the typical structure and control strategy of a VSC-HVDC connected offshore system. In this two-terminal system, the left converter station is connected to the onshore SM-based network and controls the DC link voltage, while the right side is the offshore windfarms. Thus, the right converter station will control the active power and reactive power for optimized utilization of wind energy.

To reach independent regulation of reactive power and active power, vector current control is usually used in the VSC-HVDC system [12]. By introducing a synchronous rotating d-q reference frame, the d-axis current and q-axis current can be controlled independently with a decoupled controller. Moreover, when the d-axis is aligned with the voltage space vector, the active power or the DC link voltage can be controlled by the d-axis current, while the q-axis current can be used to regulate the reactive power or the grid voltage, as shown in Figure 1.

However, this conventional control method has no contribution to the system inertia and frequency support. Take the system shown in Figure 1 as an example, when the switchable load \( S_2 \) is suddenly tripped, the system frequency will be change and might lead to instability of the power system. The change rate of frequency at the PCC is dominated by the swing equation (per-unit) as [13]

\[
\frac{2H}{\omega_{grid}} \frac{d\omega_{gen}}{dt} = P_S - P_L + P_C
\]

where \( H \) is the normalized inertia constant, \( \omega_{grid} \) is the synchronous electrical radian frequency. \( P_S, P_C \) denote the power from the power system and the converter station, respectively. \( P_L \) is
the load power which is composed of fixed load $S_1$ and switchable load $S_2$ in the case of Figure 1.

Consequently, the frequency deviation and the rate of frequency change can be limited to a safety range if the converter station can provide a fast change of active power to compensate the change of load power. This can be done by proper control of the active power or the DC link voltage, which is equivalent to increasing the inertia of the power system.

3 EXTENDED GEC CONTROL FOR VSC-HVDC SYSTEM

The proposed extended GEC controller is based on the concept of generator emulation control. To have a better understanding, the concept of GEC is first explained briefly in this section [14].

![Figure 2: (a) Simplified model of SM; (b) Dynamic model of SM with droop control [14]](image)

Given three phase balanced conditions and ignoring the machine losses, the simplified model of a SM shown in Figure 2(a) can be used for transient stability studies. It is composed of a constant internal voltage and its direct axis transient reactance [13]. As a result, the output power of the SM can be represented as

$$P_O = \frac{V_O V_S \delta}{X}$$  \hspace{1cm} (2)

where angle $\delta$ is the phase difference between the induced internal voltage and the output voltage. By considering possible frequency fluctuation when the SM is connected to a power system, the output power can be further expressed as

$$P_O = \frac{V_O V_S}{X} \int (\omega_{gen} - \omega_{grid}) dt = k_s \int (\omega_{gen} - \omega_{grid}) dt$$  \hspace{1cm} (3)

Considering a generating unit consisting of a SM and its prime mover, the inertia behavior of the SM can be represented by the famous swing equation [13]

$$\frac{J \omega_{gen}}{p^2} \frac{d\omega_{gen}}{dt} = P_{in} - P_O$$  \hspace{1cm} (4)

where $J$ is the total moment of inertia of the rotating masses, and $p$ is the number of poles of the SM. $P_{in}$ is the mechanical power supplied by the prime mover, and $P_O$ is the electrical power output of the generator. Consequently, the electrical frequency $\omega_{gen}$ can be expressed as
Xudan Liu, Andreas Lindemann and Nikolaus Wirtz

\[ \omega_{\text{gen}} = \frac{P^2}{J\omega_{\text{gen}}} \int (P_{\text{in}} - P_o) dt = k_g \int (P_{\text{in}} - P_o) dt \]

(5)

In the power system, a power-frequency droop is usually added to provide prime frequency support, which is very helpful for keeping stability of the power system during large power unbalance. Combing (3) and (5), a dynamic model of the SM with power-frequency droop are build and shown in Figure 2(b). According to this model, the transfer function between the output active power and the grid frequency in the high frequency range can be represented as

\[ \frac{P_o(s)}{\omega_{\text{grid}}(s)} \approx \frac{-P_o(s)}{\Delta \omega(s)} = \frac{k_s}{s} \]

(6)

And in the low frequency range, the SM is mainly dominated by the droop characteristic, can be expressed by

\[ \frac{P_o(s)}{\omega_{\text{grid}}(s)} \approx -k_{\text{droop}} \]

(7)

Thus, these two equations show the inertia and frequency support characteristics of the SM, which are also the targets we want to emulate in the VSC-HVDC system.

As reported in [14], the inertia and frequency support characteristics of the SM can be emulated by a simple PLL as shown in Figure 3(a), since the transfer function of this PLL can be written as

\[ \frac{P_o(s)}{\omega_{\text{grid}}(s)} = \frac{k_s}{s + k_p} \]

(8)

which shows very close high frequency and low frequency performance as the SM shown in Figure 2(a).

As a consequence, the needed compensating power can be obtained if this PLL is used to provide reference signal for the active power loop or the DC link voltage controller. Therefore, a new extended DC link voltage controller is proposed by introducing this PLL into the DC regulated loop, as shown in Figure 3(b). Since this PLL can represent the inertia and frequency support characteristics of the SM, it is used to provide reference signal for the DC link controller. Therefore, the stored energy in the DC link capacitors can be used to compensate the sudden change of load power.
Given the strategy is used under per-unit system, the output of GEC loop will be 1 under normal condition and will change according to the characteristics of SM during frequency fluctuations. To amplify this change, $V_{DC}^*$ is subtracted from the output of GEC loop to get only the needed reference change. To amplify the effect of the frequency deviation, the factor $k_S$ can be adjusted, and the performance of extended controller with different $k_S$ will be simulated and compared later. However, the DC link voltage should be regulated within a reasonable range for safe operation of the VSC-HVDC system. Therefore, the reference signal of the DC link voltage must be limited. As a result, larger DC link capacitors may be needed if this method is used to emulate a large inertia.

4 **Simulation Verification**

To verify the validity of the proposed strategy, a simulation model with the structure shown in Figure 1 has been built within Matlab/Simulink. The detailed parameters are given in Table 1. In the simulation, two loads are supplied by a synchronous generator and the HVDC converter station. One load can be switched on and off to observe the frequency change and response of the whole system.

Due to the high time constant of the power system, it takes a long time to reach a new steady state after the disturbance. To accelerate the simulation, the phasor model of power converter is used, in which only the fundamental frequency is considered. Conventional three-phase two-level topology is used during the simulation.

<table>
<thead>
<tr>
<th>Table 1: Parameters of simulated system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Items</td>
</tr>
<tr>
<td>-----------</td>
</tr>
<tr>
<td>VSC Rated power</td>
</tr>
<tr>
<td>Rated DC Voltage</td>
</tr>
<tr>
<td>Rated AC Voltage</td>
</tr>
<tr>
<td>Reactance</td>
</tr>
<tr>
<td>Capacitance</td>
</tr>
</tbody>
</table>

Figure 4 shows the simulation results. The switchable load is switched on at 5s. As a result, the synchronous generator has to compensate the changes of load immediately, as shown in Figure 4(a). Without GEC control, the converter station will have no contribution to this situation. However, since the mechanical power of prime mover can not be increased so fast as the electrical power, the rotor speed will be deceased suddenly due to the imbalance of power, as shown in Figure 4(c). The whole system may become instable when the change of the rotor speed is too big.

When GEC control is introduced to the VSC-HVDC system, the converter station can provide compensated active power very quickly as shown in Figure 4(a). So the imbalance between the mechanical power and the electrical power can be relieved, which results in a slower change of rotor speed and smaller rotor speed deviation as well. However, it is seen from Figure 4(d) that this compensated active power will lead to the decrease of DC link voltage, since the needed energy is absorbed from the DC link capacitors. Therefore, bigger DC link capacitors may be needed if larger emulated inertia is wanted. A compromise must be made between the volume of the DC link capacitors and the possible emulated inertia.
The effect of $k_S$ is also shown in Figure 4. By choosing appropriate $k_S$, the deviation of DC link voltage can be limited to the desired range.

5 CONCLUSIONS

An extended DC-link voltage controller based on the generator emulation concept has been proposed in this paper, which can emulate the inertia of SM and increase the stability of power system. Simulation results were provided to verify the validity of the proposed method. With the proposed extended controller, the HVDC converter station can provide fast active power to compensate the load change. Therefore, the frequency change can be limited within a safety range. The described concept will in particular be required and useful when connecting large scale offshore windfarms to the grid.

![Simulation results](image)

Figure 4: Simulation results

6 REFERENCES


COMPARISON OF GENERATOR SYSTEMS FOR SMALL POWER WIND TURBINES

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Keywords: Small vertical axis wind turbine, Generator concepts, Annual energy yields

Summary: This paper compares different generator systems for small power wind turbines in regard to the generated electrical energy. Due to the varying velocity of the wind there are long periods of small power with less than 10 % of the maximum power. Therefore the total electrical energy of a small power wind turbine is calculated with the varying wind power and the losses inside the generator system. Especially the losses independent of power become a high influence on the total production of electrical energy. The comparison of the results on the annual energy between different generator concepts indicated that geared permanent magnet synchronous generators with inverters are the best selection for the powertrain.

1 INTRODUCTION
So far the development of small wind turbines in Germany plays a subordinate role in comparison with other countries such as the USA and the UK. Meanwhile, the energy transition in Germany points the tendency to decentralized power generation from new renewable energy. With the growing awareness of expansion potentials of small wind turbines, which describes this power supply concept exactly, the small wind turbines obtain rising demand. The typical maximum power of these turbines is 1…100 kW. Most of them shall use the wind power above high buildings.

According to the construction of rotation axis the small wind turbines distinguish between horizontal and vertical installations. Experience shows that the horizontal wind turbines have been widely used in the past decades due to economic consideration, because they have mostly a higher power coefficient than vertical wind turbines.

The progress in the technical state of the vertical wind turbines is still limited by the reason of economic issues. Because of the significant energy losses it is mostly not profitable to use small wind turbines at low wind speed. Besides that the trade market is also not clearly regulated. There are no extra certifications for the small wind turbines. For this purpose an efficient variable speed system of vertical wind turbines is explicated in this paper. The requirements for the power train system of turbine, gearbox, generator, converter and brake of small wind turbines are analyzed and the possible solutions are developed based on those requirements. Furthermore the optimal choice of gearbox, generator and inverter is also described in this paper.

For the turbine the study is focused on the selection of appropriate turbines, which can provide the greatest energy at the prevailing wind speed. The gearbox and the generator are responsible...
M. Eng. M. Zhang, Prof. Dr. -Ing. C. Fräger

for the energy conversion from mechanical power to electrical power. In this case the choosing of the generator speed plays a major role for high energy efficiency. This is accompanied by the appropriate choice of the gear ratio and gear structure for efficient power conversion. In addition, the selection of the inverter is based on an efficient electric power transformation from the generator into public power supply. The generator speed also determines the size and the loss: High speeds ensure small dimensions with low losses.

2 DESCRIPTION OF WIND DATA IN GERMANY WITH A DISTRIBUTION FUNCTION

To calculate the expected energy production of wind turbines the knowledge about wind conditions is particular important. However, the multi-year measurements at a proposed site with thousands of individual values are impractical and the average annual wind speed, which provided by weather station is not sufficient for an accurate energy calculation. For this purpose there is an analytical approximation, namely Weibull distribution, based on statistics of the wind data to describe the wind speed frequency. The Weibull distribution function categorizes the regional wind conditions at a certain height in defined wind speed classes. The distribution density function $h_w$ is defined [1]:

$$h_w(v) = \frac{k}{A} \left(\frac{v}{A}\right)^{k-1} e^{-\left(\frac{v}{A}\right)^k}$$  \hspace{1cm} (1)

$A$ is a scale factor and corresponds to the characterizing wind speed of time series. $k$ describes the shape of the distribution. A bigger value for $k$ corresponds to less deviation of the mean wind speed. And $v$ represents the wind speed. The typical values of $A$- and $k$-factors for different regions in Germany are listed in the following table.

Table 1: Weibull parameters for different regions in Germany at 10 m height, according to [2]

<table>
<thead>
<tr>
<th>Region</th>
<th>$A$ [m/s]</th>
<th>$k$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coast</td>
<td>5.54</td>
<td>1.79</td>
</tr>
<tr>
<td>Central Uplands</td>
<td>4.5</td>
<td>1.66</td>
</tr>
<tr>
<td>North Germany Plain</td>
<td>4.03</td>
<td>1.58</td>
</tr>
</tbody>
</table>

The relative frequency $h_i$ depending on the classifying width $\Delta v$ can be calculated from the density function $h_w$. Here is anticipated $\Delta v = 1$ m / s, then:

$$h_i = h_w(v_i)\Delta v$$  \hspace{1cm} (2)

Figure 1 represents the relative frequency distributions with Weibull parameters from Table 1, which are dependent on the locations and the wind speed.
ROTORS AND ENERGY PRODUCTION OF VERTICAL WIND TURBINE

a. Selection of the rotor

The common rotor forms of vertical axis wind turbines are Savonius-, Darrieus- and H-rotor. Due to its low tip-speed ratio and the relatively small power coefficient with $c_{p,max} \approx 0.2$ [3] the Savonius-rotor is not suitable for energy production. Conversely, the Darrieus-rotor and H-rotor (a modification of the Darrieus-rotor) have a more effective power coefficient with $c_{p,max} \approx 0.4$ [3] because of its specific design with aerodynamic lift principle. The H-rotor with its own competitive advantages, e. g. a particularly simple structure, low cost and quiet performance, is gaining more commercial applications compared to the other rotors. The disadvantage of the H-rotor is the inability of independent startup, so that a traction device is necessary. A comparison of these vertical axis wind turbines is shown in Table 2. Overall, the H-rotor is selected for the further calculation.

Table 2: Comparison of the vertical axis wind turbines with different rotor types

<table>
<thead>
<tr>
<th></th>
<th>Savonius-rotor</th>
<th>Darrieus-rotor</th>
<th>H-rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power coefficient</td>
<td>low</td>
<td>high</td>
<td>high</td>
</tr>
<tr>
<td>Rotor</td>
<td>resistance rotor</td>
<td>lift rotor</td>
<td>lift rotor</td>
</tr>
<tr>
<td>Manufacturing</td>
<td>simple</td>
<td>complex</td>
<td>simple</td>
</tr>
<tr>
<td>Starting aid</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Noise</td>
<td>quiet</td>
<td>medium</td>
<td>quiet</td>
</tr>
<tr>
<td>Selection for the study</td>
<td>-</td>
<td>-</td>
<td>✓</td>
</tr>
</tbody>
</table>
b. Annual energy yield of a vertical wind turbine

The mechanical power, which the rotor converts from wind energy, is calculated as follows:

$$P_m = \frac{\rho}{2} F c_p(\lambda) v^3$$  \hspace{1cm} (3)

$\rho$ represents the air density, $F$ is the rotor swept area, $c_p$ the power coefficient, $\lambda$ the tip-speed ratio and $v$ is the wind speed. The power coefficient is calculated for a certain ratio of rotor speed and wind speed, or rather the tip-speed ratio. This means, the power coefficient (aerodynamic efficiency) is a function of the tip-speed ratio. For example, Figure 2 shows the power coefficient curve of an H-rotor. It is clear to find that the power coefficient reaches its maximum value $c_{p,\text{max}} = 0.4$ at the tip speed ratio of $\lambda_{\text{opt}} = 5.28$.

![H-rotor Power coefficient](image)

**Figure 2: H-rotor Power coefficient $c_p$ (approximation)**

Taking the mechanical and electric losses in the drive train into account, the electrical output power can be calculated as follows:

$$P_e = P_m \cdot \eta_{\text{mech.-electri.}}$$ \hspace{1cm} (4)

$$\eta_{\text{mech.-electri.}} = \eta_{\text{gearbox}} \cdot \eta_{\text{generator}} \cdot \eta_{\text{inverter}}$$ \hspace{1cm} (5)

These inevitable power losses are mainly caused by transmission, electric generator and possibly the frequency inverter. Small wind turbines with a power up to about 100 kW are usually equipped with helical gearboxes [3]. Here the efficiency of helical gearbox is considered as a constant 96%. On the opposite the efficiency of a generator is dependent on rotational speed, rating power and the type of generator. It rises with an increasing rational speed as well as the power rating and decreases in the partial load range. Therefore, the efficiency of the generator should be calculated according to the respective operating state. The efficiency of frequency inverter has a similar performance as that of generator and is also calculated for appropriate operating state.

The annual energy yield is resulting from the electrical power $P_e$ and the relative frequency $h_i$ by summing the individual class income for a year with $T_{\text{year}} = 8760$ h. The range from cut-in wind speed $v_E$ to cut-off wind speed $v_A$ is taken into consideration.

$$E = T_{\text{year}} \sum_{i = \frac{v_A}{\Delta v}}^{\frac{v_E}{\Delta v}} h_i \cdot P_{ei}$$ \hspace{1cm} (6)
4 Calculation of the Annual Energy Yield of a Vertical Axis Wind Turbine

a. The annual energy yield for variable speed generator systems

The variable speed generator systems consist of an electric generator connected with a frequency converter. Synchronous as well as asynchronous generators can be used to produce the electrical power. The frequency converter is responsible to transform the entire electrical power. Consequently, the total efficiency of generator system decreases by the inverter. In the mechanical drive train a gearbox can be settled between wind rotor and generator at the high speed shaft. Alternatively, a gearless drive is also possible to implement (direct drive). In recent years the demonstrations of direct drive systems with permanent magnet synchronous generator are already applied for small wind turbines. Figure 3 illustrates a fundamental structure of variable speed generator system.

\[ E = T_{\text{year}} \sum_{i} h_i \cdot \frac{\rho}{2} Fc_p(\lambda_{\text{opt}}) v_i^3 \cdot \eta_{\text{mech.-electri}}. \]  

In this case, the power coefficient \( c_p \) is chosen from the characteristic curve in Figure 2 and \( h_i \) can be obtained from (1) and Table 1. For more precise data it is needed to specify the location of the wind turbine and the dimension of the rotor.

b. The annual energy yield for fixed speed generator systems

In the fixed speed generator systems in Figure 4 generators are directly coupled to power grid. In this case, synchronous generators, asynchronous generators as well as pole changing asynchronous generators can be used. A gearbox between wind rotor and generator is usually necessary to limit the dimensions and weight of the generator to an acceptably dimension.

Figure 3: Variable speed generator system for wind turbine

In use of variable speed generator system the power output will be improved, because rotor speed is controlled and regulated depending on wind speed. This regulation allows the rotor to operate with its optimal power coefficient during partial load. In full load, the electrical power is limited to the generator rated power by stall control for small wind turbines. Thus, the calculation of annual energy yield for a variable speed generator system is:

Figure 4: Fixed speed generator system for wind turbine
The constant frequency power grid enforces the generator to produce a voltage with the same frequency, thus the speed of rotor is also constant. In contrast to variable speed performance, the maximum of rotor power coefficient by fixed rotor speed can be achieved only at a certain wind speed. Therefore, an optimum of rotor speed is especially significant for a maximization of power generation from restrictive wind availability. The rotor speed must be selected at a particular wind speed, where the highest power coefficient of rotor is achieved and at the meantime, the energy density of wind frequency distribution has its peak as well. For example, there is an energy density distribution of the North German plain in Figure 5 and the peak of energy density is reached at design wind speed of $v_A = 7 \text{ m/s}$. For different installation places the design wind speed varies depending on the wind frequency distribution. The optimum of rotor speed can be calculated according to (8), so that the maximum of power coefficient curve and energy density distribution are at the same wind speed.

$$n_{\text{opt}} = \frac{\lambda_{\text{opt}} \cdot v_A}{2\pi \cdot R} \quad (8)$$

![Graph](image)

**Figure 5:** Curves of power coefficient of the rotor, the frequency distribution and the energy density over the wind speed for the North German Plain

With an appropriate fixed rotor speed the rotor power coefficient is only dependent on wind speed. Then the calculation of the annual energy yield for a fixed speed generator system can be calculated as follows:

$$E = T_{\text{year}} \sum_{i=v_E/\Delta v}^{v_A/\Delta v} h_i \cdot \frac{1}{2} F_{Cp}(v_i) v_i^3 \cdot \eta_{\text{mech.-electri.}} \quad (9)$$
5 RESULTS AND EVALUATION

For a quantitative analysis of the annual energy yield of small vertical axis wind turbines there are two types of H-rotor with technical data shown in Table 3.

Table 3: Technical data of the H-rotors

<table>
<thead>
<tr>
<th>Rotor type</th>
<th>WRE.030</th>
<th>SpinWind</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated power</td>
<td>3 kW</td>
<td>10 kW</td>
</tr>
<tr>
<td>Diameter of rotor $D = 2R$</td>
<td>3.3 m</td>
<td>4.7 m</td>
</tr>
<tr>
<td>Height of rotor $H$</td>
<td>2.2 m</td>
<td>8.5 m</td>
</tr>
<tr>
<td>Rotor swept area $F$</td>
<td>7.26 m²</td>
<td>39.95 m²</td>
</tr>
<tr>
<td>Cut-in wind speed $v_E$</td>
<td>2 m/s</td>
<td>3 m/s</td>
</tr>
<tr>
<td>Rated wind speed $v_N$</td>
<td>14 m/s</td>
<td>11.5 m/s</td>
</tr>
<tr>
<td>Cut-out wind speed $v_A$</td>
<td>16 m/s</td>
<td>16 m/s</td>
</tr>
<tr>
<td>Rated rotor speed $n_N$</td>
<td>100-120 U/min</td>
<td>125 U/min</td>
</tr>
</tbody>
</table>

Several generator systems listed in Table 4 are compared according to the energy yield. The rated power of the used generator differ between fixed and variable speed systems. For rotor WRE.030 the generators are selected with a rated power about 2 kW by fixed speed systems. By variable speed systems the generators for the same rotor have a rated power about 5 kW. As the same, for rotor SpinWind the generators are appropriately selected with about 10 kW by fixed speed generator systems and 15 kW by variable speed generator systems. The efficiency profiles depending on the load of applied generators are shown in Figure 7 and Figure 8. Figure 6 shows the efficiency of the frequency inverter.

Table 4: Possible generator systems for small wind turbines

<table>
<thead>
<tr>
<th>Fixed speed generator systems</th>
<th>Variable speed generator systems</th>
</tr>
</thead>
<tbody>
<tr>
<td>gearbox + permanent magnet synchronous generator direct coupled to the grid</td>
<td>Low speed gearless permanent magnet synchronous generator with inverter</td>
</tr>
<tr>
<td>gearbox + asynchronous generator direct coupled to the grid</td>
<td>gearbox + permanent magnet synchronous generator with inverter</td>
</tr>
<tr>
<td>gearbox + pole changing generator direct coupled to the grid</td>
<td>gearbox + electrical exited synchronous generator with inverter</td>
</tr>
<tr>
<td></td>
<td>gearbox + asynchronous generator with inverter</td>
</tr>
</tbody>
</table>

Figure 6: Efficiency of frequency inverter for rotors WRE.030 with 5 kW and SpinWind with 20 kW
The above efficiency curves of the generators are estimated from real produced machines. Figure 9 shows the annual energy yields for the H-rotors with different generator systems.
Figure 9: Estimated annual energy yields for WRE.030 and SpinWind with different wind generator systems and locations.
For small wind turbines, it is important to find an energy efficient drivetrain, which ensures low loss and a high energy production. It can be seen in Figure 9, that variable speed generator systems lead to an impressive increase in the annual energy output of small wind turbines. They use the aerodynamically limited wind availability better than fixed speed generator systems. The results shows that, for all these considering locations the generator system with gearbox, permanent magnet synchronous generator and inverter is the best concept among these generator systems.

By using the pole changing asynchronous generator with fixed speed generator systems, a remarkable increase of annual energy yield for the bigger rotor (here for SpinWind) is achieved. In addition, the location has also a major impact on energy production. The results of installation on the coast region demonstrates a significant annual energy yield than the locations in the Central Uplands or the North German Plain.

The rotor power is proportional to the rotor swept area. However, due to the higher efficiency of generator systems at the higher power the energy production increases disproportionately to the rotor swept area. For example, by generator system with geared permanent magnet synchronous generator and inverter the energy yield increases to 6.2 times while the rotor swept area rises just by a factor of 5.5.

6 CONCLUSIONS
- In this paper the calculation of the annual energy yield for small wind turbines is described. The selection of the rotor, the optimal rotor speed, the transmission and the generator systems are discussed.
- The results for the selected H-rotors show different annual energy yields for the different generator systems.
- The highest annual energy yield is produced by variable speed generator system with geared permanent magnet synchronous generator and frequency inverter. The fixed speed generator system with pole changing asynchronous generator direct coupled to the grid improves the annual energy yield for the larger rotor.

7 REFERENCES
THE SCALABILITY OF LOADS ON LARGE DIAMETER MONOPILE OFFSHORE WIND SUPPORT STRUCTURES

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Keywords: Large rotors, Upscaling, Structural response, Ringing

1 INTRODUCTION

The trend is that the diameter of the offshore wind turbine rotor is increasing to capture more energy per turbine erected. Using theoretical upscaling as presented in [1], the increased rotor diameter will also give increased top-mass, larger rotor moment of inertia, lower rotational rotor frequencies, larger aerodynamic loads and a longer tower. The support structure needs to be stiffened to avoid the tower passing frequency of the blades. For a monopile this can be achieved with larger diameter and thickness of the monopile. However, an increased interaction between rotor and structure with non-linear wave loads and rotational frequencies may lead to both higher fatigue damage and larger extreme loads. The interaction effects discussed in this paper are not included in state of the art design practice and may significantly influence the cost of the monopile.

This study aims at identifying the loads driving the selection of the most cost critical dimensions of a large diameter monopile. The consequences of selecting a large diameter turbine on both the dimensions and the loading of the monopile are discussed. Negative design drivers, such as the diameter of monopile at water level, will be in focus since they have large effect on both natural frequencies and hydrodynamic loads. In addition, non-linear hydrodynamic loads may be critical for the balance between fatigue load and extreme load and may change from a fatigue load driven design to an extreme load driven design. This is discussed and exemplified by case studies.

The majority of research has been concentrated around the 5 MW reference wind turbine [2]. There is a lack of reference turbines that have a large rotor and large pile diameter similar to the commercial trend in the offshore wind industry. The study presented here is therefore using what we will refer to as the 6 MW Statkraft wind turbine. The blade design is based on the 5 MW reference wind turbine from NREL, but the weights of the nacelle and the blades are similar to a large industrial wind turbine.
2 METHOD

A number of numerical simulations were run using the dynamic analysis software FEDEM Windpower [3]. The model used is the 6MW Statkraft wind turbine, which characteristics are listed in the ‘Model characteristics’ subsection.

a. Integrated analysis

Integrated analysis is necessary to fully capture the interaction between different aspects of loading and response of the structure. Aerodynamic damping is one example of interaction, where the tower top response is damped by the aerodynamic response of an operating wind turbine rotor. This will be explained in further details in the ‘aerodynamic damping’ subsection.

The 6MW wind turbine was modeled in FEDEM Windpower [3], that has the capability to model aerodynamic loads, hydrodynamic loads, structural models, soil models and control system [4].

b. Model characteristics

The weights of the main component of the 6 MW Statkraft wind turbine are shown in Table 1 and the first 7 eigenmodes at stand-still are listed in Table 2.

<table>
<thead>
<tr>
<th>Table 1: Main properties of the wind turbine</th>
<th>Table 2: The eigenfrequencies and mode shapes of the 7 lowest structural eigenmodes at stand-still.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blade</td>
<td>Te</td>
</tr>
<tr>
<td>Nacelle</td>
<td>Te</td>
</tr>
<tr>
<td>Hub</td>
<td>Te</td>
</tr>
<tr>
<td>Top mass</td>
<td>Te</td>
</tr>
<tr>
<td>Tower</td>
<td>Te</td>
</tr>
<tr>
<td>Transition piece</td>
<td>Te</td>
</tr>
<tr>
<td>Foundation</td>
<td>Te</td>
</tr>
<tr>
<td>Soil pile</td>
<td>Te</td>
</tr>
<tr>
<td>Mode no</td>
<td>Frequency [Hz]</td>
</tr>
<tr>
<td>1</td>
<td>0.2815</td>
</tr>
<tr>
<td>2</td>
<td>0.2824</td>
</tr>
<tr>
<td>3</td>
<td>0.7567</td>
</tr>
<tr>
<td>4</td>
<td>0.7967</td>
</tr>
<tr>
<td>5</td>
<td>0.8509</td>
</tr>
<tr>
<td>6</td>
<td>0.9401</td>
</tr>
<tr>
<td>7</td>
<td>1.1551</td>
</tr>
</tbody>
</table>

The rotor diameter is 154 m, and the hub height is 106.5 m. The width of the support structure at the water level is approximately 7.15 m. The water depth analysed is 28.4 m, and the soil pile has a length of 42 m and a diameter of 8 m. Figure 1 illustrates the dimensions.

The controller used has a variable speed at below rated wind speeds, and fixed speed with pitching of the blades at above rated wind speeds. This controller is adapted from the NREL 5MW wind turbine [2] and has been tuned to get realistic wind turbine characteristics.

c. Simulation specifications

The model represents a 6MW offshore wind turbine geographically located at Doggerbank. The water depth, soil conditions and met-ocean data have been chosen accordingly to this location.

The JONSWAP wave spectrum was chosen to model the wave conditions, with the significant wave height ($H_S$) and the spectral peak period ($T_P$) corresponding to the considered
load case. The peak enhancement factor (or spectral peakedness, see [4]) was set to 3.3. The turbulent wind field was simulated with TurbSim [5] according to the considered load case.

The study is limited to natural frequency analysis and to load cases 1.2 and 6.1 in DNV-OS-J101[6]. Load case 1.2 is a fatigue limit state (FLS) analysis, and load case 6.1 is an ultimate limit state (ULS). Based on hindcast data, a coupled three parametric wave model has been estimated and used throughout the study both for FLS and ULS loads. Only a limited number of results will be presented to demonstrate the large turbine scale effects, which are not usually in focus in state of the art design.

The soil is modelled as vertically distributed springs and connected to the soil pile with p-y curves. These are based on geological data provided in [7] and in accordance with the API standards [8].

3  THEORETICAL MODELS ILLUSTRATED WITH RESULTS

d. Aerodynamic damping

Aerodynamic damping is a well known phenomena for horizontal axis wind turbines and is an important dynamic property to estimate response correctly when designing a support structure for an offshore wind turbine. The aerodynamic damping will reduce the dynamic response of the first elastic bending mode of the substructure in the fore-aft direction. This is illustrated in Figure 2, where the load spectra of the substructure at the point of maximum utilization (14 m below seabed) with only wave loads and no rotational speed is in blue, and
the response when a turbulent wind field and the rotor is rotating is in red. It is evident that
the response of the first elastic bending mode of the substructure is damped by the
eaerodynamic damping.

Figure 1: Aerodynamic damping of the NREL 5MW RWT, Statkraft 6 MW and DTU 10 MW RWT.

Figure 2: Load spectra in the soil pile, comparing an operating wind turbine (14 m/s wind) and a wind turbine at stand still (no wind).

Kühn [9] suggested the following relation to estimate the aerodynamic damping, $\zeta$:

$$
\zeta = N_b \frac{\rho \Omega}{4\pi f_0 M_0} \cdot \int_{R_{root}}^{R_{tip}} \frac{dC_l}{d\alpha} c(r) r \, dr
$$

where $N_b$ is the number of blades, $\rho$ is the air density, $\Omega$ is the rotational speed, $f_0$ is the
eigenfrequency, $M_0$ is the modal mass, $dC_l/d\alpha$ is the slope of the lift coefficient and $c(r)$ is
the chord distribution along the blade and $r$ is the radius. The relation is based on a stationary
rotor aerodynamics, and is only valid for wind turbines operating at a high tip speed ratio and
near the rated wind speed. For simplicity, we will assume the slope of the lift coefficient equal
to 2$\pi$, which is true for a thin foil in attached flow.

By applying theoretical scaling laws [1], the trend of aerodynamic damping for increasing
diameter of the rotor can be investigated. The classical scaling laws neglect 2$^{nd}$ order
aerodynamic effects and assume linear structural behavior. The mass scale as $s^3$, the lengths
as $s^1$ and the frequencies as $s^{-1}$. Using the theoretical scaling and Eq (1), the aerodynamic
damping is invariant of the rotor size.

In the future, new technology will emerge and it is therefore not expected that the
theoretical upscaling is correct in its prediction [1]. New materials are emerging, and this
may lower top mass, which increases the aerodynamic damping. Figure 1 illustrates the
aerodynamic damping ratio of the NREL 5MW reference wind turbine [2], the Statkraft 6MW
turbine and the 10 MW DTU reference wind turbine [10].

The Statkraft 6 MW wind turbine is more damped compared to the two reference wind
turbines. The main reason for this is the low mass of the wind turbine that is not designed
using scaling, but based on Siemens 6 MW public available data in [11]. It is expected that
using the same turbine technology, an upscaling of the aerodynamic damping will be approximately constant, similar to the trend from 5 MW to 10 MW.

e. Rotor loading due to 6P interaction with 2nd elastic tower bending mode

The excitation of the first bending mode of the support structure is a well-known issue when designing support structures. The support structure is therefore designed such that the first bending mode is not close to the rotational frequency of the rotor. Another less known issue is the interaction between the 6P frequency and the second bending mode of the support structure. The Campbell diagram in Figure 3 illustrates how the 1st bending mode is located in safe distance from the excitation loads at 1P and 3P. The 2nd bending mode is however relative close to the 6P frequency. For smaller wind turbines, the 2nd bending mode is higher and not close to the 6P. For the NREL 5 MW the 2nd tower bending mode the 1st substructure fore-aft is 0.25 Hz, and the 2nd substructure fore-aft mode is 1.65 Hz [12].

The excitation frequency at 6P is related to rotational sampling of the turbulent windfield. As the wind turbine blade is rotating, the low-frequency wind spectra is sampled at frequencies that are close to an integer of the rotor rotation, i.e. 1P, 2P, 3P etc. For a three-bladed rotor, this will be visible as 3P, 6P, 9P in the tower response.

The relative rotor loads have from previous research shown to be reduced as the rotor diameter increases [1]. The reason is that the spatial coherency of the incoming wind decreases. The studies referred in [1] only considered the rotor loads, and not a fully integrated offshore wind turbine, and does therefore not consider the loading due to the 6P interaction effects with the 2nd elastic bending mode.

Figure 4 shows the load response spectra for a fatigue load case simulation, comparing the response load of a stiff and a flexible structure. The maximum response load occurs at a frequency close to the wave dominant frequencies, but using a logarithmic scale contributions at higher frequency are visible. The aerodynamic load variation due to the blades passing in the tower shadow creates a cyclic load at the 3P frequency that is well known from wind turbine manufacturers. Having the 3P frequency close to the natural period is common
avoided by designing the structure to have the first eigenfrequency of the substructure below the 3P and above 1P. The Campbell diagram in Figure 3, which illustrates the 1P, 3P and 6P frequencies as well as the two lowest elastic substructure bending modes in fore-aft direction, shows that the first substructure bending mode is well below the 3P loading above rated.

In Figure 4, the 3P does not excite any of the structural modes. This is illustrated by the response for the stiff and the flexible structure being equal at the 3P frequency. However, the 6P frequency has a higher response for the flexible structure, relative to the rigid structure. The increase in response at the 6P frequency indicates that the 2nd substructure bending mode is being excited.

The interaction between the 2nd substructure mode and the 6P is an effect that is getting more important as the wind turbine increases in size. The reason is mainly the reduced 2nd substructure mode that is getting closer to the 6P frequency. A more detailed assessment of the significance for fatigue loads are left for further studies.

f. Extreme sea state – ‘ringing’

The method used to calculate the hydrodynamic loads on the structure is the ‘FNV 3rd order’ (Faltinsen-Newman-Vinje, [13]) and accounts for loads up to 3rd order, meaning that contributions of order higher than 3rd are truncated. The hydrodynamic excitation force calculated by the FNV method as found in [14] is given by the following equation:

\[
F_{FNV} = 2\pi \rho R^2 \int_{-h}^{0} u_t(z) dz + 2\pi \rho R^2 u_t|_{z=0} \zeta^{(1)} + \pi \rho R^2 \int_{-h}^{0} wu_z dz + \pi \rho R^2 \zeta^{(1)} (u_t z^{(1)} + wu_z - \frac{2}{g} u_t w_t)|_{z=0} + \pi \rho R^2 \frac{u^2}{g} u_t|_{z=0} \beta \left(\frac{h}{R}\right)
\]

\[O(\epsilon)\]

\[O(\epsilon^2)\]

\[O(\epsilon^3)\]

Where \( \rho \) is the water density, \( u \) and \( w \) are the horizontal and vertical first order velocity components, \( \zeta^{(1)} \) is the first order wave elevation, \( h \) is the water depth, \( R \) is the cylinder radius, \( g \) is the gravitational acceleration, \( \epsilon \) refers to the wave steepness and subscripts indicate differentiation. \( \beta \) is given by

\[
\beta \left(\frac{h}{R}\right) = \int_{0}^{\frac{h}{R}} (3\Psi_1(Z) + 4\Psi_2(Z)) dz
\]

with \( \Psi_1 \) and \( \Psi_2 \) defined in [15]. These functions represent the steady state and near field non-linear interaction with the cylinder for a deeply penetrating and constant diameter cylinder. Note that the sum of all third order load components is represented as a lump force at the free surface. This is an approximation which may be inaccurate for the evaluation of the higher order moment in shallow water.
The slenderness parameter, $\delta$, has been omitted for simplicity. It is shown in [14] that all terms of Equation (2) are the order of $\delta^2$. The FNV formulation assumes that $\epsilon \sim \delta$, which is equivalent to saying that the wave amplitude is close to the cylinder diameter. It is important to notice that this load model represents the nonlinear interaction between the long wave and the monopile column type of structure known to generate ringing for deep water oil and gas bottom fixed structures [16]. There are several studies based on a Morison load model with an underlying non-linear kinematic model valid for intermediate to shallow waters, refer for example Schloer [17]. However, these studies are not conclusive on the importance of non-linear hydrodynamic interaction effect in the water depth investigated.

FEDEM Windpower uses the deep water assumption shown in [15], which states

$$\beta\left(\frac{h}{R}\right) \rightarrow 4 \quad \text{for} \quad \frac{h}{R} \rightarrow \infty$$ (4)

For the present load case, the spectral peak period is $T_p = 15.2$ s, which gives a wavelength $\lambda=360$ m, and the water depth is 28.4 m. The ratio between water depth and wavelength is then less than 0.1, which according to [18] means shallow water conditions. Therefore the deep water assumption is inaccurate. It is not known whether this assumption is conservative or non-conservative. There is a need for further validation using flexible cylinders with different wave length – diameter ratios at shallow water depth – 20 to 40 meter.

In the following discussion, ‘1st term’ refers to the first line of Equation (2), ‘2nd term’ refers to the 2nd line and so on. The 1st term corresponds to the first harmonic, the 2nd term corresponds to the second harmonic and the 3rd and 4th terms correspond to the third harmonic (that is, varying with $\omega$, $2\omega$ and $3\omega$ respectively, $\omega$ being the wave circular frequency).

Extreme sea state conditions were simulated with a 50-year return storm, corresponding to the load case 6.1 in DNV-OS-J101 [6]. The met-ocean data was selected according to the geographical location of the turbine, leading to $H_s = 11.1$ m, $T_p = 15.2$ s, a wind speed of 48 m.s$^{-1}$ (with an extreme wind model) and a current of 92 cm.s$^{-1}$. Wind and waves are aligned and under these wind conditions the turbine is idling.

31 time series of 3 hours with different seeds for wind and wave spectrum were generated, the maximum loads at the point of maximum utilization (14 m below seabed) were extracted and fit into a Gumbel distribution [19]. The Gumbel plot for the shear force at the point of maximum utilization is shown in Figure 1. To assess the importance of non-linearities, the FNV method was checked against a linear model. The linear model is based on the Morison equation, where the waves are assumed linear and the kinematics are extrapolated to the true water surface by using Wheeler stretching. Added mass and drag coefficients were respectively set to $C_M = 1.8$ and $C_D = 0.8$ [20].

For the results obtained with the Morison equation, it appears that the maxima follow a linear trend (when plotted on a log(log) scale). For the results calculated with the FNV method, the lowest maxima seems to fit a linear trend, but the highest three do not follow that trend, showing how some highly non-linear phenomena are occurring for those realizations.
Figure 5: Gumbel fit for the Morison and the FNV methods

Figure 6 is an excerpt of the time serie of the bending moment at the point of maximum utilization where hydrodynamic loads where calculated in three different manners: with 1\textsuperscript{st} order terms of Equation (2) only, with 1\textsuperscript{st} and 2\textsuperscript{nd} order terms, and with 1\textsuperscript{st}, 2\textsuperscript{nd} and 3\textsuperscript{rd} order terms. In this figure it appears that the 3\textsuperscript{rd} order FNV method captures very high excitation around 3080 s. This dynamic amplification is due to the fact that the third harmonic terms (3\textsuperscript{rd} and 4\textsuperscript{th} terms in Equation (2)) oscillate at a frequency around 3/T_\textsubscript{P} = 0.2 Hz close to the first eigenfrequency of the structure 0.28 Hz, therefore exciting the first mode and creating large dynamic responses.

From Equations (2) and (4), it appears that all the terms of the excitation force are proportional to the square of the cylinder radius. However, the previous observations show that the response loads are very dependent on the eigenfrequencies of the structure which are
again linked to the rotor and monopile diameters. It is therefore necessary to carry an in-depth study about interaction between hydrodynamic loads and structural response when upscaling the wind turbine. Short wave length diffraction effects may also be of importance showing a different trend than the $R^2$ dependence [13].

The phenomena, known as ‘ringing’ [13], has been observed and studied for offshore oil and gas platforms, but has not been a concern within the wind industry. Because wind turbines have a larger top mass relative to substructure dimensions compared to oil and gas platforms, it is expected that their sensitivity to dynamic amplification in extreme wave events is also larger. Other effects, like viscosity, flow separation or shallow water conditions might also be of relevance, but is not considered dominating for the upscaling of the cylinder radius.

4 CONCLUSIONS

The results show that the upscaling in wind turbine design is not a linear and trivial task. In the extreme load analysis one needs to include the nonlinear hydrodynamic loads in the analysis to correctly account for higher order effects such as ringing. Ringing may have a large effect on the structural response of the wind turbine that could be driving the design of the foundation. The study focuses on large diameter monopiles, which is an inertia dominated structure, and cannot be transferred to drag dominated structures such as jacket foundations.

The detailed results from the fatigue analyses are not presented, but the main contributors to the fatigue damage of large wind turbines are considered. The focus is on the scalability of the dynamic response of the first and second bending mode of the support structure in the fore-aft direction.

The wind turbine is designed so that the first bending mode is only excited by the hydrodynamic loads. The aerodynamic damping will reduce the response. The aerodynamic damping ratio is constant for the increasing wind turbine, and if the structure is scaled according to the aerodynamics, only the hydrodynamic loads are giving the response.

The response of the second bending mode of the wind turbine has previously not been considered as an important fatigue contributor. As the offshore wind turbines are increasing in size, this will change. The second bending moment may get closer to the rotational sampling of the turbulence at 6P. This will increase the response and increase the fatigue damage of the structure.

The design presented here has a low utilization, both for fatigue and ultimate load analysis. However the balance between FLS and ULS driven loads are sensitive to non-linearities in the ULS loads and are thus showed to be important for a robust design procedure. The aim is to optimize the structure based on the knowledge gained through this study. The large turbine effects observed in this study is sensitivity to nonlinear waves, and the interaction between 6P and the 2nd substructure bending mode.

5 ACKNOWLEDGMENTS

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DETERMINISTIC AND STOCHASTIC APPROACH TO CONSIDER DYNAMIC SEA ICE LOADS IN DNV/GL/IEC/ISO STANDARDS

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Keywords: Sea Ice, Loads, Load Case, Dynamics, Frequency Lock-in

Summary: The present paper gives an overview of the requirements concerning the consideration of sea ice within the design of offshore wind turbines. Two different methods are analysed to cover the dynamic ice loading and structural response within the design. The simplified methods provided by the well-established standards and guidelines deliver conservative loads compared to the advanced method. The procedure of the advanced method is described herein. The consideration of different sea ice phenomena due to the dynamic ice-structure interactions within the load calculations of the complete turbine as well as the consideration of higher eigenmodes of the complete dynamical system are real benefits of the advanced method.

1 INTRODUCTION

With on-going realisation of offshore wind farms, areas where sea ice occurs are of interest too, e.g. the Baltic Sea. The focus on sea ice is important due to the fact that sea ice loading can become a design driving relevance both for extreme and fatigue loads for the entire offshore wind turbine. This is based on the dynamic behaviour of the ice floe and the interaction with the structure. The common experience and requirements are mainly based on oil & gas structures which are rather stiff compared to offshore wind turbines which are very elastic structures (especially monopiles) and very sensitive to excitations. Herein an overview of the requirements provided by well-established standards and guidelines is given and a simplified method is compared to an advanced method as described below.

2 STANDARDS AND GUIDELINES

2.1 Design basis

Within the design basis the environmental parameter influences of the considered offshore wind farm are documented. The respective ice data are often estimated from observation, measurements and historical data, collected in ice charts, e.g. in the Sea Ice Atlas.

As for wind and metocean data appropriate institutes can deliver sea ice data. The required long-term data such as extreme ice thickness with a recurrence period of 50 years can be derived by the application of statistical methods.
The sea ice has to be specified by the below parameters and properties to assure an appropriate consideration. For the Baltic Sea recommended values are given. The Baltic Sea herein covers the area between the Darss Sill and the Gulf of Bothnia, Finland and Riga, is composed of the Arkona Sea, Bornholm Sea and Gotland Sea with its western and eastern sub-region.

2.1.1 Ice thickness

Normal sea ice thickness can range from 0 to 30cm. Usually, ice thicknesses below 4cm are irrelevant because the failure by bending. Typical extreme sea ice thicknesses are about 30-70cm.

2.1.2 Ice bulk salinity

The salinity varies from 6‰ in the north to about 20‰ in the south.

2.1.3 Ice brine volume

The volume of enclosed saline brine influences porosity and density of sea ice. Typical brine volumes are in the range of 20ppt to 100ppt, depending on salinity, temperature, type and age of the ice.

2.1.4 Ice porosity

Naturally grown sea ice contains various inclusions and irregularities which lead to a porosity of typically 3ppt to 20ppt.

2.1.5 Ice temperature

The surface temperature of the sea ice cover is dominated by the air temperature and the gradient between the surface and bottom depends mainly on the history of air temperatures during recent days, the ice density and structure.

2.1.6 Ice density

The sea ice density depends on salinity, temperature and the age of the ice. Typical values are in range of 912kg/m³ to 925kg/m³.

2.1.7 Ice strength

Tensile strength, compressive (=crushing) strength and flexural (=bending) strength are basic properties of sea ice used in any analytical or empirical model. Approximation methods to calculate these values are given in [1].

2.1.8 Ice flow velocity

Sea ice in the Baltic Sea is mainly driven by wind. Currents are rather irrelevant. As an estimation of the sea ice speed the following relation may be used:

\[ v_{\text{ice}} \approx 0.025 \cdot v_{\text{wind,10m}} \]
Low ice speed below 0.04m/s can lead to intermittent crushing. Moderate ice speed in the range of 0.04m/s to 0.1m/s can lead to frequency lock-in. High ice speed of more than 0.1m/s can lead to continuous brittle crushing. See section 2.3 below.

2.1.9 Probability of occurrence

Sea ice cover does not occur every winter. Statistical analyses from several projects have shown that only once in every 5 years sea ice can be expected.

2.2 Design load cases

Sea ice loads acting on an offshore wind turbine are both static and dynamic. Static loads have their origin either in temperature fluctuations or changes in water level in a fast ice cover. Dynamic loads are caused by wind and current induced motions of ice floes and their failure in contact with the support structure.

Forces exerted on a structure by sea ice are to be evaluated for their effect on local structural elements and for global effects on the structure as a whole. The following loads due to sea ice shall be considered according to IEC 61400-3 [2], GL [3] and DNV [4] where relevant:

- Horizontal load from temperature fluctuations; thermal ice pressure
- Horizontal and vertical load from water level fluctuations or arch effect
- Horizontal load from moving ice floes
- Pressure from hummocked ice and ice ridges
- Dynamic ice loading

An overview of the design load cases considering the sea ice loads in combination with further environmental conditions and other design situations to comprise all load constellations within the design lifetime is given for GL [3] in Table 1 as well as IEC 61400-3 [2] and DNV [4] in Table 2.

Within GL [3] the dynamic interaction between the structure and the moving sea ice cover which leads to lock-in situations shall be analysed during power production of the turbine in DLC 9.1 and during idling situation in DLC 9.3.

The extreme sea ice conditions with a recurrence period of 50 years shall be considered and investigated in DLC 9.2 during power production of the turbine and in DLC 9.4 during idling situation. Especially the idling cases can lead to high loads due to the absence of the aerodynamic damping within the idling conditions of the turbine.

<table>
<thead>
<tr>
<th>Design situation</th>
<th>DLC</th>
<th>Wind condition</th>
<th>Marine conditions</th>
<th>Other conditions</th>
<th>Type of analysis</th>
<th>Partial safety factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power production</td>
<td>9.1</td>
<td>NWP $V_{in} \leq V_{hub} \leq V_{out}$</td>
<td>Current</td>
<td>Dynamic sea ice load</td>
<td>F/U</td>
<td>*/E</td>
</tr>
<tr>
<td></td>
<td>9.2</td>
<td>NWP $V_{in} \leq V_{hub} \leq V_{out}$</td>
<td>Current</td>
<td>Extreme sea ice $H_{ice} = H_{50}$</td>
<td>U</td>
<td>E</td>
</tr>
<tr>
<td>Parked (idling)</td>
<td>9.3</td>
<td>NWP $V_{hub} \leq 0.8 \cdot V_{ref}$</td>
<td>Current</td>
<td>Dynamic sea ice load</td>
<td>F/U</td>
<td>*/E</td>
</tr>
<tr>
<td></td>
<td>9.4</td>
<td>NWP $V_{hub} \leq 0.8 \cdot V_{ref}$</td>
<td>Current</td>
<td>Extreme sea ice $H_{ice} = H_{50}$</td>
<td>U</td>
<td>E</td>
</tr>
</tbody>
</table>

Table 1: Design load cases for sea ice design conditions according to GL [3]
### Table 2: Design load cases for sea ice according to IEC 61400-3 [2] and DNV [4]

<table>
<thead>
<tr>
<th>Design situation</th>
<th>DLC</th>
<th>Ice condition</th>
<th>Wind condition</th>
<th>Water level</th>
<th>Type of analysis</th>
<th>Partial safety factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power production</td>
<td>E1</td>
<td>Horizontal load from temperature fluctuations</td>
<td>NTM $V_{hub} = V_t +/- 2m/s$ and $V_{out}$</td>
<td>NWLR</td>
<td>U</td>
<td>N</td>
</tr>
<tr>
<td></td>
<td>E2</td>
<td>Horizontal load from water fluctuations or arch effect</td>
<td>NTM $V_{hub} = V_t +/- 2m/s$ and $V_{out}$</td>
<td>NWLR</td>
<td>U</td>
<td>N</td>
</tr>
<tr>
<td></td>
<td>E3</td>
<td>Horizontal load from moving ice floe at relevant velocities $H = H_{50}$ in open sea $H = H_m$ for land-locked waters</td>
<td>NTM $V_{hub} = V_t +/- 2m/s$ and $V_{out}$</td>
<td>NWLR</td>
<td>U</td>
<td>N</td>
</tr>
<tr>
<td></td>
<td>E4</td>
<td>Horizontal load from moving ice floe at relevant velocities $H = H_{50}$ in open sea $H = H_m$ for land-locked waters</td>
<td>$V_{in} &lt; V_{hub} &lt; V_{out}$ NWLR</td>
<td>F</td>
<td>*</td>
<td></td>
</tr>
<tr>
<td></td>
<td>E5</td>
<td>Vertical force from fast ice covers due to water level fluctuations</td>
<td>No wind load applied NWLR</td>
<td>U</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>Parked (idling)</td>
<td>E6</td>
<td>Pressure from hummocked ice and ice ridges</td>
<td>EWM (turbulent) $V_{hub} = V_1$</td>
<td>NWLR</td>
<td>U</td>
<td>N</td>
</tr>
<tr>
<td></td>
<td>E7</td>
<td>Horizontal load from moving ice floe at relevant velocities $H = H_{50}$ in open sea $H = H_m$ for land-locked waters</td>
<td>NTM $V_{hub} &lt; 0.7V_{ref}$</td>
<td>NWLR</td>
<td>F</td>
<td>*</td>
</tr>
</tbody>
</table>

Abbreviations used in Tables 1 and 2:
- DLC: design load case
- EWM: extreme wind speed model
- NTM: normal turbulence model
- NWP: normal wind profile model
- NWLR: normal water level range
- F: fatigue strength
- U: ultimate strength
- N: normal
- E: extreme
- *: partial safety factor for fatigue

In Annex E to the IEC 61400-3 [2] guidance is given for the application of the ice loads via formulas.
2.3 Dynamic ice actions on vertical structures

The brittle nature of ice may lead to periodic dynamic loading. Dynamic locking of the ice breaking frequency to the wind turbine eigenfrequencies shall be investigated. Dynamic simulation of the load case shall be performed.

The dynamic ice-structure interaction process is influenced by the ice velocity and the waterline displacement of the structure. Within EN ISO 19906 [1] three modes of interaction are described as follows.

Intermittent ice crushing, as shown in Figure 1a), can arise if a compliant structure is exposed to ice action at a low speed. The interaction involves a loading phase and an unloading phase. In the loading phase, the structure moves in the same direction as the ice. The ice edge experiences ductile deformations and the ice action gradually increases. The external ice action and the internal forces of the structure are usually in a static balance when the ice action reaches a maximum value. At the peak value of ice action, brittle crushing starts at the ice edge, leading to relaxation vibrations in the structure that decay during the unloading phase. The rate of decay depends on the total damping provided by the soil and the structure.

Frequency lock-in, as shown in Figure 1b), can occur at intermediate ice speeds, ranging typically from 0.04 m/s to 0.1 m/s, as the time-varying ice actions adapt to the frequency of the waterline displacements of the structure. The vibrations of the structure are typically sinusoidal in this condition. Similar to intermittent crushing the ice-structure interaction exhibits alternating phases of ductile loading and brittle unloading. The time history of the ice action depends on the characteristics of the ice and the structure.

Continuous brittle crushing, as shown in Figure 1c), occurs at higher ice speeds, typically when higher than 0.1 m/s. Both the ice action and the response are random.

![Figure 1: Modes of time-varying action due to ice crushing and the corresponding dynamic component of the structure response, taken from [1]](image-url)
3 METHODS

3.1 Simplified method

Within the above mentioned standards and guidelines [1], [2], [3], [4] simplified functions are provided to consider the dynamic ice loading and structural response. As an example the simplified method according to GL [3] is applied herein and used for comparison tasks.

The load oscillation period is equal to the support structure’s first natural period. The amplitude of the load oscillation may be assumed to be about $\frac{1}{4}$ of the static horizontal ice load with a mean value equal to $\frac{3}{4}$ of the sea ice load.

In Figure 2 typical Baltic Sea parameter with an ice thickness of $h=25\text{cm}$ are used.

![Simplified Ice Signal](image)

**Figure 2: Detail of the 600s time series with a simplified ice signal acc. to GL [3]**

The static horizontal ice load is calculated with the formulas provided in EN ISO 19906 [1], sections A.8.2.4.3.2 and A.8.2.4.3.3 as follows.

$$F_G = p_G \cdot A_N$$

$p_G$ is the ice pressure averaged over the nominal contact area associated with the global action

$A_N$ is the nominal contact area

$$p_G = C_R \left( \frac{h}{h_1} \right)^n \left( \frac{w}{h} \right)^m$$

$p_G$ is the global average ice pressure, expressed in megapascals

$w$ is the projected width of the structure, expressed in metres

$h$ is the thickness of the ice sheet, expressed in metres

$h_1$ is a reference thickness of 1m

$m$ is an empirical coefficient equal to -0.16

$n$ is an empirical coefficient, equal to $-0.50+h/5$ for $h<1.0\text{m}$, and to $-0.30$ for $h\geq1.0\text{m}$

$C_R$ is the ice strength coefficient, expressed in megapascals
3.2 Advanced method

The herein described procedure so-called “advanced method” is based on a representative offshore site with sea ice occurrence. The considered wind turbine is mounted on a monopile foundation.

The advanced method is divided into three steps. The first step includes the setup of the complete turbine model from blade tip to pile toe and delivers the eigenmode solution of the structure as an input parameter for the second step. Within the second step ice analysis specialists generate time series of ice forces under consideration of the respective structure and structural response. The ice time series are used for the load calculations of the complete structure within the third step. A detailed description of the whole procedure is given in the following sections.

3.2.1 First step - bladed model and modal analysis

A model of the considered wind turbine with support structure is set up with the integrated software package GH Bladed, Version 4.3, see Figure 2.

The secondary structures are considered as point masses at the respective elevations. The soil conditions are considered via non-linear p-y curves of the respective site. For the overall damping of the support structure 1% critical damping is applied.

After finalisation of the full turbine model set up a modal analysis is performed.

The energy-rich mode with the biggest impact in the load calculations is the first bending eigenmode of the substructure, see Figure 3. During ice-structure interaction, as described in section 2.3, the structure moves due to the pushing ice floe (loading phase). After reaching the maximum ice strength the structure relaxes in the unloading phase. The vibrations cause excitations of the structure in the first foundation eigenmode.

Dynamic ice-structure interaction depends, besides the ice speed, on the waterline displacement. For slender, elastic structures as offshore wind turbines the maximum waterline displacement usually corresponds to the second bending eigenmode of the substructure, coupled with the rotor blade eigenmodes. Therefore, ice-induced vibrations can be design-driving for both the substructure and the rotor blades. A schematic is given in Figure 4.
3.2.2 Second step - generation of ice signals

Eigenmode solutions of the respective structures (including rotor dynamics) are provided to the ice analyst. The eigenmode solutions as well as site related ice conditions are used for the consideration of the ice force interaction between the ice edge and the foundation structure. A detailed description of the used software known as PSSII (Procedure for dynamic Soil-Structure-Ice Interaction) and the applied methods are given in [5]. As a result stochastic simulations were conducted for design load cases required by the standard. A few examples of ice forces are shown as follows:

![Figure 5: Intermittent crushing – details of the generated ice force time series](image)

![Figure 6: Frequency lock-in – details of the generated ice force time series](image)

![Figure 7: Continuous brittle crushing - details of the generated ice force time series](image)

It is shown that the dynamic ice actions on vertical structures as described in EN ISO 19906 [1] and section 2.3 are considered in the generation of the ice forces.

3.2.3 Third step - load calculations

The load calculations are performed with GH Bladed. The wind turbine model as described in section 3.2.1 is used together with the controller of the pitch-regulated turbine. A turbulent stochastic wind field is generated and used within the load calculations. The ice forces described in section 3.2.2 are used as external forces acting at waterline on the structure. The longitudinal direction (main direction of the ice floe) and a lateral component are considered.

4 RESULTS

4.1 Simplified vs. advanced method

In Figure 8 the simplified method as described in section 3.1 is compared with the advanced method as described in section 3.2. Exemplarily, the resultant bending moment at mudline is shown. For all load calculations an ice thickness of h=25cm is considered. For the advanced method four load calculations with different ice velocities are used. The simplified method and the worst-case of the advanced method deliver nearly the same extreme loads. With respect to fatigue loads the advanced method delivers smaller load cycle ranges.
4.2 Time series with and without sea ice

The load calculations of a typical offshore wind turbine in the south Baltic sea consider a power production situation at rated wind speed. The blue time series represents an ice thickness of h=25cm. The red time series uses only turbulent wind. Figure 9 shows a comparison of the resultant bending moments at mudline. The ice load time series delivers a higher load level than the wind load time series. The sea ice impact delivers nearly 50% higher loads (peak value).

4.3 Different ice phenomena with corresponding bending moments at mudline

In Figures 10, 11 and 12 details of the shear forces at water level and the governing bending moments at mudline are shown which correspond to the generated ice force time series shown in Figures 5, 6 and 7.
Figure 10: Intermittent crushing – governing bending moment at mudline

Figure 11: Frequency lock-in – governing bending moment at mudline

Figure 12: Continuous brittle crushing – governing bending moment at mudline
Figure 13 shows a comparison of the maximum bending moments based on different ice phenomena. The results are evaluated on basis of the time series shown in Figures 10, 11 and 12. The highest loads are obtained by the frequency lock-in case. The extreme load achieved by the continuous brittle crushing is about 25% lower. Both load calculations consider an ice thickness of \( h = 25 \text{cm} \). The intermittent crushing load case delivers very low loads due to the fact that the considered ice thickness is only \( h = 9 \text{cm} \).

![Comparison of Maximum Bending Moments due to Different Ice Phenomena](image)

**Figure 13: Comparison of the maximum bending moments due to different ice phenomena**

### 5 Conclusion

It is shown that the advanced method delivers nearly the same extreme loads compared to the simplified method for an ice thickness of \( h = 25 \text{cm} \). Furthermore, the load cycle ranges become considerably smaller due to the stochastic process.

The advanced method considers higher eigenmodes of the complete dynamic system during ice-structure interaction. The dynamic ice-structure interaction process is influenced by the ice velocity and the waterline displacement of the structure. For slender, elastic structures like offshore wind turbines the maximum waterline displacement usually corresponds to the second bending eigenmode of the structure, coupled with the rotor blades. Therefore, consideration of the coupled eigenmodes during ice-structure interaction is important for the calculation of the design-driving loads for both substructure and rotor blades.

Furthermore, the advanced method is able to consider different phenomena during ice-structure interaction, like intermittent ice crushing, frequency lock-in and continuous brittle crushing. The consideration of ice action speed, elastic ice response, nonlinear ice failure process and soil-structure interaction provides putative realistic data. This and the reliability of the advanced method shall be verified via measurements.
6 REFERENCES

WAVE INDUCED FATIGUE LOADS ON MONOPILES -
NEW APPROACHES FOR LUMPING OF SCATTER TABLES AND SITE SPECIFIC INTERPOLATION OF FATIGUE LOADS

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Keywords: Frequency domain, wave loading, monopile, lumping, site interpolation

Summary: Offshore wind turbines are subject to dynamic excitation from wave loads. Especially when monopile substructures are used, significant fatigue loads can be induced by waves, which are then governing the design. Calculations in the frequency domain are very efficient to compute such wave induced loads and by applying some simplifications, very compact equations can be derived for the determination of fatigue loads. Based on such simplified formulas further methods for lumping of scatter diagrams and for interpolations of fatigue loads for different positions within a wind farm are presented in this paper.

1 INTRODUCTION
As monopile substructures for offshore wind turbines gain market share in ever deeper waters [1], wave excitation becomes more and more important. It is therefore crucial to gain good understanding of the relevant parameters and to develop tools for rapid calculation of fatigue loads, which are typically governing for the structural dimensions. Calculations in the frequency domain are a helpful method in this respect and with some simplifications, which can be applied to offshore wind turbines supported by monopiles, very compact equations can be derived to compute wave induced fatigue loads. Based on a method developed by the author in [2] some practical applications of the method are discussed in this paper.

2 FREQUENCY DOMAIN ANALYSIS
Calculations in the frequency domain are used frequently in the Oil&Gas industry to compute wave induced response of offshore structures. General information about this method can be found in Hapel [3] and Barltrop [4]. The general theory of frequency domain calculations is not repeated here for brevity. Symbols in general follow the notation used by Hapel [3], unless noted otherwise.
3 A SIMPLIFIED METHOD TO DETERMINE WAVE INDUCED FATIGUE LOADS

The method developed in [2] is briefly summarized in the following. As this paper does not include all notations and relevant background information, it should ideally be read together with [2].

3.1 Assumptions for the proposed simplified method

The following simplifications are made, which are acceptable for offshore wind turbines mounted on monopile substructures:

1. Only the first mode is considered for response calculations, as higher modes are outside of the frequency content from wave excitation.

2. Low structural damping is assumed, typically a modal damping ratio of $\xi_0=1.0\%$ is used for offshore wind turbines founded on monopiles.

3. As structural damping is low, the response can be assumed to be narrow-banded and only the region close to the first natural frequency $\omega_0$ is relevant for all terms which are a function of $\omega$.

4. Drag loading is neglected as this is small for fatigue waves.

5. Hydrodynamic damping is neglected as the velocity of the structure is small.

3.2 Summarized formula

With the assumptions listed above, a simplified expression can be derived to compute wave induced fatigue loads. The detailed derivation and definitions can be found in [2]. Fatigue loads are calculated as damage equivalent loads (DEL). See [2] regarding the conversion to other number of reference cycles.

$$\Delta M_{eq,Nref,Hz} = 1.8825 \cdot \sqrt{S_{z0}^{\omega_0}} \cdot \frac{\Phi_0(z_{unc})}{K_0} \cdot \frac{1}{\xi_0} \cdot \omega_0^{3/4} \cdot H_{a,0} \cdot H_{TB}$$

(1)

With:

$$H_{TB} = \omega_0^2 \int_{z=0}^{z_{unc}} \Phi_0(z) \cdot \mu(z) \cdot z \cdot dz$$

Transfer function tower bottom

(2)

$$H_{a,0} = \rho \cdot \omega_0^2 \int_0^{d} C_M(z) \cdot \left[ \frac{\pi}{2} \cdot \frac{D(z)^2}{4} \right] \cdot \eta_0(z) \cdot \Phi_0(z) \cdot dz$$

Hydrodynamic transfer function

(3)

These formulas can be easily evaluated analytically, only modal analysis needs to be performed numerically.
3.3 Conclusions

Some important conclusions can be drawn from this expression:

1. DELs are proportional to \( \left( \frac{1}{\xi_0} \right)^{0.5} \) – i.e. if damping is e.g. doubled, then fatigue loads decrease by 30%! This illustrates that damping is one of the major factors to assess reliably. Furthermore, damping assumptions should not be overly conservative to enable an economic design.

2. Damage is proportional to the square root of spectral wave energy at the first natural frequency. This is important when lumping of the scatter diagram shall be performed, as will be shown later.

3. Mode shape and hydrodynamic properties around the still water level are of particular importance. The hydrodynamic transfer function is linearly proportional to mode shape amplitudes in the wave loaded zone, as can be seen from Eq. (3). Reducing modal amplitude below still water level is therefore particularly helpful to reduce fatigue loads.

4. In total, fatigue loads are proportional to \( \omega_0^3 \), when all other parameters are unchanged. This is an indication that a large head mass (from the turbine) is not necessarily disadvantageous, as this decreases the natural frequency.

4 LUMPING OF THE SCATTER DIAGRAM

In order to decrease the required number of calculations, the scatter diagrams (wind speed vs. wave height and wave height vs. wave period) are often condensed (or “lumped”). Ideally, lumping of the \( H_S-T_P \)-diagram is done in a way that both the quasi-static contribution and the dynamic (resonant) contribution is captured.

![Figure 1: Quasi-static and dynamic moment lines for a wave-loaded monopile](image)
M. Seidel: Wave induced fatigue loads on monopiles – New approaches for lumping of scatter tables and site specific interpolation of fatigue loads

This is schematically shown in Figure 1. The quasi-static moment line does only show internal member forces below the highest point of wave load attack. The dynamic moment line shows load all over the structure, this moment line is dominated by the structural response in the first mode.

4.1 Weighting

Damage incurred by a certain sea state is proportional to $(DEL)^m$, where $m$ is the negative inverse slope of the S-N curve. Codified S-N-curves for welded details have values of $m=3$ and $m=5$. In order to calculate DELs, a S-N-curve with only one slope must be used, and $m=4$ is then used as the representative value. This must be considered when lumping sea states.

4.2 Quasi-static lumping: Equivalent significant wave height

Wave loads on individual members do have a quasi-static effect, i.e. the wave loads cause internal member forces in the members where they apply. Wave loads are described by the well-known Morison’s equation (4), see e.g. Hapel [3] for details.

$$F(t) = \frac{\pi}{4} \cdot \rho \cdot C_M \cdot D^2 \cdot \ddot{u}(t) + \frac{1}{2} \cdot \rho \cdot C_D \cdot D \cdot u(t) \cdot |u(t)|$$

Equation (4)

Wave induced forces are therefore proportional to the water particle acceleration (Eq. (5)) for the inertia term and the square of water particle velocity (Eq. (6)) for the drag term. Water particle acceleration and velocity are linearly dependant on wave amplitude for linear waves:

Water particle velocities (horiz.): $$u(t) = \zeta_u \cdot \omega \cdot \frac{\cosh(k \cdot (h + z))}{\sinh(k \cdot h)} \cdot \cos(k \cdot x - \omega \cdot t)$$

Equation (5)

Water particle accelerations (horiz.): $$\ddot{u}(t) = \zeta_u \cdot \omega^2 \cdot \frac{\cosh(k \cdot (h + z))}{\sinh(k \cdot h)} \cdot \sin(k \cdot x - \omega \cdot t)$$

Equation (6)

It follows that quasi-static fatigue loads can be assumed to be proportional to $H_S^\lambda$, where $\lambda$ is 1 if the wave loading is inertia dominated and 2 if the wave loading is drag dominated.

An equivalent wave height can therefore be computed for each wind speed as:

$$H_{s_{eq}} = \left( \frac{\sum H_{s,n}^{\lambda_m} \cdot p(n)}{\sum n \cdot p(n)} \right)^{\frac{1}{\lambda_m}}$$

As wave loads on monopiles are inertia dominated, $\lambda = 1$ applies.
4.3 Resonant (dynamic) lumping: Equivalent peak period

Additional to the (local) quasi-static contribution, global dynamic excitation does occur. For slender structures, like monopiles, this is often the dominant effect. For stiff structures, like jackets, this effect may be negligible.

Dynamic fatigue loads are proportional to \( \sqrt{S_{\omega_0}(\omega_0)} \) (see Eq. (1)), where \( S_{\omega_0}(\omega_0) \) is the spectral energy of the wave spectrum at first natural frequency. This has been derived for a narrow band response, which is a good approximation in case dynamic excitation is significant.

Weighting on basis of the spectral value at the natural frequency is the idea of following approach, as dynamic fatigue loads are proportional to \( \sqrt{S_{\omega_0}(\omega_0)} \). The following relationship does then apply:

\[
\sqrt{S_{\omega_0}(\omega_0)}_{eq} = \left( \frac{\sum_n \left[ \sqrt{S_{\omega_0}(H_{S,n} \mid T_{P,n} \mid \omega_0)} \right]^n \cdot p(n)}{\sum_n p(n)} \right)^{1/m}
\]

(7)

with \( p(n) \): Probability for sea state \( n \) with \( H_{S,n} \) and \( T_{P,n} \).

The spectral values \( S_{\omega_0}(H_{S,n} \mid T_{P,n} \mid \omega_0) \), depending on wave height, peak period and first natural frequency, have to be determined for each entry in the scatter matrix. As the gradient of the wave spectra is high adjacent to the peak period, a peak period bin size of one second (which is typically the case) will not lead to accurate results. Hence, a refinement of the peak period by a factor of 10 is recommended by means of a spline interpolation.

The spectral values are computed based on the refined scatter matrix. After the application of the weighting formula shown above, an equivalent spectral value is determined for the respective wave height row of the matrix. Based on the equivalent spectral value and the significant wave height the equivalent peak period can be recalculated. This is a back-calculation (iterative procedure) which ensures that the target value for the spectral energy at natural frequency is achieved.

Two solutions exist for this back-calculation, one \( T_P \) value on the ascending part of the spectrum and one \( T_P \) on the descending part (see Figure 2 for an example). For monopile configurations with Multi-MW turbines the higher \( T_P \) in general is the representative one as the natural frequency of the turbine is low. This must be identified for each specific case.
When fatigue loads for a complete wind farm need to be calculated, it is often not practical (or even impossible) to perform load iterations for every position. A few positions (with varying structural properties, soil conditions and water depths) are then calculated and these loads must be applied to all structures. This can be done conservatively or some sort of interpolation must be adopted.

One possible parameter to choose for interpolation is natural frequency. The results from five different complete load simulations for a North Sea wind farm are shown in Figure 3. The data sets cover different water depths, but also different load iterations with differences in structural dimensions. The general trend shows what can be expected: The fatigue loads decrease with a higher natural frequency, which can be attributed to smaller wave excitation. But although the general trend is captured, large differences can be seen for the regression line vs. actual results. Such an interpolation quality is not suitable for design purposes.

Water depth as an parameter provides even poorer correlation, as the impact of soil stiffness is not captured in this case.
A better parameter can be found based on the method derived before. If Eq. (1) is simplified, it can be stated that fatigue loads are proportional to the following parameter $S$:

$$S = \frac{\omega_0^{0.75}}{K_{0,\text{norm}}} \cdot \sqrt{S_{\xi_0}(\omega_0)}_{eq} \cdot \sqrt{\frac{1}{\xi_0}} \cdot H_{T^B} \cdot H_{a,0}$$

(8)

This expression is valid if the fatigue loads are governed by dynamic wave excitation, which is often the case for monopiles. If wind loads are governing (i.e. for a very stiff system) then this site parameter would not be a good choice.

This site parameter can be used to evaluate fatigue loads for all positions within a wind farm. Load simulation can be performed for the sites having minimum and maximum site parameter and interpolation can be used in between. Interpolation can be performed with much simpler tools (e.g. Excel) as all steps can be done in a spreadsheet, except for the modal analysis for each site. The latter can be performed in a structural analysis program (or even that can be done in Excel, but this requires a bit more work).

Results are shown in Figure 4 for the same data set as used for Figure 3. It can be seen that the correlation is now very good, which makes the site parameter a suitable parameter to perform interpolation, esp. during the stage where design iterations are performed and quick response cycles from structural designer to load calculations engineer are required.
In this paper, new approaches for lumping of a scatter diagram (scatter matrix) and interpolation of site specific fatigue loads have been demonstrated based on on frequency domain considerations. Simplifications relevant for monopile substructures have been used to determine a compact formula which allows rapid calculation of wave induced fatigue loads. These approaches allow for more accurate and fast calculations of wave induced fatigue loads for offshore wind turbines with monopile support structures.

7 REFERENCES

MEASUREMENT BASED INVESTIGATIONS OF DESIGN LOAD PARAMETERS FOR OFFSHORE WIND TURBINES

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Keywords: Offshore wind energy, wind parameter, wave parameter, sea state statistics, design wave

Summary: Significant load parameters for OWT are the reference wind speed as well as the significant and maximum wave height with a return period of 50 years. For the statistical analysis of extreme values with very long return periods, the expiring distribution areas far away from the mean value are important. Thereby, the type of distribution of the population and extreme values of this population may not always coincide. Thus, starting with different extreme value periods, annual extreme value distributions are extrapolated and load parameters according to [1, 2] and [3] are calculated. The sensitivity of load parameters towards the extrapolation basis is shown and recommendations for suitable reference periods are given.

1 INTRODUCTION

The knowledge of environmental conditions is very important for a safe and economical design of offshore wind turbines (OWT). Usually, the structural loading of OWT is determined by simulations based on local measurement series or so called hindcast data. Important input parameters of these simulations are wind speed and wave height. For fatigue analyses and operating conditions these parameters can be determined directly from measurement series. For extreme load parameters with return periods of 50 or 100 years measurement data naturally do not exist. These parameters are extrapolated beyond the actual period of observation by using statistical methods. For a reliable extrapolation sufficiently long measurement series are required. The uncertainty in estimation of such extreme events and the importance of sufficiently long measurement series shall be demonstrated on the FINO 1 measurement platform. The FINO 1 platform was built in 2003 because of the lack of long term measurements of wave heights and other parameters. This research platform was designed for a 100-year wave with an assumed wave height of 17,0 m. Nowadays, 100-year waves are assumed to be much higher at this location. Selfmade analyses of FINO 1 measurement data have indicated already for the 50-year wave a calculated wave height of approximately 19,0 m. In this case the value of a 100-year wave amounts more than 20,0 m, provided that no wave breaking occurs. These results agree much better with seastate measurements in extreme storm events in recent years.
In this paper measurement series of the FINO1 platform in the period between 1.1.2004 and 1.1.2014 are analysed. Therefore, statistical methods for the determination of extreme load parameters for OWT are presented, applied to measurement data and investigated for their applicability. Finally, recommendations for relevant load parameters are given.

2 DATABASE

The following investigations based on the measurement time series of the research platform FINO 1 in the southern North Sea (N 54° 0,86’ E 6° 35,26’). The platform is operating on behalf of the Federal Ministry for the Environment, Nature Conservation, Building and Nuclear Safety (BMUB) in about 30 m water depth. The measurement time series are provided by the Federal Maritime and Hydrographic Agency (BSH).

The wind speed is measured vertically resolved by eight cup anemometers installed in 33 - 100 m height. Measurement time series of 10-minutes mean values of wind speed are provided for each height. The significant wave height is collected by a sea state buoy (Datawell Directional Waverider buoy), which is anchored in about 200 m distance to the platform. The significant wave height is defined as four times the standard deviation of all wave heights of a wave spectrum, see [1]. This corresponds to approximately the average of the 1/3 highest waves. The measured time series of significant wave height consists of so called 30-minutes values, determined from a 20 minutes long wave spectrum. In order to obtain statistically reliable values a multi-year observation period from 1.1.2004 - 1.1.2014 is analysed.

The characteristic load parameters are determined according to [1, 2] and [3] based on 10-minutes mean value of wind speed and 3-hours mean value of significant wave height. According to the guidelines a stationary sea state can be expected for the period of three hours. Measurement failures are eliminated from the time series and 3-hours mean values are calculated from the available 30-minutes values of the significant wave height. The following analyses base on the these prepared, 10-years long measurement time series of 10-minutes mean values of wind speed and 3-hours mean values of the significant wave height.

3 METHODS

For extreme load parameters with return periods of 50 or 100 years measurement data naturally do not exist. Usually, statistical methods like the total sample, the annual maxima or the peak over threshold method are used for the estimation of these parameters, cf. [4]. Inspite of a 10-years data basis, the measurement period is too short for a reliable use of the annual maxima method. The total sample method use the whole data observed during a number of years. The main disadvantage of this method is the fact that the distribution type of a parent population and of extreme values of this population does not have to be identical, see Figure 1. For the statistical analysis of extreme values with large return periods the areas far away from the mean value of a distribution are important.
Figure 1: Histogram of the 10-minutes mean values and 4-weeks extreme values of wind speed in 100 m height and various probability density functions

In contrast to the total sample method, in this paper extreme values of different reference periods are determined and extrapolated to annual extreme value distributions. In this way, it is ensured that in particular the areas of high quantiles are suitable described by the distribution function. At the same time the method do not rely on measurements of many decades, as is the case using the annual maxima method.

The extreme values of wind speed and significant wave height are well approximated by a Gumbel-distribution (extreme value distribution type I) for maximum values, cf. Figure 1 and [5]. This visual impression is confirmed by the chi-square and the Kolmogorow-Smirnow goodness of fit test. In the following the relevant load parameters are determined according to eq. (1) on the basis of the Gumbel distribution function.

\[ F_x(x) = \exp \left(-\exp \left(-a \cdot (x - u)\right)\right) \]  

This distribution function is adjusted by the parameters $a$ and $u$ to the measurement distribution. The distribution parameters can be determined directly from the distribution or by a regression analysis. In this paper, the distribution parameters are determined directly from the distribution. In [6] the parameters are also determined by a linear regression. Both methods lead to similar results.

The distribution parameters are determined depending on the standard deviation $\sigma_{Ext}$ and the mean value $m_{Ext}$ according to eq. (2) and (3).

\[ a = \frac{\pi}{\sqrt{6} \cdot \sigma_{Ext}} \]  

\[ u = m_{Ext} - \frac{0.577216}{a} \]  

For the selected distribution type the shape of the distribution is independent of the reference period, i.e. the standard deviation remains constant. A modified reference period leads to a parallel shift of the distribution function. The annual extreme value distribution is determined with Eq. (4) to (6), see also [7].
The parameter $N$ represents the number of observed extreme values per year. The statistical values and the distribution parameters of the annual extreme value distribution of wind speed and significant wave height are based on 4-weeks extreme values as shown in Table 1.

Table 1: Distribution parameters of the 1-year extreme value distribution of wind speed in 100 m measurement height and significant wave height based on 4-weeks extreme values

<table>
<thead>
<tr>
<th>Parameter</th>
<th>$m_{\text{Ext}}$ [m/s]</th>
<th>$\sigma_{\text{Ext}}$ [m/s]</th>
<th>$a$ [s/m]</th>
<th>$u$ [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wind speed</td>
<td>31.52</td>
<td>4.24</td>
<td>0.30</td>
<td>29.61</td>
</tr>
<tr>
<td>Significant wave height</td>
<td>7.00</td>
<td>1.54</td>
<td>0.83</td>
<td>6.30</td>
</tr>
</tbody>
</table>

The 98%-quantile for the reference period of one year corresponds to the characteristic value with a return period of 50 years, see [1]. Thus, the characteristic load parameters can be determined with the inverse distribution function of the annual extreme values $F_{u}^{-1}(x)$, see Eq. (7).

$$F_{u}^{-1}(x) = u - \frac{1}{a} \cdot \ln[-\ln(x)]$$

Due to the parallel shift of the distribution for changed reference periods, different reference periods of extreme values of different reference periods can be used as extrapolation basis. Furthermore it is possible to consider only the extreme values of the winter half-year by adapting the appropriate quantiles. The regarded quantile of a characteristic 50-years value bases on 6-months reference periods is given by Eq.

$$q = l \cdot \frac{1/2}{50 \text{ jahre}} = 0.99$$

The load parameters obtained with the proposed method are partly compared with the results of the peak over treshold method. Using the peak over threshold method only values above a threshold are described by a distribution function, see also [1]. For the Gumbel distribution the parameters can be determined according to eq. (2) and (3). However, the quantile is calculated according to eq. (9). Where $R$ is the return period and $\lambda$ the ratio of observed extreme events divided by the observation period in years.
\[ q = I \cdot \frac{1}{\lambda \cdot R} \]  

(9)

4 RESULTS

a. Reference wind speed \( V_{\text{ref}} \)

At the site FINO 1, extreme wind speeds are well approximated by a *Gumbel* distribution, see Figure 1. It can be assumed that extreme values of a long reference period are more suitable for extrapolation to an annual extreme value distribution. This is well founded because for the statistical analysis of extreme values areas far away from the mean value of a distribution are important. However, a sufficient number of measurement data is necessary to ensure a reliable determination of the distribution function. In Figure 2 mean values, standard deviations and 98% quantiles of the annual extreme value distribution of wind speed based on different observation periods are shown.

![Figure 2: Mean values, standard deviations and 98% quantiles of the annual extreme value distribution of wind speed based on different observation periods and evaluation methods](image)

Obviously for sufficiently long time periods the mean values and quantiles trend to a limit. In this case, the two methods for determining the distribution parameters (analysis of the distribution function and linear regression) indicate a good agreement. The assumption that extreme values of a short reference period are less suitable for extrapolating annual extreme value distributions is confirmed. For wind speed the use of 1-week or 4-weeks extreme values as basis of extrapolation has little influence on the result. Using 4-weeks extreme values as extrapolation basis the reference wind speed at FINO 1 amount to \( V_{\text{ref}} = 42.5 \) m/s.

A height resolved study of wind speeds for the considered measurement period is still pending. However, studies of shorter measurement periods indicate a lower power law exponent than previously assumed, cf. [5].
b. Extreme significant wave height $H_{s50}$

The distribution of 4-weeks extreme values of significant wave height can also be approximated by a Gumbel distribution, cf. Figure 3. The chi-square and the Kolmogorow-Smirnow goodness of fit test confirm this assumption.

Due to the identical distribution type, the methods described in the previous section can also be applied to the extreme value distribution of the significant wave height. In Figure 2 mean values, standard deviations and 98% quantiles of the annual extreme value distribution of significant wave height based on different observation periods are shown.

Mean values and quantiles of the extreme significant wave height also trend to a limit, provided a sufficiently long reference period. The two methods for determining the
distribution parameters (analysis of the distribution function and linear regression) indicate a good agreement, too. The little differences of the annual extreme value distribution based on 8-weeks extreme values can be caused by a less number of measurement values. An increasing reference period leads to a decreasing number of values to determine an appropriate distribution function. A reliable determination of a suitable density function becomes more difficult. Depending on the extrapolation basis and the method for determining the distribution parameters, the significant wave height $H_{s50}$ is approximately 11.0 m.

Comparing this result with results of the peak over threshold method, deviations are found. Using the peak over threshold method, also assuming a Gumbel distribution, $H_{s50}$ is approximately 10.5 m. These deviations can be explained by the adaptation of the distribution. The distribution parameters $a$ and $u$ approximate the distribution function to the entire histogram. The shape of the distribution function is influenced also in areas of high quantiles by the cut off values below the threshold, cf. Figure 5. The use of truncated distributions should produce more realistic results. Corresponding investigations are still pending.

![Figure 5: Histogram of significant wave height $H > 3.0$ m and density function (Gumbel)](image)

### 5 CONCLUSIONS

Measurement series of several years are required for a realistic load calculation at OWT. For the first time a data basis for reliable extreme value predictions exists with the long measurement series created at the research platform FINO1. In this paper statistical methods for the determination and verification of extreme load parameters are presented. The extreme values of wind speed and significant wave height are very well approximated by a Gumbel distribution (extreme value distribution type I) for maximum values. Starting from different extreme value distributions, the distribution functions are extrapolated to annual extreme value distributions and load parameters determined with reference to [1, 2] and [3]. For the determination of load parameters extreme values with a long reference period should be used, because they approximate better the range of upper quantiles. However, a sufficient number of values is required for the reliable determination of the distribution function.
6 REFERENCES


VALIDATION OF A HIGH CYCLE ACCUMULATION MODEL VIA FE-SIMULATIONS OF A FULL-SCALE TEST ON A GRAVITY BASE FOUNDATION FOR OFFSHORE WIND TURBINES

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Keywords: Gravity Base Foundation, FE-Simulations, Settlements, High Cycle Accumulation Model.

Summary: Up to now, there is a lack of validated analysis procedures for the determination of accumulated deformations in the subsoil of foundations for offshore wind turbines. This contribution shows first results of such a validation of the High Cycle Accumulation model (HCA) [4],[5]. Basis of this validation are results from a full-scale test on a gravity base foundation subjected to high cyclic loading. The essential simplification of the subsoil, the material parameters and the input parameters for the FE-simulations are shown. The comparison of calculated and measured accumulated settlements of the foundation showed a quite good agreement. Furthermore it is worked out, that the initial relative density plays a significant role for the prediction of the accumulated deformations of the foundation.

1 INTRODUCTION

Foundations for offshore wind turbines are often designed as piles or gravity base foundations. These foundations have to ensure the compliance regarding maximum tilting of the wind turbine due to static and cyclic loading. Especially the cyclic loading resulting from wind and water waves may lead to increased tilting or progressive settlements within the structures life cycle. Up to now, there is a lack of validated and standardized analysis procedures for these accumulated deformations in the subsoil of offshore wind turbine foundations. Some procedures have been proposed recently, e.g. for monopiles in the EA Pfähle [1], but it is emphasized in [1], that they may not be seen as state of the art as none of them has been validated yet. Usually these procedures have been tested against small scale model tests only and are a priori limited to pile foundations.
In order to close the gap between small scale tests and real in situ foundations and to gain additional information about shallow foundations subjected to cyclic loading, the Ed. Züblin AG performed a full-scale test on a gravity base foundation, see Figure 1. Cyclic loadings, typical for an offshore wind turbine during storm events were applied. The foundation and the subsoil have been extensively instrumented so that a large amount of measured data on the cyclic soil – structure interaction was gained [2],[3].

Figure 1: Profile of the Test Foundation with the Loading Facility [3].

Meanwhile, a procedure for the proof of serviceability for offshore wind turbine foundations was developed at the Institute of Soil Mechanics and Rock Mechanics (IBF) of the Karlsruhe Institute of Technology (KIT). It is based on Finite Element calculations (FE) in combination with a High Cycle Accumulation model (HCA) [4],[5]. The procedure has so far been tested by back calculations of small scale model tests on monopiles [6] and on shallow foundations. However, these calculations are always influenced by drawbacks due to the model test scale, e.g. small stress level and very small deformations. A validation of the procedure with the results of a large scale test would give much more confidence. The comparison of results from the full-scale test with the prediction by the IBF procedure is the subject of this paper.

2 THE PROCEDURE OF THE IBF

The IBF procedure for the proof of serviceability for offshore wind turbine foundations is based on FE-calculations. It combines two soil models, one for the description of quasi static loading and one for high cycle loading. The quasi static loading phases are calculated using Hypoplasticity [7] with intergranular strain [8],[9]. These are the implicit steps in Figure 2, where stress-strain-relations are computed. The high cycle loading in the explicit steps is described by the HCA [4],[10], which allows the computation of accumulated strain and/or relaxation of stress due to a large number of cycles. Figure 2 shows also a basic assumption of HCA: the division of strain and stress paths resulting from cyclic loading in an oscillating part and a trend (accumulation). The HCA model describes only the trend with the oscillating part (strain amplitude) being an input parameter. The trend is described in analogy to a creep process, replacing the increments of time $\Delta t$ by increments of the number of cycles $\Delta N$ [11]. The main constitutive equation of the HCA links the rate of the effective stress $\dot{\sigma}$ with the strain rate $\dot{\varepsilon}$, using a barotropic elastic stiffness $E$:
\[ \dot{\varepsilon} = E : (\dot{\varepsilon} - \dot{\varepsilon}^{\text{acc}} - \dot{\varepsilon}^{\text{pl}}) \]  

\( \dot{\varepsilon}^{\text{acc}} \) describes the rate of strain accumulation. It is defined by the intensity of strain accumulation \( \dot{\varepsilon}^{\text{acc}} \) and the direction of accumulation \( \textbf{m} \). \( \dot{\varepsilon}^{\text{acc}} \) is a product of six functions \( f_i \), which stand for the different influences on the accumulated strain: the strain amplitude \( \varepsilon^{\text{ampl}} \) (function \( f_{\text{ampl}} \)), the number of cycles \( N \) (\( f_N \)), the void ratio \( e \) (\( f_e \)), the mean pressure \( p^{\text{av}} \) (\( f_p \)), the stress ratio \( \eta^{\text{av}} = q^{\text{av}} / p^{\text{av}} \) (\( f_Y \)) and polarization changes (\( f_{\pi} \)): 

\[ \dot{\varepsilon}^{\text{acc}} = \dot{\varepsilon}^{\text{acc}} \textbf{m} = f_{\text{ampl}} f_N f_e f_p f_Y f_{\pi} \textbf{m} \]  

Details of these functions and of the direction of accumulation can be found in [10] and [11], where also the determination of the material constants for HCA is described.

Figure 2: Procedure for the FE-Calculations with the High Cycle Accumulation Model after [10].

3 SUBSOIL OF THE TEST FOUNDATION

Based on the data from boreholes on the test field, the subsoil of the test foundation can be simplified by two main layers: an upper layer of fine quartz sand and a lower layer of medium-coarse quartz sand [12]. Especially the medium-coarse sand shows a certain scatter in the mean grain size \( d_{50} \) (see Figure 3 a)). This was captured by seven additional soil layers between -17 m and -31 m with mean grain size \( d_{50} \) and uniformity coefficient \( C_u = d_{60} / d_{10} \) gained from the borehole below plate C. The fine sand layer above (-7 m to -17 m) and the medium-coarse sand layer below (-31 m to -60 m) this region were represented by two mixed samples with mean values for \( d_{50} \) and \( C_u \) (fine sand (fS): \( d_{50} = 0.10 \) mm, \( C_u = 1.62 \); medium-coarse sand (mS): \( d_{50} = 0.36 \) mm, \( C_u = 2.38 \)). With these mixed sands, the whole set of parameters for the chosen soil models was determined in numerous static and cyclic laboratory tests [11], see Tables 1 and 2. The HCA material parameters for the additional soil layers between -17 m and -31 m were determined by a reduced laboratory program according to the procedure described in [13], using only one cyclic triaxial test per layer. The material parameters for these layers are shown in [14].

Table 1: Material parameters for Hypoplasticity

<table>
<thead>
<tr>
<th>Sample</th>
<th>( \varphi_c [°] )</th>
<th>( e_{d0} [-] )</th>
<th>( e_{c0} [-] )</th>
<th>( e_{i0} [-] )</th>
<th>( h_s [\text{kPa}] )</th>
<th>( n [-] )</th>
<th>( \alpha [-] )</th>
<th>( \beta [-] )</th>
</tr>
</thead>
<tbody>
<tr>
<td>fS</td>
<td>32,6</td>
<td>0,612</td>
<td>0,948</td>
<td>1,090</td>
<td>1,59E+08</td>
<td>0,185</td>
<td>0,135</td>
<td>1,35</td>
</tr>
<tr>
<td>mS</td>
<td>32,9</td>
<td>0,434</td>
<td>0,743</td>
<td>0,854</td>
<td>3,86E+08</td>
<td>0,135</td>
<td>0,185</td>
<td>1,95</td>
</tr>
</tbody>
</table>
Table 2: Material parameters for Intergranular Strain and HCA (all dimensionless)

<table>
<thead>
<tr>
<th>Sample</th>
<th>$R$</th>
<th>$\beta_R$</th>
<th>$\chi$</th>
<th>$m_R$</th>
<th>$m_T$</th>
<th>$C_{amp}$</th>
<th>$C_e$</th>
<th>$C_p$</th>
<th>$C_Y$</th>
<th>$C_{N1}$</th>
<th>$C_{N2}$</th>
<th>$C_{N3}$</th>
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</thead>
<tbody>
<tr>
<td>fS</td>
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<td>6</td>
<td>4.2</td>
<td>2.1</td>
<td>1.07</td>
<td>0.37</td>
<td>0.01</td>
<td>2.04</td>
<td>4.38·$10^{-4}$</td>
<td>0.103</td>
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</tr>
<tr>
<td>mS</td>
<td>$10^{-4}$</td>
<td>0.1</td>
<td>6</td>
<td>4.0</td>
<td>2.0</td>
<td>1.12</td>
<td>0.34</td>
<td>0.03</td>
<td>1.90</td>
<td>8.92·$10^{-4}$</td>
<td>0.247</td>
<td>0</td>
</tr>
</tbody>
</table>

Cone penetration tests (CPT) were conducted below every plate (A to D in Figure 1) of the foundation. The measured cone resistances are shown in Figure 3 b). A semi-empirical method [15] was used to gain the relative density ($I_D = (e_{\text{max}}-e) / (e_{\text{max}}-e_{\text{min}})$) distribution from the cone resistance. For FE-simulations, these single values have been averaged between the plates and for each considered soil layer (compare Figure 3 c)). The fine sand was simulated with a relative density of 68 %, while the medium-coarse sands had varying relative densities with the overall mean value of about 76 %.

Figure 3: a) Mean Grain Size in situ and for mixed Samples. b) Cone Resistance under every Plate of the Foundation. c) Relative Density derived from the Cone Resistance and Simplification for FE-Calculations.

4 CONSTRUCTION PHASES AND CYCLIC LOADING

The test foundation was constructed within an excavation pit, surrounded by sheet pile walls. The subgrade level was 7 m below the ground level, so that loose and partly cohesive upper soil layers were avoided. After the construction of the excavation pit, the four precast concrete plates of the foundation were lifted into the excavation pit and the concrete box girders were build. When the shaft was finished, the foundation was filled with sand and the excavation pit was also flooded [16]. All these construction phases were modeled step by step in the FE-calculations. Only the sheet pile wall was modeled as “whished in place”. The soil above the subgrade level was considered as a dead load and the excavation was created by a release of this load in the excavation pit area, confined by the sheet pile wall.
The cyclic loading on the test foundation was applied by a system of hydraulic cylinders and tension cables fixed on the shaft head. Within the test period of about six months, the foundation was loaded with about 1.5 million load cycles, divided into a calibration and test phase at the beginning and 18 subsequent load groups, each simulating at least one storm event [3]. As the main settlements and tilting of the foundation occur during the first load group, only these load cycles with the preceding cycles in the calibration and test phase were simulated in the FE-calculations. The cycles during the calibration and test phase have been grouped and reorganized to three packages in order to speed up the calculation. Each group of cycles needs at least four calculation steps (compare Figure 2) with about 100 increments each. This reorganization is shown in Figure 4 and based on the assumption, that Miner’s rule is approximately applicable to sand under cyclic loading [17].

Figure 4: a) In situ Loading till the first Storm and b) Reorganization of the Precycles.

5 FE-MESH AND OBSERVATION POINTS

Due to the foundations symmetry, only the half system is represented by the FE-mesh, see Figure 5. It is created by 87,029 solid brick elements (C3D8R in ABAQUS) with linear shape functions and reduced integration. 15,832 of these elements were contact elements, defining the Mohr Coulomb friction between the soil and the sheet pile wall, the temporary sand layer and the foundations plates.

Figure 5: FE-Mesh and Observation Points for the Comparison with Experimental Findings

For the comparison with the in situ measurements, some observation points were chosen in the mesh. They correspond to vertical extensometers, placed on the outside of the box girder
and anchored 30 m below subgrade level (V1 and V2 in Figure 5) and to vertical extensometers, placed 0.5 m below the plates A and C and anchored in different depth levels, see Figure 5. Furthermore, the tilting of the foundation and excess pore water pressures as well as soil pressures were compared to the experimental findings. Calculations were performed with and without consideration of the excess pore water pressure (PWP). Neglecting the excess pore water pressure is acceptable, since no accumulation of PWP appeared in the cyclic tests of the foundation [3]. Furthermore, exemplary simulations considering the PWP showed almost the same accumulated settlements as the drained simulations. This means, that the temporary change of PWP within every load cycle did not have a significant influence on the strain amplitude and thus on the intensity of strain accumulation $\varepsilon^{\text{acc}}$. Nevertheless, the comparison with the test results is based on simulations with PWP, since these simulations are the most suitable representation of the in situ conditions.

6 COMPARISON OF FE-SIMULATIONS WITH TEST RESULTS

Figure 6 a) compares the settlements during the construction phase measured at the vertical extensometer, fixed at 0.5 m below the subgrade level of plate A (Ext.-A05) with the calculated settlements. The models prediction is much too stiff, which can be explained by the well known overshooting of the intergranular strain after load reversals [9]. This is demonstrated in Figure 6 b), where the settlements due to FE-simulations without modeling the excavation (no load reversal) and without intergranular strain are compared to the regular simulation in Figure 6 a). They lead to much larger settlements, which are close to the measured ones. Since these simulations aim to validate the procedure for high cycle loading, this poor agreement with in situ data during the static loading phases was accepted for the time being.

![Figure 6: a) Settlements of the Test Foundation compared with FE Analysis for the Construction Phase. b) Impact of Intergranular Strain on the FE-Simulations.](image)

The comparison of the accumulated settlements in the calibration and test phase as well as during the first simulated storm event are shown in Figure 7. Here the values of the vertical extensometer on the box girder are shown (A V1/2). The vertical strokes of the calculated settlement curves correspond to the implicit steps and show the amplitude of the settlements,
while the HCA provides the trend of the settlements in the explicit parts (see Figure 7 b)). The prediction fits quite well to the measured data. The discrepancies in the development of the settlements in Figure 7 a) are related to the simplification of the loading, see Figure 4.

Figure 7: a) Settlements of Plate A in the Test and Calibration Phase and b) during the first Storm Event.

Not only a realistic estimation of accumulated settlements but also of the tilting of the structure is essential for the assessment of the foundations serviceability. This tilting was evaluated using the differential settlements of plate A and C, as shown in Figure 8 b). The agreement of the predicted tilting with the measured data for the storm event is not really satisfying yet. The differences might occur due to different relative densities below the single plates, which have not been taken into account up to now. In the simulations, the mean values shown in Figure 3 c) were considered homogeneous over the whole soil layer. Further investigations regarding this issue will follow in future.

Figure 8: a) Settlements of Plate C during the first Storm Event and b) Tilting of the Foundation due to Differential Settlements.

The void ratio has a very strong impact on $\varepsilon_{acc}$. Thus, one decisive parameter for the presented simulations is the assumed relative density $I_D$. Everyone involved in the estimation of $I_D$ from CPT-results knows, that there is always a remaining uncertainty, even with the usage of sophisticated procedures like the one described in [15] which was used in this
contribution. In order to estimate the influence of this input factor, simulations with different relative densities were conducted (see Figure 9). In order to speed up simulations, a coarser mesh was used for these calculations without PWD. It is obvious that even a variation of only 10 % of $I_D$ (e.g. $I_D,\text{mean}=0.68$; $I_D,+10\%=0.75$; $I_D,-10\%=0.61$) changes the results significantly. This has to be kept in mind for every simulation with a high cycle accumulation model.

Figure 9: Influence of a Variation of Relative Density on the Accumulated Settlements in the First Storm.

7 CONCLUSION

The good agreement between measured and calculated settlements can be regarded as the last step towards the validation of the HCA model, since it has already proven to be suitable for the simulation of small scale monopile tests [6]. The influences on the calculated tilting of the foundation are still under examination. The determination of the material parameters and the initial conditions for the FE-simulation are challenging, but worth the effort. The HCA model not only delivers settlements and tilting, it gives also a deep insight into the whole soil-structure interaction. For example, the redistribution of contact pressures under the plates due to cyclic loading [3] was very well reproduced by the FE-simulations. Furthermore, the procedure is very flexible with regard to any type of foundation (see e.g. simulations of monopile foundations in [18]) or boundary value problem.

8 REFERENCES


Numerical Modelling of Bubble Curtains

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Keywords: bubble curtain, sound mitigation, numerical modelling, wave equation, pile driving

Summary:
The pile driving associated to the construction of offshore wind turbines causes high sound pressure levels in the sea. This noise emission can be potentially dangerous for the marine life, especially mammals. In Germany the limit for underwater noise generated by pile driving activities is 160 dB of sound-exposure level at a distance of 750 m from the pile. To prevent emitted sound levels from exceeding this limit, noise mitigation systems must be used during the construction.

The bubble curtain is a commonly used system to reduce the acoustic pressure. Air is pumped through a tube along which a number of holes is distributed and rises due to buoyancy in form of bubbles. The bubble curtain can be placed directly next to the pile or at a certain distance.

The physical processes causing the sound attenuation effect of the bubble curtain are not clearly understood. This is partly due to the fact that direct measurements of the properties of the bubble curtain, such as the bubble size distribution, are very difficult to obtain because of the stochastic nature of the bubble flow and the uncertain boundary conditions in the sea [1]. In this paper a numerical method for modelling in detail a bubble curtain is presented. The propagation of a plane wave through a mixture of water and air bubbles is modelled with finite elements. Due to the unaffordable computational cost of modelling the complete bubble curtain, a statistically representative section of it is analysed. In this phase of the modelling the thermal and viscous effects at the bubble-water interface have not been considered.
1 INTRODUCTION

The offshore wind energy is an important part of the energy policy of the German Federal Government. During the construction of the foundation, piles with diameters of up to 7 m are rammed with high energy to the ground to ensure the stability of the tall structures of the wind turbines. Part of this energy is radiated from the pile as pressure waves in the sea and seabed.

The resulting noise is a major threat to the marine environment. In particular the harbour porpoise orients itself and hunts by hearing. The endangered specie is therefore directly affected by the ramming of the piles. A limit of 160 dB SEL is required at a distance of 750 m to protect the environment. To meet this limit value different types of sound mitigation systems are used during the construction.

A commonly used system is the bubble curtain. In it air is pumped through a tube lying on the ground. After leaving the tube through defined holes it rises due to buoyancy. The bubble curtain can be located next to the pile or at a certain distance.

The sound attenuation of a bubble curtain is well documented by measurements, but the several physical effects involved are still not clear. Some possibilities are: the reflection of the pressure wave at the bubble surface, also called geometrical scattering, the reflection at the interface between the water and the air-water mixture due to the change in compressibility and the scattering of the pressure wave due to the pulsation of the bubbles around their eigenfrequency [2].

To investigate the different effects, a detailed model of a bubble curtain with a gaussian bubble size distribution is presented. For a simple bubble curtain containing bubbles of only one size, a good agreement could be observed between the resulting eigenfrequency of the bubbles and the Minnaert frequency [3](Figure 1). In this paper the behavior of the bubble curtain is investigated under an impulsive excitation, which is similar to a pressure wave measured during the pile driving. In the next step these results are compared with an analytical model, which only considers the change of compressibility in the water-air mixture.

![Figure 1: a) Pressure inside a bubble excited by an incident harmonic pressure wave. b) Sound attenuation of a bubble. [3]](image)
2 Detailed Model of the Bubble Curtain

A section of the bubble curtain is considered to model the noise mitigation properties. Its size is chosen so that it displays the statistical properties of the bubble curtain: a so-called representative volume element. A volume with dimensions $30r \times 30r \times 80r$ is defined, being $r$ the medium bubble radius.

The bubble size distribution describes the amount of bubbles with a certain size occurring in the bubble curtain. It is assumed that the distribution matches a Gaussian distribution. The mean value and standard deviation are taken from [4]. The average bubble radius is around 4 mm and the standard deviation around 2 mm. In order to obtain a discrete distribution, the resulting continuous bubble spectrum is divided into nine classes with a class width of 1 mm. Due to the fact that there are no two identical bubbles in the real world, the radius of each bubble is multiplied by a random factor around one to reach a small deviation of the bubble size in each class. The actual number of 28 bubbles considered in the volume is determined by the air fraction of 1%. The bubbles are arranged in two plains perpendicular to the incident pressure wave, located at a distance of $\pm 5r$ from the middle plain of the representative volume.

To obtain the pressure field around the bubble curtain, the wave equation in time domain or the Helmholtz equation in frequency domain are solved. The sound attenuation is determined by comparing the incident pressure with the averaged pressure on the backside of the bubble curtain. The boundary conditions are chosen as follows: the plane wave incides the domain on the left side. The boundary conditions on the front, back, top and bottom are periodic. This condition reproduces a periodically continued bubble curtain. Plane wave radiation conditions are defined on the right side of the domain.

3 Results

On a first step the behavior of the bubble curtain is investigated under an incident impulsive pressure wave. The duration of the pressure wave is 1 ms: similar to the ones observed during pile driving. The pressure amplitude has been set to 10000 Pa. Figure 2 shows two different slices of the pressure field in the representative volume element during a time interval between 0.08 ms and 0.36 ms. The pressure wave enters the domain on the left side and strikes the bubble curtain at around $t = 0.12$ ms. A part of the wave with a small amplitude (around 700 Pa) crosses the bubble curtain and leaves the domain on the right side. A bigger part of the wave energy is scattered backwards and leaves the domain on the left side. It can be observed that the pressure inside the bubbles differs from that in their surroundings. It is changing periodically over the time.
Figure 2: Pressure field around the bubble curtain during the time interval 0.08ms-0.36ms.

Figure 3 displays the pressure inside three different bubbles with radii 0.4 mm, 1.4 mm and 4.4 mm. The observed time interval consists of two different parts: first (0.0 ms – 1.0 ms) the impulse incides the observed domain and strikes the bubble curtain, then (1.0 ms - 2.5 ms) there is no incident pressure field. During the first part the averaged pressure in the bubbles rises up to 1500 Pa. After the impulse is gone, the pressure in the bubble keeps oscillating around 1500 Pa.

The pressure in the bubbles behaves differently depending on the size of the bubble. It can be observed that the pressure inside the smallest bubble \((r = 0.4 \text{ mm})\) oscillates with a high frequency. The biggest bubble \((r = 4.4 \text{ mm})\) is only compressed during the impulse and starts to oscillate afterwards at low frequency.

The pressure in each bubble oscillates with the respective eigenfrequency of the bubble. The eigenfrequency of each bubble shows a good agreement with the Minnaert frequency, which takes into account the size of the bubble and the ambient pressure [5].
In the next step the bubble curtain is analyzed in the frequency domain and compared with an analytical model based on the *Woods* equation, which only considers the change of compressibility in water due to air [2].

Figure 4 shows the resulting transmission loss of both models for an air fraction of 1% between 63 Hz and 6300 Hz in third octave bands. Especially in the lower frequency range (up to 400 Hz) a good agreement between both models can be observed. For higher frequencies the results diverge strongly. This can be explained by the strong sound attenuation effect related to the resonance of the bubbles. The grey box marks the frequency range of the bubble eigenfrequencies. Two measurements from FINO 3 conducted at distances of 245 m and 910 m are plotted to validate the trend of both models [6]. The trend of the numerical model shows a qualitatively good agreement with the measurements despite the unknown air fraction and bubble size distribution during the measurements.
4 CONCLUSION
In this paper a method based on the numerical solution of the wave equation has been used to model the behavior of a bubble curtain under an impulsive excitation. It has been shown that the bubbles in the bubble curtain oscillate in their eigenfrequencies under an impulsive pressure wave. The resulting sound attenuation in frequency domain has been compared with an analytical model, which only considers the change of compressibility in the water due to the air, and measurements conducted in different distances to the pile. A good agreement between the measurements and the resulting trend of the numerical model could be seen. This leads to the conclusion that the oscillation of the bubbles has to be considered, when determining the sound attenuation effect of a bubble curtain.

5 ACKNOWLEDGEMENTS
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6 REFERENCES


IBRATION BASED STRUCTURAL HEALTH MONITORING (SHM)
FOR OFFSHORE WIND TURBINES – FROM MONITORING THE
BLADES TO A CONCEPT FOR FOUNDATION STRUCTURES
IWEC 2014

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Keywords: Offshore Support Structures, Rotor Blades, Structural Health Monitoring (SHM), Condition Monitoring System (CMS)

Summary: After numerous spectacular cases of damage on rotor blades between 2003 and 2005, Wölfel conducted two R&D projects and established the basis for an SHM system which is now available as a product and for UnderwaterINSPECT, another R&D project, for which a large-scale test will be performed in the foundation test pit of the Leibniz University in Hanover.

In the R&D projects CMS and SHM Wind, the basics of damage detection and assessment were developed by means of measurement-based analysis of vibrations. For this it was necessary to develop a suitable fully automatic method for the monitoring of wind turbine rotor blades. This method should ensure a clear distinction between the structural response due to damage and the normal vibration behavior and it should be robust enough to work reliably under rough environmental conditions. This was achieved with a combination of several methods of vibration analysis (e.g. operational modal analysis) and statistical pattern recognition. The latter is mainly used to compensate for the dynamic effects of changes in environmental and operating conditions. In addition to the method, highly sensitive sensors were developed with a very good signal-to-noise ratio.

Today, Wölfel offers systems for ice detection and rotor blade monitoring. However, the development of the method will be continued and is planned to result in a system for the monitoring of foundation structures. For this purpose, a large-scale test on a scale of 1:10 is planned in the foundation test pit. In this test, an offshore wind turbine on a monopile foundation will be provided with an SHM system, which measures, among other things, accelerations, strains, temperatures and the structure inclination. The signal analysis is based on the further development of the already presented SHM systems; testing of further combinations of methods is planned. In the "learning phase", operating loads and extreme loads will be applied to the monopile and dynamic measurements will be made. Then the behavior will be measured after "benign" changes due to changes in environmental and operating influences and after the "malign" occurrence of damage.
The findings from the large-scale test will be the basis for an SHM system for the monitoring of foundation structures of offshore wind turbines.

1 INTRODUCTION

The R&D project CMS was preceded by several spectacular cases of rotor blade damage. Not only individual blades were lost, several manufacturers had to continuously check the blades of numerous turbines individually in the field by visual inspection every few weeks. In some cases, hundreds of blades had to be replaced as a precaution at the expense of the blade manufacturer. This led to the idea to transfer the method of structural health monitoring (SHM), which had been used primarily for bridges and in the aerospace industry so far, to rotor blades. This was done in the project "Innovative Condition Monitoring System for sustainable monitoring of safety-related components, CMS" with a "two-eye" monitoring system, in which low-frequency structural vibrations were used for global monitoring and for statements on the load bearing capacity of the rotor blade (first eye). The second eye was based on acoustic emission, which was used for damage localization [1, 2].

After the successful completion of CMS, the SHM system was enhanced in a follow-up project "Model-based Structural Health Monitoring for rotor blades of wind turbines", SHM Wind, and successfully tested for the first time on rotor blade test stands and on a wind turbine. With two identical blades, a more or less identical characteristic damage caused by overload could be generated in the same place during the test. The global eye of the SHM system could properly identify this damage and the previous degradation of the blade. This did not only demonstrate the functionality of the system, it also provided important evidence of the reproducibility of the measurement results [3].

These test results were a good reason to continue with an industrial development after completion of the project. Meanwhile, two SHM systems are available as products, SHM.Blade® and IDD.Blade®. However, the test results also provided the impetus for further development of the method with regard to issues like stability of offshore foundation structures or grout monitoring.

2 BASIC ASPECTS OF DAMAGE DETECTION

The basic equations of a linear dynamic system already show that changes in stiffness and mass have an effect on the system’s vibration behavior. This is the basic principle of vibration-based damage identification approaches. The development of methods for vibration analysis with regard to damage detection was strongly driven by the rapid development of measuring technology with extremely sensitive low-noise sensors, electronics, communication and information technologies. Regardless of whether the different approaches concerning SHM described in literature are model-based or only based on signal analysis, almost all approaches are still based on the assumption of a linear-elastic system that is excited by known or unknown stochastic forces which are normally distributed and stationary.

These assumptions were sufficient to detect, for example, artificially introduced damage under laboratory conditions. However, the application of these methods in practice quickly showed that the simplified theoretical assumptions were inadequate to describe real systems and thus they are not suitable for the rough practice.
Especially for the examination of a complex dynamic system in situ, like a wind turbine, these simplified assumptions are no longer sufficient. The assumed stationary system is superimposed by transient, stochastic and periodic excitations of the turbine. Due to variable boundary conditions, temperature fluctuations, variations in humidity etc., the dynamic system of a wind turbine is highly non-linear (see field “Measurements” in Figure 1). Thus, the compensation of the influence of environmental and operating conditions, the EOCs, on the dynamic behavior of a wind turbine is of particular importance.

By considering the context described above, SHM.Blade® and IDD.Blade® recognize structural changes due to loss of stiffness (damage) or mass change (ice). The detection is based on a reference state, which is assumed to represent the unchanged - that means undamaged - ice-free blade. If, at a later stage, the measured signals indicating a change in relation to the reference state, this is a sign of one of the two possible changes, namely ice or damage (see field "Pattern Recognition" in Figure 1). To evaluate the measurement data, the damage-sensitive features extracted from the signals are used. The extraction of features uses a signal analysis method in the time domain: operational modal analysis, OMA, based on stochastic subspace identification. With this method it is possible to transfer the measured time histories in a state-space model and to determine the eigenfrequencies, modal damping and vibration modes as features of the system. After the feature extraction, the actual changes with regard to ice or structural damage have to be separated from the numerous effects caused by the EOCs, because e.g. a change in the pitch angle can have similar results regarding the natural frequencies as a structural damage. The compensation of EOC effects requires a thorough structural dynamic understanding of the object that is monitored.

If a change, e.g. due to damage, has been detected, it is interpreted on a statistical basis whether the change is significant. The significance of the change together with the knowledge about the feature thresholds lead to a decision (see field "Decision" in Figure 1) regarding the detection of a relevant structural damage.

The Wölfel systems for rotor blade monitoring work fully automatically and independently according to the flow chart described in Figure 1. The systems are able to compensate production-related variations in mass and stiffness of the individual rotor blades, as each blade automatically determines its references by means of a self-learning process. Further details about the theoretical principles and practice-relevant aspects of the Wölfel systems are mentioned in [7], [8] and [9]. This capability will be of special importance in connection with the monitoring of foundation structures described later, because every foundation structure is an individual in terms of its embedding in the sediment!
3 EXAMPLES FOR DAMAGE AND ICE DETECTION WITH SHM.Blade®

In order to automatically extract the requested damage-sensitive features from measured time histories, first so-called stability plots are used in the context of the mentioned operational modal analysis. They are often used in the hope to separate mathematical poles from physical system poles. For this purpose, the dynamic characteristics (features) for different state-space model orders are entered in a diagram; the features that vary only “slightly” from one model order to the other, are regarded as "stable". Such a stability plot, which is based on the measurement data of a rotor blade, is shown in Figure 2. The black circles show that the eigenfrequencies, modal dampings and mode shapes have not changed significantly from one model order to the other. The solid curve represents an average power spectral density (PSD) from the signals. This shows that the stable poles are close to the peaks in the PSD. However, PSD is not relevant for the feature extraction. The stability plot is not very suitable for an automatic method without engineering intervention in the selection of characteristics, though. For this purpose, further algorithms have to be used.
In the following it is explained how dynamic structural properties can be used in the context of damage and ice detection of rotor blades.

a. Rotor blade damage detection

This example is based on long-term vibration measurements at a rotor blade of the 40 m class. Simultaneously sampled accelerations were measured with 12 measuring channels. The excitation was generated stochastically by a shaker, but the excitation force was not used for data evaluation. Originally, the rotor blade was intact; deliberately induced fatigue load caused a crack of a few decimeters at the trailing edge. In this case, the temperature had a significant influence on the blade dynamics. Without temperature compensation, damage detection would not have been possible, because the temperature influences had masked the damage effect. A damage indicator which was automatically extracted from the experimental measurement data and an associated threshold value is shown in Figure 3. The process of damage development is accurately reflected by the indicator.
b. Online rotor blade ice detection

The following example describes ice detection on the rotor blade of a wind turbine with the fully automatic and certified system IDD.Blade®. Ice detection is performed in almost any operating condition of a wind turbine. These operating conditions include: 1) turbine is idle with the blades out of the wind, 2) turbine is waiting for wind, 3) turbine is rotating at the nominal rotational speed, 4) turbine is rotating below nominal speed and above switch-on speed. The varying environmental and operating conditions affecting the blade dynamics are generally compensated. It also has to be considered that due to the variable operating conditions it is not always possible to identify all features at the same time. However, in order to solve this problem, appropriate statistical pattern recognition methods are uses. Another necessity that arises in the case of the rotating blade is a separate interpretation of the structure eigenfrequencies at locations where these frequencies cross higher harmonic excitation components from the tower-blade passage.

Figure 4 shows the change of the eigenfrequencies from an automatic monitoring process over a period of about 5 months. During this time, the EOCs of the turbine have constantly changed; in certain periods, there was ice on the blades. Using identified frequencies as an example, Figure 4 illustrates that a meaningful analysis is not possible without taking into account the EOCs.
4 ENHANCEMENT OF THE SHM SYSTEM FOR THE MONITORING OF OFFSHORE FOUNDATION STRUCTURES

The aforementioned SHM system will be further enhanced for the monitoring of offshore foundation structures. This primarily includes research on the methodical approaches. And here we have come full circle regarding this lecture and the occasion of this lecture, because the required tests will be carried out in cooperation with FhG IWES in the foundation test pit at Hanover University, which is inaugurated today.

The prerequisite for the methodical development is measurement data of the expected changes or damage. This data is to be obtained in a large-scale test on a scale of 1:10. At the same time, an FEM model of the test will be developed. With this model, the test will be recalculated. By comparison of the measuring results and the test results it is possible to tune the FEM model in such a way that it accurately simulates the test (model update). In view of the highly non-linear simulation object (material data of the sediment, water saturation, influences from dynamics, etc.), this is quite a challenge. A verified simulation model, however, is suitable for a prognosis. This allows to simulate measurement data of changes of the structure and their boundary conditions and especially the EOCs, which cannot be realized in the test pit.

For the test, the monopile will be provided with accelerometers, strain gauges or FBG, tilt sensors and sensors to record relative displacements. Operating loads from the turbine will be introduced with an electrodynamic shaker and extreme loads will be produced by the hydraulics of the test pit. Some structural changes and changes of the boundary conditions are artificially introduced. Thus, experiments with different scourings will be performed (see Figure 6 showing the simulation of a scouring). Other changes will be caused by extreme loads, for example. This includes a degradation of the sediment.
5 CONCLUSION

Wind turbines are highly non-linear dynamic systems, which cannot be properly simulated by linear elastic models. In addition, changes in environmental and operating conditions lead to similar displacements of the modal parameters as damage or ice. SHM systems which are based on the analysis of vibration data have to allow for this. For structural health monitoring and ice detection this was successfully realized. In another project, the presented SHM system is planned to be further developed for the monitoring of offshore foundation structures. The required data for this is to be provided by parallel tests and simulations.

6 REFERENCES

A RUGGED STRAIN MEASUREMENT METHOD FOR ROTOR BLADES BASED ON TEM WAVEGUIDES

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Keywords: Strain Measurement, TEM Waveguide, Plasma Treatment, Frequency Domain Reflectometry (FDR), Time Domain Reflectometry (TDR)

Summary: Rotor blades for wind turbines are made of glass-fibre reinforced compound material. Their size in off-shore wind farms exceeds 50m in length and 15t in weight. They have to be designed to withstand wind and weather over approximately 20 years of lifetime. The ability to monitor the mechanical stress is crucial in order to reduce maintenance costs and to maximize operational availability.

This work is part of the AGIP founded research project High-Performance Fibre-Composite Rotor Blades for Wind Turbines through Plasma-Treatment [1]. The aim of this subproject is to develop alternative strain measurement methods for online monitoring of mechanical stress imposed upon rotor blades. Strain sensors have to be compatible with the glass-fibre compound and its production process. This goal was reached by embedding a passive TEM waveguide into the rotor blade structure. An advantage is the low cost and the superior ruggedness of e.g. a twisted pair copper cable or a stripline.

1 STRESS TESTS

During development and production rotor blades have to undergo severe testing in order to withstand mechanical and electrical stress over their lifetime. Figure 1, Figure 2 and Figure 3 depict static, dynamic and lightning tests respectively.

Figure 1 Static stress test, source: LM Wind Power
Obviously these tests are important to understand and verify stress and lifetime prediction. But they cannot replace in-situ measurements covering the entire lifetime of a rotor blade. Such measurements allows for both verification of design assumptions and an online monitoring of the effective mechanical stress as a function of location, weather and age. The online monitoring can be used to prediction the residual lifetime, cutting maintenance costs and maximizing operational availability.

2 COMPARISON OF STRAIN MEASUREMENT METHODS

In order to measure the mechanical stress imposed upon a rotor blade during its entire lifetime an appropriate sensor has to be chosen. The sensor has to be rugged, simple and should enable integral strain measurements over the entire length of the rotor blade.

Table 1 shows commonly used sensors. They all rely on a strain-imposed elongation of the material. In contrast to strain gauge and fibre Bragg grating which allow only for local measurements a laser interferometer can detect an overall change in length. But a laser interferometer is neither rugged nor simple [2].

As an alternative the use of a simple two-wire TEM waveguide is proposed. It can be inserted into the glass fibre compound during manufacturing and enables a simple measurement of the overall elongation as will be discussed in the following chapters.
Three possible measurement setups using a TEM waveguide are depicted in Figure 4. The physical quantity altered by elongation is either a change in resistance, which can be obtained via a DC measurement, or the change of the reflection/transmission property, which can be obtained both in time domain (TD) or in frequency domain (FD).

Table 1 Comparison of strain measurement methods

<table>
<thead>
<tr>
<th>Method</th>
<th>Strain gauge</th>
<th>Laser interferometer</th>
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<tr>
<td>Fiber Bragg grating</td>
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<tr>
<td>TEM waveguide</td>
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Three possible measurement setups using a TEM waveguide are depicted in Figure 4. The physical quantity altered by elongation is either a change in resistance, which can be obtained via a DC measurement, or the change of the reflection/transmission property, which can be obtained both in time domain (TD) or in frequency domain (FD).

Figure 4 Using a TEM waveguide as alternative strain sensor utilising a) change in resistance, b) change in reflection and c) change in transmission due to elongation
The most simple setup, Figure 4b), utilises the reflection of a propagating wave. The length $\ell$ of the waveguide is chosen to be the length of the rotor blade. Using several windings a multiple of the rotor blade length can also be realised. The characteristic impedance $Z_W$ of a waveguide is geometry and material dependent. A reflection is enforced by a false termination $Z_E$ at the far end of the waveguide $Z_E \neq Z_W$. In order to maximise the reflection a termination $Z_E = 0$ or $Z_E \to \infty$ is best. Within the scope of this paper $Z_E = 0$ and frequency domain reflection (FDR) measurements are chosen, because it is the easiest to manufacture and the most accurate to measure.

3 Simulation

Analytical and numerical results will be presented first, in order to explain the frequency domain measurement procedure. Figure 5 shows a TEM waveguide with a matched generator impedance $Z_i = Z_W$ and a mismatched termination $Z_E \neq Z_W$. The incident and the reflected harmonic waves are depicted. Both waves superimpose such that a standing wave occurs. Assuming a lossless waveguide with the per unit length inductance $L'$ and capacitance $C'$ the wavelength $\lambda = \frac{\nu}{f} = \frac{1}{f \sqrt{L' \cdot C'}}$, the characteristic impedance $Z_W = \sqrt{\frac{L'}{C'}}$ and the propagation constant $\gamma = \sqrt{L' \cdot C'}$ can be calculated.

The characteristic impedance of a twisted pair cable usually measures $Z_W = 100\Omega$. Assuming for example the length $\ell = 10m$ and feeding the waveguide with an amplitude of $U_Q = 1V$ the voltage $U_A$ can be calculated. Using $Z_i = 10\Omega$, $50\Omega$, $100\Omega$ the resulting voltage $U_A$ is depicted in Figure 6. The steepest curves belong to $Z_i = 10\Omega$. Adding mechanical strain will change the overall length of the waveguide. This will in turn change the frequency at which both waves extinguish each other at the near end of the waveguide.
Using an elongation in-between 0% and 1% this effect is shown in Figure 7. Measuring two voltages $U_{A1}$ and $U_{A2}$ at slightly different frequencies $f_1$ and $f_2$ with $\Delta f = f_1 - f_2$ the voltage change $\Delta U = U_{A1} - U_{A2}$ can be calculated. The arrows mark the different voltages.

It can be shown, that the voltage change is proportional to the elongation $\Delta U \propto \Delta x$. This relationship is shown in Figure 8 using $\Delta f = 50, 100, 200, 500, 1000$ kHz. Obviously $\Delta U \propto \Delta f$ also holds. Within limits the sensitivity of the strain sensor can be adjusted by changing $\Delta f$. 

---

**Figure 6** Standing wave $\ell = 10 \text{m}, Z_w = 100 \Omega$ and $Z_i = 10 \Omega, 50 \Omega, 100 \Omega$

**Figure 7** Standing wave $\ell = 10 \text{m} + \Delta x, Z_w = 100 \Omega, Z_i = 10 \Omega$ and $\Delta x = 0, 20, \ldots, 100 \text{mm}$

**Figure 8** Voltage change $\Delta U$ as a function of $\Delta x$ and $\Delta f$
4 Measurements

In order to verify the simulated data a TEM waveguide specimen with $\ell = 1.5\text{m}$ and $Z_W = 50\Omega$ is embedded into a glass fibre compound as a stripline. These specimens are mounted in a tensile strain machine and stressed such that a maximum elongation of 0.2% ($\Delta x_{\text{max}} = 3\text{mm}$) occurs. Because of the mechanical properties of the specimen a force of up to $F_{\text{max}} = 1\text{kN}$ is necessary. A spectrum analyser equipped with a tracking generator is used both as generator and as detector. Figure 9 and Figure 10 show the measurement setup.

In contrast to the calculations presented in chapter 3 the embedded stripline is lossy. Therefore the simulation has been rerun varying the per unit resistance of the copper track in-between $R' = 1\ \text{m}\Omega \ldots 5\ \text{m}\Omega$.

A comparison of simulation and measured data is shown in Figure 11. Both curves match well using the per unit resistance $R' = 5\ \text{m}\Omega$.

The measurements in Figure 11b) are performed varying the force within $0 < F < 1\text{kN}$.
The voltages $U_{A1}$ and $U_{A2}$ have been taken with $\Delta f = 4 \text{MHz}$ and $\Delta U = U_{A1} - U_{A2}$ has been calculated. Both, the force that was necessary in order to stress the specimen and the resulting readout as a function of the elongation are depicted in Figure 12. For comparison the simulated function is included in Figure 12b) proving the proposed principle.

**Figure 11** Comparison of $U_A(f)$ a) simulation with $R' = 1 \frac{\text{m}^2}{\text{m}} \ldots 5 \frac{\text{m}}{\text{m}}$ b) measurement

**Figure 12** a) measured force $F(\Delta x)$ b) comparison of $\Delta U(\Delta x)$ measured and simulated

5 CONCLUSION

This paper presents a rugged strain measurement method based on TEM waveguides. It has been analysed in detail and the predicted performance has been proven using a small-scaled specimen. The principle is compatible with large glass fibre compound structures such as rotor blades.

6 REFERENCES

AIRBORNE SOUND BASED DAMAGE DETECTION FOR WIND TURBINE ROTOR BLADES USING IMPULSE DETECTION IN FREQUENCY BANDS
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Keywords: Acoustic emission, damage detection, wind turbine rotor blade, airborne sound

Summary: This paper presents a cracking sound detection algorithm for damage detection of wind turbine rotor blades.

1 ABSTRACT
Composite rotor blades of wind turbines are subjected to high static and dynamic loads. The load can cause small damages which can accumulate over time to critical structural damage. A system detecting defects in early stages helps to react fast and avoid critical damage. Such a method will enable the wind turbine operator to increase safety and minimize the economical burdens caused by downtime, repairing, replacing and maintenance. Therefore many research projects try to pave the way to a reliable and practical early damage detection system for wind turbine rotor blades.

One promising approach is acoustic emission event detection. Acoustic emission stands for stress waves emitted by a damage process. While other acoustic emission approaches use ultrasonic surface acceleration as input signals, we propose using the airborne sound in audible frequencies. The aim is to detect cracking sounds emitted by the damage process. In this paper we present an improved version of our algorithm for detecting these cracking sounds by using impulse detection in individual frequency bands. The performance with the new algorithm increased significantly. In the recordings of a 76 day full scale rotor blade fatigue test we detected 104 cracking sounds while there were only six false positive detections. Compared with the previous algorithm, the amount of detected cracking sounds is about three times higher and there are significantly less false alarms.

2 INTRODUCTION
Wind turbine rotor blades have to match the criteria of being light weight and large at the same time. Therefore all modern blades are made of composite materials. During operation the blades are subjected to high static and dynamic loads. This permanent load can lead to small damages which accumulate over time to critical structural damage. In situations where damage occurs in an important part of a structure, structural health monitoring systems can be able to improve safety and minimize the costs for maintenance, repairing and replacing. Therefore the damage has to be detected in early stages to react fast and avoid critical damage. In modern wind turbines controlling units are already implemented like the blade pitch control or an emergency shut-off system. These systems might be also triggered by a structural health monitoring system to avoid critical damage.

There are two certified rotor blade monitoring systems on the market [1] which are able to detect ice, but these systems are not capable of detecting damage in early stages. Many
research results of nondestructive testing methods were published so far. Several approaches can be used for automatic damage detection of wind turbine rotor blades. An overview can be found in [2] [3]. For detecting damage automatically, reliably and in early stages many research projects focus on the acoustic emission event detection approach. The aim of this approach is to detect components of the stress wave caused by the damage process. For this sensors mounted on the surface of the blade are used. With this approach small damages can be detected [2] [3] [4] [5]. The sensors operate in ultrasonic frequencies, therefore the amount of sensors is relatively high due to the size of modern blades and high internal damping of composite materials [4] [5].

Using acoustic emission for damage detection in an operating wind turbine is an unsolved problem. The higher risk of damage from lightning strikes caused by the electrical conductive wires is the main problem which prevents testing an acoustic emission approach in an operating rotor blade. So far there are only few results published using an acoustic emission system in an operating wind turbine rotor blade. The ability of detecting damage while operation could not directly been tested due to the lack of damage. Environmental noise during operation was observed which has to be taken into account to avoid false detections [6] [7].

In contrast to existing approaches, in [9] we proposed a damage detection approach by identifying cracking sounds in audible frequencies from 20 Hz to 20 kHz using the airborne sound. The idea is based on the observation of audible cracking sounds during rotor blade tests [8]. We assume that only up to three sensors are necessary for monitoring the whole rotor blade due to the relatively low attenuation of the air in such frequencies. In addition, fiber optic microphones can be used for recording sound inside a rotor blade. Their cords are optical fibers which are non-metal, so they do not increase the risk of damage from lightning strikes. This makes the approach applicable in an operating rotor blade.

3 Cracking Sound Model

In [9] we presented the first results of using airborne sound for rotor blade damage detection. In that paper a model of impulse-like cracking sounds is described. The model was developed using airborne sound recordings during parts of a certification test of a 55 m rotor blade. The recordings consisted of one 76 day flapwise fatigue test and four static tests which took about ten minutes each. The sound was recorded inside the rotor blade using three electrical microphones and one fiber optic microphone.

With this recordings a model of the cracking sound was developed. The model is further described by an example cracking sound in the static test recordings with low environmental noise. A part of the recording is displayed using the power spectrogram (Figure 1). The cracking sound is displayed in the spectrogram as triangular shaped area. The power in very low frequencies is noise from the rotor blade test.

The first characteristic of the cracking sound model is the impulse-like increase in power which can be found at 0.15 s in Figure 1. The maximal power occurs in this time period which we refer to as first part of the impulse. Subsequently there is an approximately exponential decrease in power over time. In Figure 1 the power is displayed in decibels, therefore the decrease is approximately linear. The power of every impulse is distributed over a wide frequency range, so the impulses are non-tonal. Depending on the impulse, the frequency where maximum power occurs can vary in a wide range. In our recordings the value occurs in
the range of 150 Hz to 10 kHz. The example impulse has its maximal power at 500 Hz. Another characteristic of all cracking sounds is a specific decrease in power towards higher frequencies, beginning with the frequency where maximal power occurs. The decrease is linear in decibel with a specific constant slope.

4 Detection Algorithm

In [9] we also presented a cracking sound detection algorithm which is based on the comparison of the input signal with the cracking sound model. In this paper an improved detection algorithm is presented, including an impulse detection feature based on individual frequency bands and a procedure for the extraction of frequency boundaries. The detection algorithm uses five audio features \( f_1 \)-\( f_5 \) to represent the model characteristics. The features are based on the power spectrum \( P(k,l) \) calculated by a windowed short time Fourier transform. Here \( k \) is the frequency index and \( l \) the time index. All features are compared with threshold parameters \( \delta \) to see if the signal is similar to the cracking sound model. The principle flow chart of the algorithm is displayed in Figure 2. The power gradient feature and the bandwidth at which the signal is impulse-like are responsible for the improved performance of the current algorithm. In the following Subsections all important steps of the detection algorithm are described.

Figure 1: Power spectrogram with cracking sound and minor noise of the rotor blade test.

Figure 2: Flow chart of the detection algorithm.
a. Power gradient feature

In the previous algorithm, the first feature was the increase in power over all frequencies which displayed if the signal is impulse-like. This led to missed impulse-like sounds when at the same time an impulse and a noise signal with high decrease in power occurred. So we modified this feature. The power gradient feature used now analyzes if the signal is impulse-like over a wide frequency range. For this, the increase in power in all frequency bands is calculated by Equation 1. The raising time of the impulse model is denoted by $l$. If the increase in power is greater than the threshold $g$, the frequency band is defined as impulse-like. The power gradient feature $f_1$ is the number of all impulse-like frequency bands as shown in Equation 2. Very low frequencies are left out. These frequencies can provide high power and are not correlated with the cracking sound. The frequency range reduction is adjusted with the parameter $k$.

\[
g(k,l) = \begin{cases} 1, & \text{if } P(k,l) - P(k,l-l) > g_b \\ 0, & \text{else} \end{cases}
\]

(1)

\[
f_1(l) = \sum_{k=k_s}^{k_w} g(k,l)
\]

(2)

If a significant number of bands shows a sudden power increase the feature exceeds the threshold parameter $\delta_{\text{min}}$ and the signal becomes a candidate for a cracking sound with the time index $l_c$ and is further processed.

The frequency bands where the signal is impulse-like is extracted and used for the features processed after the power gradient feature. Frequency components which are out of the impulse-like frequency bandwidth are not taken into account. This leads to a better separation of noise when checking the three features tonality, spectral slope and spectral similarity. The bandwidth is approximated by combining all bands where the power increases, if the frequency gap between the bands is smaller than a defined boundary. If the amount of combined impulse-like frequency bands is greater than one, the band $k_w$ with the widest bandwidth is taken. The boundary frequencies indices $k_b$ and $k_e$ are used for further processing.

\[
k_b \leq k_w \leq k_e
\]

(3)

b. Noise reduction

In case a power gradient according to the model is observed, the algorithm reduces the influence of not impulse-like signals in the power spectrogram by using spectral subtraction. The assumption for spectral subtraction is a stationary signal. This assumption is approximately valid if a short time period is taken. The signal model specifies that the rise time of the power is shorter than the decay time. Therefore the subtracting spectrum is taken a short time period before the beginning of the impulse, at the time $l_c - l_r$. The noise reduced spectrogram $P_s$ is only used for calculating features taken from the first part of the impulse.

\[
P_s(k,l_c) = 10 \log_{10} \left[ P(k,l_c) - P(k,l_c-l_r) \right]
\]

(4)
c. Tonality feature
There are three features calculated using the noise reduced power spectrum of the first part of the impulse. The first feature represents the tonality of the signal. For this the spectral flatness measure \[10\] is used. The geometric mean of the power spectrum is divided by the arithmetic mean as shown in Equation 5.

\[
f_2(l_c) = \frac{\prod_{k=k_b}^{k_p} P_s(k, l_c)^{-1}}{\sum_{k=k_b}^{k_e} P_s(k, l_c)}
\]

The result is a value between zero and one. Where zero represents a maximal tonal signal and one a maximal noise-like signal. The cracking sound model specifies the sound as non-tonal so the feature value should not be lower than the threshold \(\delta_{2,\text{min}}\).

d. Spectral slope feature
The second feature, using the noise reduced power spectrum of the first part of the impulse, is the specific slope in the power spectrum. This feature represents the linear decrease in decibel towards higher frequencies, beginning at the frequency \(k_p\) where maximal power occurs. This frequency has to be taken from the interval \(k_w\). The gradient \(f_3\) of a simple linear regression is calculated as an approximation of the signal slope by

\[
f_3(l_c) = \frac{\sum_{k=k_p}^{k_e} (k - \bar{k})(P_s(k, l_c) - P_s(l_c))}{\sum_{k=k_p}^{k_e} (k - \bar{k})^2}.
\]

The overbar in Equation 6 means the arithmetic mean using the frequency interval \(k_p\) to \(k_e\). The upper and lower threshold parameter \(\delta_{3,\text{max}}\) and \(\delta_{3,\text{min}}\) define the allowable deviation from the characteristic model slope. In Figure 3 the power spectrum of the first part of the cracking sound is displayed. The frequencies \(f_p\) and \(f_c\) are corresponding to the frequency indices \(k_p\) and \(k_e\).

e. Spectral similarity feature
The third feature, calculated using the noise reduced power spectrum of the first part of the impulse, compares the impulse with the model curve. This feature represents the specific linear decrease in decibel from the frequency with maximal power \(k_p\) towards higher frequencies as well as the noise-like characteristic of the impulse.

The power spectrum of the impulse model curve \(P_{prot}\) is defined by a characteristic exponential decline of power towards higher frequencies. The representation of the signal in decibel linearises this decline.
The feature $f_4$ measures the similarity of the signal spectrum and the model spectrum. The similarity is displayed in the euclidean distance normalized by the bandwidth of the spectrum by

$$f_4(l_c) = \frac{1}{k_e-k_p} \sqrt{\sum_{k=k_p}^{k_e} [P_{prot}(k) - P_s(k,l_c)]^2}.$$  

(7)

The threshold parameter $\delta_{4,\text{max}}$ is the upper limit for a positive detection.

f. Impulse decay feature

The last feature $f_5$ of the algorithm is the decay of the impulse. The model decay defines a linear decrease in signal power in decibel over time. The feature is calculated using the power over time in the frequency $k_p$. The time period starts at the time index $l_c$. The length of the time period is adjusted with the value $l_d$. The decay of the signal is approximated calculating the gradient of the simple linear regression by

$$f_5(l_c) = \frac{\sum_{i=l_c}^{l_c+l_d} (I-I)[10\log_{10}(P|k_p,l)-10\log_{10}(P|k_p)]}{\sum_{i=l_c}^{l_c+l_d} (I-I)^2}.$$  

(8)

Here the overbar symbolizes the arithmetic mean in the interval of $l_c$ to $l_c+l_d$. The decay of the example cracking sound and the approximated decay is shown in Figure 4. The time $t_c$ corresponding to the time index $l_c$ and $t_d$ corresponding to $l_d$ are marked in the figure. The upper and lower threshold parameters $\delta_{5,\text{max}}$ and $\delta_{5,\text{min}}$ define the detection interval for this feature. In case the five features are detected at a time frame $l_c$ the algorithm indicates that a cracking sound has occurred.
5 Results

The improved detection algorithm presented in Section 4 was used to process the rotor blade stress test recordings described in Section 3. The data of the recordings was manually labeled. The amount of cracking sounds in the flapwise fatigue test is unknown due to the long recording time. We built a training-set and a test-set by splitting all static test recordings equally and taking one hour of representative fatigue test data for both sets. We assume that the lower power impulses are less important for the damage detection purpose, so we split up the impulse signals into two groups, signals with a signal peak to average noise ratio of less than 30dB were labeled as low power impulses and signals with a ratio of more than 30dB were marked as high power impulses. The recordings of the optical microphone were used to get the further described results. The test-set and training-set are identical to the sets used in the previous publication [9].

With the improved detection algorithm there are only slightly better detection results in the test-set. A sensitivity of 84 % for signals with high power and a sensitivity of 52 % for cracking sounds with low power are achieved while there are no false positive detections. Nevertheless there is a significantly better detection performance in the fatigue test recording. Processing the 76 days fatigue test data provides 104 correctly detected cracking sounds and only six false positive detections. The amount of correctly detected cracking sounds is three times higher compared to the results of the previous algorithm and at the same time the amount of false positive detection decrease from 67 to six. We assume that the higher noise in the fatigue test leads to the better results in this recordings.

No critical structural damage occurred during the rotor blade tests, so in the fatigue test the cracking sounds happened significantly more often in the beginning of the test, where the structure adapted to the load.

Figure 4: Power over time in frequency with maximal power and linear regression curve.

The sensitivity is the number of correctly identified impulses divides by all impulses in the set.
6 SUMMARY
In this paper, an improved version of an algorithm for detection of impulse-like cracking sounds in rotor blades of wind turbines is presented. The proposed algorithm for damage detection is based on a model of the cracking sound. The model characteristics were displayed in the following features: power gradient, tonality, spectral slope, spectral similarity and decay. All features are calculated and checked by the algorithm. The impulse detection displayed in the power gradient feature was modified to measure the increase in power of frequency bands. With this modification the method is able to separate the decay of tonal background signals and cracking sound impulses. This provides better detection results in the 76 day recordings of full-scale fatigue test, 104 cracking sounds were detected while there are only six false positive detections. Compared with the previous version of the algorithm the amount of detected cracking sounds is about three times higher and the number of false alarms are significantly lower.

7 REFERENCES
INVERSE CALCULATION OF WIND AND WAVE LOADS FOR WIND TURBINE SUPPORT STRUCTURES

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Keywords: Inverse load calculation, wind and wave loads for wind turbines, offshore loads for lifetime extension

Summary: The knowledge of realistic loads that occur during the lifetime of wind turbines allows an analysis of the current state of their supporting members, which are the tower and the foundation. Such an analysis enables the extension of the structural lifetime over the predicted period of 20 years and consequently enhances the cost-effectiveness of wind turbines. An approach for the determination of realistic loads is the inverse load calculation. This approach determines the realistic load values from measurements of the dynamic structural responses of the tower. This paper presents a numerical study using the comprehensive simulation code FAST. The application of the inverse load calculation procedure to a 5 MW offshore wind turbine model is shown. As a result, wind and wave loads are calculated inversely. In order to discuss the quality of the results, the inversely calculated loads are compared to the applied wind and wave loads.

1 INTRODUCTION

The wind energy industry, as a relatively young sector, will face a new challenge soon. The designed lifetime of 20 years expires for many wind turbines. In order to cope with this demand, the GL developed a guideline that addresses the determination of the lifetime extension of wind turbines [1]. A main issue is the knowledge of realistic loads [2]. There are several research projects that aim on determining realistic loads for wind turbines from measured data using an inverse approach (e.g. [3], [4] and [5]). But so far, all of these researches are limited to the determination of only aerodynamic loads. The presented paper will describe the determination of combined aerodynamic and hydrodynamic loads. A numerical model of a 5 MW offshore wind turbine is used for this study. The wind turbine model is created and calculated in FAST. FAST considers the aero-servo-hydro-elastic characteristics of the wind turbine structure. Using FAST has the advantage that the applied loads
are known so that these applied loads can be compared to the inversely calculated loads in order to assess the result quality gained from the inverse calculation. This paper describes the ongoing research on the topic of the inverse load calculation for wind turbines. In [6] a similar wind turbine model as used in this study is investigated with only aerodynamic loads. Different load cases had been considered that contained effects of the wind turbine control. Hence, this study takes the following aspects for granted: Aerodynamic loads can be calculated inversely with a very good accuracy and the wind turbine control does not disturb the accuracy of the inverse calculation.

2 APPROACH FOR THE INVERSE LOAD CALCULATION

The inverse load calculation is based on the following principle: The dynamic responses of the support structure are measured. Additionally, a mathematical system description of the wind turbine structure is needed. Combining the structural responses with the system description equals an inversion of the typical design procedure in which the defined loads are combined with the designed system description in order to gain the system responses.

Figure 1: Load assumptions for the inverse calculation

The objective of the suggested approach for the inverse load calculation is the determination of a rotor thrust force (representing the aerodynamic wind load) and a resulting wave load (representing the hydrodynamic wave load), as depicted in Figure 1. The structural responses are measured along the tower height. Accelerations are recorded. Additionally, strain measurements at the tower bottom are required for the determination of the static load component of the aerodynamic load.
Mathematically, the inverse calculation is an ill-conditioned problem. In order to solve the inverse problem, a frequency-domain based method is used. This method is based on the second-order ordinary differential equation for a linear, time-invariant vibrating system. A more detailed description of the underlying theory is discussed in [7]. The solution of the inverse problem in the frequency domain is given in eq. (1).

\[ F_{m,1}(j\omega) = H^+_{m,n}(j\omega) \cdot Y_{n,1}(j\omega) \]  

(1)

The load vector \( F_{m,1}(j\omega) \) contains the m inversely calculated forces. The n measured system responses are represented by the vector \( Y_{n,1}(j\omega) \). In terms of the presented numerical study, the simulated structural displacements represent the measurement data. Eq. (1) describes an over-determined system of equation with \( m \leq n \). This means, that the number of measurement locations (and consequently the number of DOF’s) can be greater than the number of inversely calculated loads. This over-determination allows smoothing random errors in the measurement data and enables the consideration of an arbitrary number of DOF’s. The solution of the over-determined system of equation follows a least-squares approach that is accomplished with the pseudo-inverse \( H^+_{m,n} \) that is shown in eq. (2).

\[ H^+_{m,n} = \left[ H^T_{m,n} \cdot H_{n,m} \right]^{-1} \cdot H^T_{m,n} \]  

(2)

The FRF-matrix \( H \) has to be known for the solution of eq. (2). The FRF-matrix contains the system description, defined by the eigenfrequencies and damping ratios of each DOF. These parameters assemble the generalized FRF-matrix \( H^{-1}_g \) according to eq. (3).

\[ H^{-1}_g(j\omega) = -\omega^2 M_g + j\omega B_E + K_g \]  

(3)

The transformation of the generalized FRF-matrix \( H^{-1}_g \) to the FRF-matrix \( H \) is based on the multiplication with the modal matrix \( U_0 \), as shown in eq. (4).

\[ H^{-1}(j\omega) = (U_0^{-1})^T \cdot H^{-1}_g(j\omega) \cdot U_0^{-1} \]  

(4)

The required parameters that are needed to set up the system description – the eigenfrequencies, the damping ratios, and the modal matrix – are derived from the FAST model. The solution of eq. (1) gives the inverse loads in the frequency domain. The transformation of the inversely calculated loads can be done with an inverse FFT according to eq. (5).

\[ f(t) = F^{-1}\left\{ H^+ \cdot Y(j\omega) \right\} \]  

(5)

In case this approach is applied to real wind turbines, additional requirements have to be fulfilled. This requirements concern e.g. the treatment of the measurement data or the determination of the dynamic system properties. The references [8] and [9] give recommendations for the handling of the mentioned requirements.

3 DESCRIPTION OF THE INVESTIGATED 5 MW OFFSHORE WIND TURBINE MODEL

A three-bladed upwind variable-speed variable pitch-to-feather controlled turbine with a hub height of 90 m above SWL is investigated. The rotor blade lengths are 63 m. The wind turbine is carried by a tubular steel tower. Below SWL, the tower is supported by a fixed-bottom (apparent fixity) monopile substructure. The water depth is set to 20 m, which equals the height of the substructure. Thus, the height of the entire structure from mudline to hub height
is 110 m. The diameter of the steel monopile is 6 m. Figure 2 depicts the model. Detailed information about the technical specifications of the wind turbine model is given in [10].

![Offshore monopile model of the 5 MW wind turbine](image)

Figure 2: Offshore monopile model of the 5 MW wind turbine [11]

The aerodynamic load is defined by a stochastic turbulent wind field. Normal wind conditions for power production conditions are chosen. The load case chosen has a mean wind speed of 18 m/s, which represents a mean wind speed at hub height between rated and cut-out wind speed. In this load case high control activity is expected. The yaw position is constant at zero. The collective blade pitch ranges between pitch angles of 10° to 20°. The control strategy provides “constant power”. The mean rotor speed is 12.10 rpm.

The hydrodynamic loads are described by a stochastic sea state that is defined by the JONSWAP spectrum. In order to fulfill the correlation with the chosen wind speed, a significant wave height $H_S = 5$ m and a peak spectral period $T_P = 12.4$ s is used. The simulation time is set to 3,600 s. The FAST simulation time steps are $dt = 0.0125$ s. In order to limit the amount of data, a decimation factor of 4 is applied to the output data, so that a time increment of 0.05 s and a Nyquist frequency of 10 Hz are produced. Both values guarantee sufficient accuracy for the frequency-domain transformations.

4 Results

a. Description

Figure 3 shows the inversely calculated aerodynamic load, both in the time and in the frequency domain. For verification purposes, the rotor thrust force that is computed by FAST is also displayed. Figure 4 shows the inversely calculated hydrodynamic load, also in the time and in the frequency domain. Now, the verification is done by comparing the inversely calculated load to the resulting wave load that is based on the hydrodynamic forces calculated by FAST.
Figure 3: Rotor thrust and inverse aerodynamic load

Figure 4: Resulting wave load and inverse hydrodynamic load
b. Aerodynamic loads

Figure 3 shows a good match between the rotor thrust of the simulation and the inversely calculated load. The full inverse aerodynamic load is obtained by superimposing a static load component and a dynamic load component. The dynamic load component is the result of applying eq. (1) and (5). The comparison in the frequency domain shows that the inverse calculation is able to reproduce the frequencies of the rotor thrust. However, amplified peaks in the spectra occur at ca. 0.6 Hz to 0.7 Hz. The corresponding time-domain depictions show inversely calculated loads with higher amplitudes than the rotor thrust. These amplified peaks correspond to the rotor-blade eigenfrequencies. Their reason is that rotor blade dynamics are not part of the dynamic system description used for the inverse calculation. Their vibrations are interpreted as external loads. The rotor blade dynamics are neglected in order to simplify the calculation procedure.

c. Hydrodynamic loads

Figure 4 shows the inversely calculated hydrodynamic load compared to the resulting wave load that is based on the FAST output. The wave load in FAST is generated using a JONSWAP spectrum with a peak spectral period of $T_p = 12.4$ s. As a consequence, a peak occurs at 0.08 Hz in the resulting wave load (blue signal in Figure 4). Additionally, the spectrum of the resulting wave load contains energy in a narrow frequency band that ranges from ca. 0.06 Hz to 0.25 Hz. This frequency band shows a steep slope left to the maximum and a flat slope right to the maximum, which also fulfills the expectations on the JONSWAP spectrum. The inversely calculated hydrodynamic load (green signal in Figure 4) reproduces these described characteristics. The same narrow frequency band is visible, with a maximum at approximately the spectral peak and the characteristic slopes left and right to the maximum. The reasons for the differences between the hydrodynamic load signals are the following: 1) The system simplification in terms of the rotor blade dynamics also counts for the inverse calculation of hydrodynamic loads. 2) The mathematical system description of the structure is gained from a separate simulation run, a so-called linearization. The linearization in FAST does not account for the hydrodynamic loads. And 3) The presented procedure for the inverse calculation of hydrodynamic loads assumes a fixed position of the resulting wave load. The assumption represents an approximation.

d. General remarks

The inverse load calculation procedure produces reasonable results. This statement is true both for the inversely calculated aerodynamic and hydrodynamic loads when compared to the FAST reference loads. The presented inverse calculation procedure induces uncertainties to the inversely calculated loads when compared to the applied loads from the FAST simulation. These uncertainties are said to be low. Furthermore, these uncertainties always cause an overestimation of the amplitudes, visible both in the depictions of the time domain and the frequency domain.

5 Conclusions

The presented paper aims at investigating the inverse load calculation that considers combined wind and wave loads. The comprehensive simulation code FAST is used for a verifica-
tion study of the inverse load calculation. FAST allows loads to be applied from a defined stochastic wind field and enables the interacting dynamics of the wind inflow, aerodynamics, elasticity, and the control of the wind turbine to be considered. Additionally, FAST considers incident waves, sea currents, hydrodynamics, and foundation dynamics of the support structure. A three-bladed 5 MW offshore wind turbine model is used. An approach is introduced in which the wind load is reduced to the rotor thrust and the wave load is represented by a resulting wave load. The accuracy of the inverse load calculation is estimated by comparing the inversely calculated loads with the applied loads from FAST.

The presented results show that the inverse load calculation is capable of generating good estimates of the applied wind and wave loads. Important assumptions and simplifications of the presented approach for the inverse calculation procedure are discussed briefly. The conclusions are derived from only one representative load case. Thus, their general validity has to be confirmed by further verifications.

The presented study is intended to be a first step for the application of the inverse load calculation to real-world offshore wind turbines. The potential of knowing the loads of wind turbines that occur during their lifetime is seen in the prediction of a lifetime extension.

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7 REFERENCES

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AN APPROACH FOR LOCAL-GLOBAL STRUCTURAL HEALTH MONITORING FOR OFFSHORE WIND ENERGY CONVERTERS

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Keywords: Structural Health Monitoring, Stochastic Subspace Identification, vibration based, local displacement measurement, Modal Assurance Criterion

Summary: Automated Structural Health Monitoring (SHM) is a useful tool for offshore wind energy operators, because it is able to reduce maintenance cost dramatically. Due to the limited information a global non model based SHM system is able to provide, a combination of local damage detection systems with global SHM methods seems to be a valuable approach.

The work presented describes the combination of a global SHM system based on modal parameters with data gathered from local displacement measurement data. This SHM scheme is then applied at a scaled steel model of a tripile structure. The global SHM system works with changes in the modal parameters that were gathered using data driven Stochastic Subspace Identification (SSI) from acceleration data. Comparison of damaged and undamaged system states is performed using the Modal Assurance Criterion.

The data from the local displacement measurement show qualitative and quantitative changes in the structural vibration behaviour. Comparison of damaged and undamaged system states lead to a different damping behaviour. This is analysed using the logarithmic decrement.

1 INTRODUCTION

Due to their bad accessibility in the open sea, visual inspections of offshore wind energy converters (OWECs) cause tremendous expenses for maintenance. To reduce financial risks reliable SHM- systems have to become part of maintenance strategies of offshore wind energy operators. Furthermore, health monitoring is a powerful tool for surveying the actual state, and, finally, predicting the remaining lifetime of OWECs. In doing so, the profitability of wind energy can be increased significantly.

Vibration based monitoring is one of the few methods, that is able to determine a structure’s damage on a global scale. In [1], four levels of damage identification are introduced: determination of damage existence, determination of damage location, quantification of
damage, prediction of remaining service life. As stated in [2], vibration based global damage identification methods provide level 1 and level 2 in damage assessment. According to [3], measurements at hot spots are useful to make a statement about damage quantification (level 3 of damage identification). However, this statement is only valid for local spots, which is in contrast to the motivation of collecting damage information of the whole structure. Hence, a combination of global methods with monitoring of local hot spots seems to be a promising approach for the development of a holistic SHM system. With information on damage at hot spots in combination with global measurements a statement concerning structural damages of the OWEC can be given.

2 Scaled Model of an Offshore Wind Turbine
For examination and testing of coupled local- global SHM approaches a 1:50 scaled model was built. The design refers to a typical OWEC tripile substructure (see figure 1a). The blades and the turbine are simplified by an extra mass applied to the top of the tower. All components are made of steel.

For validation of the local-global SHM approach, a possibility to vary the stiffness of local structural components was integrated via two head plate joints. Local stiffnesses were modified via loosening bolts at the plate joints completely at points A and B (figure 1b). This simulated damage will be referred to as “damage A” and “damage B” within this paper. Due to the sensor arrangement, local damage can only be monitored at point A. During the testing period, the steel plate (see figure 1) was bolted to the floor. The structure was excited via a sudden release of a defined head deflection in x- direction.
Global monitoring of the Structure was conducted using the structure’s modal properties. Therefore, accelerations were measured using triaxial accelerometers with a sampling frequency of 600Hz at four different height levels on the structure (see figure 1a). Due to the small oscillations in vertical direction, only the two horizontal accelerations were recorded. System identification was performed using data driven SSI. The algorithms used are described in detail in [4].

Obviously, the identified eigenfrequencies depend on the structure’s damage state, see figure 2. While damage A causes a decrease of about 7% in the first eigenfrequency, damage B reduces the first eigenfrequency by about 26% (see figure 2a). The 3rd and 5th eigenfrequencies show a similar behaviour: Both damage A and B cause a decrease of the eigenfrequency, while the magnitude of the decrease is bigger for damage B (figure 2c and d). This is different for the second eigenfrequency (figure 2b). The second eigenfrequency could not be identified in the acceleration signals in all of the experiments. This is due to the direction of the structure’s excitation. The structure was excited in x- direction (see figure 1), but the second eigenmode is a global oscillation of the tower head in y- direction, similar to a cantilever beam (see figure 3a). This was even more critical for the fourth eigenmode. Due to this reason, the fourth mode shape is not considered any further within this paper.

Figure 2 - Identified Eigenfrequencies
Different to the other considered eigenfrequencies, the second eigenfrequency does not change significantly dependent on the damage type (see figure 2b). This is because the damaged leg of the tripile is in the neutral axis of the second eigenmode (see the dash dotted line in figure 3a). Thus the damaged leg contributes very poorly to the load transfer of the second eigenmode and has a small influence on the eigenmode.

Due to the small influence of the damaged leg to the 2nd eigenmode, the corresponding mode shape remains constant as well. With the limited measurement setup, there is no information on the undamaged legs of the tripile available and therefore they are left out in the illustration of the deformed eigenmodes (see figure 3).

While there is almost no damage dependent behaviour of the second mode shape visible, the other eigenmodes show a good dependency. Exemplarily, the fifth eigenmode is illustrated in figure 3b and c. This y-oriented eigenmode seems to be dependent on the considered damage states. Loosening bolts leads to a decoupling of the damaged leg from the structure. In contrast to the undamaged state, the leg is almost undeformed within both damage scenario A and B (see the green ellipsoid in Figure 4 c). Both damage scenarios result in a similar mode shape, therefore only the mode shape of damage A is shown here.

![Image of mode shapes](image1.png)

**Figure 3 - Mode shapes of the structure**

Damage dependency of the eigenmodes can be affirmed using the Modal Assurance Criterion (MAC) [5]. MAC- values bigger than 0.8 indicate a good correlation between two vectors, therefore every value smaller than 0.8 is set to zero here.

The fifth eigenmode could not be identified in experiment number 6, thus there are no eigenfrequency, eigenvector and MAC- values available for this time series (see figure 4).

Within the three investigated system states the MAC remains nearly constant, especially in the undamaged status, see the diagonal terms marked with dahed black boxes in figure 4. When damage is applied, the MAC- values drop below 0.8 in almost all of the experiments, see the off- diagonal terms marked with the dashed blue boxes in figure 4. This indicates a
modification of the 5th mode shape after damage application, there is no correlation between the vectors anymore. In figure 3 it is shown that the 5th mode shape is similar for both damage A and B. This statement is confirmed by the MAC, the unboxed areas in figure 4 show a weak correlation (MAC between 0.8 and 0.9) between the mode shape of damage A and B. This behaviour is similar for the other considered eigenmodes, except the second. Hence, the MAC- values can be used as a damage parameter for global SHM. It has to be distinguished in advance and individually for each structure, if the considered eigenmode is dependent on the considered damage or not.

Figure 4 - MAC for fifth Eigenmode

The values of about 0.9 in experiment number 13 and 14 indicate a stronger correlation between those eigenvectors and the undamaged system. Therefore, for exclusion of random effects, comparison of several time series of both undamaged and damaged structure as it was done here is recommended. Due to the sensitivity of the eigenfrequencies and the MAC- values, a combination of the modal parameters to a damage parameter is a promising approach. Damage localization or quantification according to [1] is not possible with this damage parameter. For this purpose, local measurements can be used.

4 LOCAL MONITORING OF DAMAGE HOT SPOTS

Local monitoring of the support structure has particular advantages to screen out parts of the structure that are affected by structural changes that ultimately lead to changes in the deformation or stiffness behaviour. In combination with the global monitoring, the specific evaluation of local measuring points can be used for the quantification of the structural state such as damage propagation. The grouted joint is a suitable structural component in OWEC in order to conduct local monitoring. A practical implementation of such a local monitoring is documented in [6].
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Tests have already been conducted on the structural model with the objective of verification and improvement of the methods for data processing. The target value for a local monitoring is the relative displacement between two points of the structure. Although the target value of the measurement is a 1D displacement, the measurement setup for the local monitoring consists of three laser distance sensors that are aligned parallel to the axis of the brace. Laser distance sensors from Welotec GmbH with a measuring range of 10 mm and a resolution of 2–5 μm were used. Magnetic attachment constructions and adjustable tripods enable an exact positioning of the sensors.

![Figure 5 - Measurement set up for local monitoring](image)

The measurement set-up of local monitoring is exemplified in figure 5. With a measuring length of 198 mm, laser No. 1 measures the relative displacement between two points of the model. The local stiffness modification of the model is positioned between these two points. The lasers No. 2 and No. 3 are used to verify the relative measurement and record the absolute displacements of the two points.

![Figure 6 - Local monitoring of undamaged model](image)
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The local displacements of the two model variations are shown in figure 6 and figure 7. The measurements start with release of the defined head deflection and run for 20 seconds. In addition to the displacement the maximum and minimum peak values of two repeated experiments are visualised in figure 6 and figure 7.

![Figure 7 - Local monitoring of damaged model](image)

To quantify the change of the vibration behaviour the logarithmic decrement is considered.

\[ \Lambda = \frac{1}{n} \ln \frac{x(t_0)}{x(t_n)} \]  

(1)

The logarithmic decrement is calculated by equation (1) in which \( x(t_0) \) is the first amplitude after the release of the defined head deflection and \( x(t_n) \) is the amplitude \( n \) peaks later. The evaluation revealed a convergence of the ratio \( \Lambda_{\text{dam}}/\Lambda_{\text{undam}} \approx 2.5 \) at \( n \geq 8 \) which corresponds to a time of 3.5 seconds. The reason for the different behaviours of the models is assumed to be that the released energy was dissipated quickly by friction in the loosened flange. The applied damage leads to an influence on the structural vibration behaviour and can be measured and quantified by local monitoring.

5 CONCLUSION

This paper demonstrates the potential of combined global vibration based SHM and local measurement data. This combination is a useful approach for the development of a holistic SHM system. While global SHM is able to detect damage existence via the MAC, the local measurement is able to detect if damage is existent at a measured damage hot spot or not and can thus contribute to damage localization and to the interpretation of damage detection. In order to get good results for the global SHM method, the correct choice of damage dependent eigenmodes is essential due to their damage sensitivity.

The comparison of the local displacement measurement data of the two model variations show explicitely that it is possible to detect qualitative and quantitative changes in the structural vibration behaviour using a local monitoring system. Furthermore, local sensors can help identifying parts of the damping matrix of the structure, which is often quite inaccurate for offshore structures.
In offshore environment, loading conditions are not that precise and perfect as in the tests presented here. For investigation of more realistic loading conditions the model will be installed in a wave flume in further works.
In this study severe damage was applied to the structure. Therefore, the objective of ongoing research is to test the methods sensitivity to smaller damages and use the modal data for automated updating of numerical models in order to perform damage localization and damage quantification as a model based SHM approach.

6 REFERENCES
Resonance analysis of offshore wind farms using doubly feed induction generators under application of grid transfiguration approach

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Keywords: Resonance, Generator, Converter, Frequency response

Summary: This paper presents an approach for a model of a doubly feed induction generator (DFIG) appropriate for the resonance mode analysis (RMA). The main idea of this approach is to consider the model of DFIG not as a close unit, but to split up into individual parts. Every part is constructed as a quadripole according to the quadripole theory [6, 7]. Every quadripole can then be added together according to the nodal approach [8,9]. This approach leads to a positive effect which can be effectively used by the RMA represented in this paper. So, it becomes possible to analyze the impact of filters the DFIG includes on the resonance phenomena within the grid of offshore wind farms. This also, allows localizing the reason for a resonance much more precisely. Thus the advantages of the RMA can be used optimally. The whole model presented in this paper is verified with relevant commercial software tools. A sensitivity analysis is accomplished.

1 Introduction

The grid connection of offshore wind farms (OWF) is associated with high challenges. The problem of resonances in the whole OWF grid connection system could become an ever-growing problem. Modern OWF are including a lot of equipment which is dominated by power electronic switch devices like DFIG. The bulk of this power electronic equipment produces a lot of harmonics within the grid [1]. So, it becomes very important to know the harmonic resonance to avoid some resonance phenomena within the connection system of OWF. The RMA used in this paper is a powerful tool to identify the harmonic resonance of the grid and to analyze the resonance phenomena [2]. The RMA needs appropriate models of all electrical devices within the connection system of OWF. The models of most classical
electrical devices like transformers and cables are well known [3]. Unfortunately there are no established standards for a DFIG model which are explicitly suitable for the RMA.

The connection of OWF in Germany is realized by two partners the owner of the connection system (transmission grid operator) and the owner of the OWF grid. Both partners have to manage resonance analysis throughout the planning phase for the grid connection of OWF. This leads to a problem, since none of the partners has an interest to deliver detail data about the topology and the parameters of the equipment used in its own system. This problem can be solved by using the approach of grid transfiguration [4, 5].

2 Resonance Mode Analysis

The RMA is a powerful tool to analyze the harmonic resonances [2, 1]. It provides helpful solutions to mitigate harmonic resonance problems. Such as the possibility to find the bus which can excite a particular resonance more easily and where the resonance can be observed. A very useful characteristic of the RMA is the possibilities to identify the components are involved in the resonance. This characteristic is very well used in the approach for modeling the DFIG presented in this paper.

3 Model of a Doubly Feed Induction Generator for a Resonance Mode Analysis

The concept is based on a single line diagram of an induction machine model used for harmonic resonance analysis [10, 11]. For RMA most physical phenomena such as skin and proximity effect cannot be neglected [12]. These phenomena are modelled in frequency dependent parameters of the DFIG. However other frequency depending effects such as the frequency dependency of the slip of an induction generator are additionally considered. The approach for a model of a DFIG is illustrated in figure 1. The main idea of this approach is to consider the direct current (DC) connection between machine side converter and grid side converter as an interface, since the share of a DC connection on the harmonic resonances is negligible.

As a part of this paper different types of filters used in DFIGs are analyzed and sufficiently dimensioned. The figure 1 shows the entire topology of DFIG-Modell, analyzed in this paper. The entire topology is a construct of particular equipment such as converter, filter and the frequency-dependent model of an induction generator. Each equipment model is modeled as a quadripole. Hereinafter every quadripole model is introduced.

Figure 1: Topology of the modeled DFIG configuration
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a. frequency-dependent model of an induction generator

The equivalent circuit diagram of an asynchronous generator with an external voltage on the secondary or rotor circuit side and the frequency-dependent parameters defined here is shown in figure 2.

\[
\begin{bmatrix}
    U_1(h) \\
    U_2(h)
\end{bmatrix} = \begin{bmatrix}
    \sqrt{h} \cdot R_1 + j h \cdot (X_{1\sigma} + X_{1h}) \\
    s_{h1} \cdot \frac{1}{iu} \cdot jhX_{1h} \cdot \frac{1}{iu} \cdot jhX_{1h}
\end{bmatrix}
\begin{bmatrix}
    L_1 \\
    L_2
\end{bmatrix}
\]

Figure 2: Topology of frequency-dependent model of an induction generator

The frequency-dependent reactances in figure 2 emphasize the skin effect inside the generator windings. The reactances are linearly dependent on the frequency [13, 14]. The frequency-dependence of the resistance depends significantly on the skin effect as well as losses as a result of eddy currents [15]. The frequency dependence of the slip [14, 16] is also considered here. The equation 1 describes the quadripole of the induction generator model.

b. Grid-side and rotor-side filter

Frequency filter can be generally classified as active, passive or hybrid filters. The filter models described in this paper are restricted to passive filters. Both rotor-side and grid-side filters are so-called LCL filters. The equivalent circuit diagram of an LCL-filter is shown in figure 3.

Figure 3: LCL-Filter

The frequency-dependent response of the LCL-filter is very complex and is described through a linear addition of an L-filter and an RC-filter. The guideline for the dimension of the frequency-dependent filter parameters can be seen e.g. in [17, 18]. The grid-side filter is connected as a star configuration. The equation (2) describes the quadripole of the grid-side filter.
The rotor-side filter is connected as a delta configuration. The reactance near to the rotor-side can be neglected due to the relatively large leakage reactance of the generator. The equation (3) describes the quadripole of the rotor-side filter.

\[
Y_{\text{RSF}}(h) = \begin{bmatrix}
R_{3,N} + j\left(hX_{L1,N} - \frac{1}{h}X_{C3,N}\right) & R_{3,N} - \frac{1}{h}jX_{C3,N} \\
R_{3,N} - \frac{1}{h}jX_{C3,N} & R_{3,N} + j\left(hX_{L2,N} - \frac{1}{h}X_{C3,N}\right)
\end{bmatrix}^{-1}
\]

\[
Y_{\text{RSF}}(h) = \begin{bmatrix}
R_{3,R} + j\left(hX_{L1,R} - \frac{1}{h}X_{C3,R}\right) & R_{3,R} - \frac{1}{h}jX_{C3,R} \\
R_{3,R} - \frac{1}{h}jX_{C3} & R_{3,R} - \frac{1}{h}jX_{C3,R}
\end{bmatrix}^{-1}
\]

4 APPROACH FOR THE GRID REDUCTION OF PASSIVE GRIDS

The aim of the grid transfiguration method or also called the grid reduction method is to acquire a grid equivalent which is a part of fully meshed or not meshed grid [5]. The nodes, the grid equivalent includes, are eliminated through the grid transfiguration method. The approach of the grid transfiguration is used for passive grids explicitly [4]. The received grid equivalents consist of admittances or impedances. They can also be connected to each other through impedances or admittances.

The approach or grid transfiguration is showing in figure 4, where two passive partial grids are illustrated. Both grids are connected to each other via connection bus bar. By using the grid transfiguration method every partial grid can be transfigured into a grid equivalent. All nodes of this partial grid are eliminated.

![Equivalent circuit diagram of a sample grid.](image)

The equation (4) describes the system in figure 4 completely with its nodal voltages and nodal currents.

\[
\begin{bmatrix}
Y_{EE} & Y_{FE} & 0 \\
Y_{KE} & Y_{KK} & Y_{KF} \\
0 & Y_{FK} & Y_{FF}
\end{bmatrix}
\begin{bmatrix}
U_E \\
U_K \\
U_F
\end{bmatrix} =
\begin{bmatrix}
I_E \\
I_K \\
I_F
\end{bmatrix}
\]

Considered that the entire grid is passive the elements of the nodal current vector are zero.
After the grid transfiguration the grid admittance matrix of equation (4) is transformed, as one can see from equation (5).

\[
\begin{bmatrix}
Y_{red} & Y_{KF} \\
Y_{FK} & Y_{FF}
\end{bmatrix}
\]

(5)

where

\[
Y_{red} = Y_{KK} - \left( Y_{KF}Y_{FF}^{-1}Y_{FK} \right)
\]

(6)

is the contribution of the transfigured grid.

With this a RMA can be done without detail information about the grid which was transfigured.

5 CASE STUDIES

The figure 5 shows a pattern grid of an OWF connected to the transmission grid.

As one can see in figure 5 two offshore wind farms are connected to the grid via high voltage direct current transmission system (HVDC). Both have eighty generators of a DFIG type. The left offshore wind farm is transfigured using the grid transfiguration method presented above. The results of the RMA with the complete grid and with the reduced partial grid of left offshore wind farm are showed in figure 6.
The grid transfiguration method, the approach for a resonance analysis model of a DFIG and the resonance mode analysis are demonstrated using the pattern grid in figure 4.

![Figure 6: results of the RMA using the proposed DFIG model for a complete grid and a partially transfigured grid](image)

6 CONCLUSIONS

Resonance phenomena within the grids of offshore wind farms become more and more a serious problem. Modern offshore wind farms include DFIGs. Unfortunately there are no established standards for a model of DFIG which is explicitly suitable for the resonance mode analysis. In this paper an approach for a model of DFIG suitable for the RMA is proposed. After verifying this model with commercial software and a sensitivity analysis the model was successfully implemented into the program of resonance mode analysis.

- The modular design of this approach based on the quadripole theory provides detailed information about the impact of the DFIG on resonances within the grid of offshore wind farms and the whole transmission grid connection system.

The grid transfiguration method for resonance analysis presented in this paper permits to use the RMA without knowing the detailed data of partial grids.

- The most important finding of the analysis of this approach is that with the elimination of nodes within the transfigured grid the possibility to identify some resonance frequencies is not given anymore.
- This means that just resonances of the not transfigured grid are identifiable. But it is well possible to identify the resonances which would lead to overvoltage on the nodes of the not transfigured grid.
7 REFERENCES

POWER SYSTEM OSCILLATION DAMPING WITH WIND POWER PLANTS

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Keywords: Power System Oscillation, Inter-Area Oscillation, Wind Power Plant, Damping Control, System Stability, Frequency

Summary: Low frequency inter-area oscillations in the European power system in the range of 0.2Hz to 1.5Hz can lead to system instability in the worst case. Therefore it is proposed to modulate a frequency proportional active power feed-in to wind power plants to increase inter-area oscillation damping. A simulation model of a wind power plant has been built and tested with the proposed control algorithm.

1 INTRODUCTION

The structure and the way of operation of the European power system have changed in the past years and decades. The connection of Eastern European countries amongst others has lead to a wide geographic extent. Furthermore electric energy transmission between operating areas has increased due to market deregulation and more renewable energies installed in the grid. At the same time the single areas of the power system are only weakly coupled [1, 2]. Another relevant aspect is the ongoing replacement of electrical drive systems that are directly connected to the grid by converter driven drives. The latter feature a constant power drain compared to the grid frequency proportional power consumption of the directly connected drives.

These changes promote the occurrence of power oscillations between the connected areas, so called inter-area oscillations with a low system damping and an oscillation frequency between 0.2Hz and 1.5Hz. During inter-area oscillations the grid frequency can exceed its limits of 50Hz ± 200mHz and power lines can be overloaded. Power system islanding or even blackout can be a consequence [3]. Figure 1 shows an event in 2011. Even after 30s the magnitude of the frequency oscillation is only slightly damped.
An approach for better system damping with electric car batteries is described in [4]. In [5] the application of the generator power system stabilizer principle on wind power plants is discussed.

In this paper it is proposed to increase inter-area oscillation damping with wind power plants by adding a portion to the active power feed-in that is proportional to the grid frequency. This counteracts the active power imbalance in the grid that occurs during inter-area oscillations.

![Fig. 1: Low Frequency Power Oscillation in 2011 [7]](image)

2 Wind Power Plant Model

In order to verify the proposed inter-area oscillation damping method, a simulation model of a 1.7MW wind power plant (WPP) has been built in Matlab Simulink (fig. 2). It consists of a wind turbine, a two-mass-oscillator, an electrically excited synchronous machine, a diode rectifier, a boost converter and a voltage source inverter (VSI).

![Fig. 2: Block diagram of the wind power plant model](image)
The wind turbine power is calculated according to equation 1 and 2 with the air density $\rho$, the turbine area $F$, the wind speed $v_{\text{wind}}$, the turbine power coefficient $c_p$, the tip speed ratio $\lambda$ and the turbine blade pitch angle $\vartheta$. It is necessary to determine the power coefficient $c_p$ from the current values of $\lambda$ and $\vartheta$ in order to obtain the correct dynamic behavior during small turbine speed deviations as it occurs with the proposed damping control strategy.

$$P_{\text{Turbine}} = \frac{1}{2} \rho F v_{\text{wind}}^3 c_p \tag{1}$$

$$c_p = f(\lambda, \vartheta) \tag{2}$$

The two-mass-oscillator is used to emulate the dominant eigenfrequency of the drive train that is caused by the bending of the turbine blades. A possible excitation for the oscillator is a gust of wind. The eigenfrequency of the modeled WPP is 2.5Hz. A mechanical oscillation in the drive train influences the electrical system and should therefore be taken into account.

The 3-phase voltage generated with the synchronous machine is rectified, converted to 1100VDC with the boost converter and fed into the power system with a VSI. With the inductor current of the boost converter the power of the WPP is controlled. The reference power is determined by a look-up table with the steady-state power value corresponding to the current generator speed (fig. 3). The values for reference power represent the optimal power exploitation of the wind turbine.

The VSI is used for maintaining the dc link voltage on a constant level of 1100V.

![Fig. 3: Block diagram of the boost converter control](image)

### 3 Control Method

As mentioned in the introduction, there are circumstances that can excite the electrical power system and lead to grid frequency oscillations. These frequency oscillations are connected with active power oscillations. Therefore it is proposed to add to the active power feed-in $P_{\text{Wind}}$ of the WPP a part $P_{\text{ref}, \text{Damping}}$ that is proportional to the deviation of the grid frequency $\Delta f$ (eq. 3, 4, 5).

$$P_{\text{WPP}} = P_{\text{Wind}} + P_{\text{ref}, \text{Damping}} \tag{3}$$

$$P_{\text{ref}, \text{Damping}} = -k \Delta f \tag{4}$$

$$\Delta f = f - f_N \tag{5}$$

If the current value of the frequency $f$ is higher than the nominal value $f_N$, the active power feed-in of the WPP needs to be reduced. So $P_{\text{ref}, \text{Damping}}$ becomes negative. For $\Delta f<0$ the active power feed-in has to be increased, so $P_{\text{ref}, \text{Damping}}$ is positive. The amount of the oscillating power $P_{\text{ref}, \text{Damping}}$ can be scaled by the factor $k$. 


For a positive frequency deviation $\Delta f > 0$, the power exploited from the wind is greater than the power feed-in into the power system. This difference in energy is stored into the rotating system consisting of the wind turbine and the drive train. Thus, the system is accelerating and for $\Delta f < 0$ it is decelerating respectively. In order to not disturb the operation of the wind power plant, the speed deviation $\Delta \omega$ caused by the proposed damping control needs to be very small compared to the operation speed. Therefore only inter-area oscillations with a period time well below the mechanical time constant of the wind power plant can be damped. For the simulated wind power plant it is $T_{\text{mech}} = 25\text{s}$. Thus, this control is not applicable for primary control, which demands to supply power to the power system in the time range of 30s to 15min.

Furthermore, only power oscillations with a frequency below the dominant eigenfrequency of the mechanical system of 2.5Hz can be damped. Otherwise the 2-mass-oscillator would be excited with its eigenfrequency by the damping control, what would lead to unacceptable high speed deviations. In order to constrain the damping control to the discussed frequencies, a bandpass is used as it can be seen in figure 4. The lower frequency limit has been chosen to 0.2Hz and the upper limit to 1.5Hz.

It is necessary to mention, that the proposed control method does not emulate a virtual inertia. For virtual inertia emulation the active power is proportional to the rate of change of frequency (eq. 6) as it has been done for example in [6].

$$P_{\text{inertia}} = c \times \frac{df}{dt}$$ (6)

4 RESULTS

The proposed control method has been tested with the WPP in figure 2. To emulate an inter-area oscillation, a frequency oscillation has been applied to the system with a frequency of 0.25Hz and a magnitude of 200mHz. During simulation a constant wind speed of 5.5m/s is assumed.

In figure 5 the simulation results of the wind power plant active power feed-in and the grid frequency are shown. About 5s after simulation start, the active power settles to its steady-state value of about 476kW. At time $t=10s$ the frequency oscillation starts. It can be observed,
that the active power drops when the frequency rises and vice versa. The phase shift between the two oscillations is about 180 degrees. Further it can be seen that the value of the active power deviation is dependent on the frequency deviation $\Delta f$. Thus the active power feed-in of the WPP features the desired frequency proportional characteristic. For a deviation of 200mHz, which is the maximum allowable value in the European system, the active power deviation is about 10kW. This is 0.6% of the systems nominal power.

Figure 6 shows the rotation speed of the wind turbine. At time $t=10s$ it has almost settled to its steady-state value of 10.225rpm. When the frequency oscillation is applied, an oscillation in the turbine speed can be observed. The speed drops, when the current active power output of the WPP is higher than the mean value and rises when it is above. The speed deviation from the stead-state rotation speed is about 0.02% and therefore negligible.

The WPP active power output response for a step of the grid frequency can be seen in figure 7. At time $t=10s$ the frequency steps from its nominal value 50Hz to 50.2Hz. Due to the rapid change, there is a small overshoot in the PLL frequency measurement. When the frequency rises, the active power output drops as in figure 5. Since for $t>10s$ the frequency deviation $\Delta f$ is constant, the steady-state output of the bandpass discussed in the previous section is zero. Thus the active power output settles back to its steady-state value and it is verified that the proposed control method does not provide primary control.
Fig. 6: Turbine Speed

Fig. 7: Active Power and Grid Frequency for a Step in Frequency
5 CONCLUSION
Inter-area oscillations in the European power system are an issue for system instability. The proposed control method adds a frequency proportional part to the active power feed-in of a wind power plant in order to increase power system damping. A simulation model of a 1.7MW wind power plant has been built and tested with a 0.25Hz frequency oscillation. It is shown that the active power output of the WPP comprises of a constant part and a part proportional to the grid frequency. Therefore inter-area oscillation damping is possible with this control method. The influence of the fluctuating power output on turbine speed is negligible. It has to be further investigated, what is the minimum gain MW/Hz for a WPP to obtain a noticeable effect on power system damping and what is the maximum gain in order to not disturb the normal operation of the WPP. The proposed power system damping control does not emulate a virtual inertia and does not supply primary control power which is shown in simulation. Wind power plants are suitable for the proposed control extension due to their great impact on the overall electric energy generation. But there is a need of financial or regulatory stimulation for wind power plant manufacturer to implement this feature.

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OPTIMISATION OF THE POWER TRANSMISSION CAPABILITY OF SUBSEA CABLES FOR THE GRID CONNECTION OF OFFSHORE WIND FARMS

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Keywords: HVDC, subsea cables, 2K-criterion, optimisation, evolutionary strategy

Summary: This document provides a solution to optimise the power transmission through a subsea cable by use of its thermal reserves by regarding the 2K-criterion.

1 INTRODUCTION

In Germany, TenneT is responsible for connecting offshore wind farms (OWF) in the German North Sea to the onshore transmission grid. Small wind farms which are near to the shore are connected by HVAC connections. For OWFs with a larger amount of installed power and a greater distance from the coast HVDC systems are used.

Special permit requirements for the offshore cables do exist when crossing the nature reserve Wadden Sea. One important restriction is the 2K-criterion. It specifies that the temperature of the sea ground must not be heated up by more than 2 K compared to the sea ground which is not influenced by the cable [1]. Hereby the fluctuating power infeed of the offshore wind farms connected to the cable, as well as in case of AC cables the reactive power compensation have to be taken into account. Hence, the emitted thermal energy and therefore the transmission capacity are limited below the technical limit of the cables, which can be found in cable data sheets.

As an important damage reduction concept for outages of grid connection systems (GCS) and interim solutions in case of delays in the project realisation, TenneT is planning to provide transmission reserves by transmitting a part of the failed power by using parallel GCS. Therefore, a method has to be developed to guarantee that the 2K-criterion is fulfilled and simultaneously transmit as much power to the shore as possible. As only offshore wind farms are connected to the HVDC grid connection system the load of the system is fluctuating due to the available wind. Further the thermal system (cable and surrounding soil) has a very large time constant of approx. 3 years (own calculation based on [3, 4]). Therefore, conventional controllers (e.g. PI-controller) cannot be used for the control of the cable current. Instead of this, TenneT developed a predictive feed forward approach based on an optimisation algorithm taking into account the 2K-criterion as a constraint.

With this optimisation method it is possible to transmit more energy by using time periods with low wind conditions. After all, the outage costs are minimised.
2 DESIGN OF GRID CONNECTION SYSTEMS

a. Overview

Figure 1 shows TenneTs standardised HVDC grid connection system (GCS). The method described below for the optimisation of the power transmission capability of subsea cables could be applied for the HVAC cables between the grid connection points of the OWF and the point of common coupling (PCC) as well as for the HVDC cables between the offshore and the onshore converter stations.

Fig. 1 Standardised HVDC grid connection system (GCS)

b. HVDC converter

For the standardised GCS HVDC converters in VSC technology are used. This type of converter consists of numerous switchable semiconductor valves, usually Insulated Gate Bipolar Transistors (IGBT), connected in series.

Dependent of the transmission power, the IGBTs will be heated up, due to the losses caused by the electric current. Therefore, cooling is very important to ensure that the semiconductors will not be damaged. The dimensioning of the IGBTs and the cooling system is done taking into account the rated transmission power of the HVDC and the reactive power required by the grid connection rules. Because of legal requirements in Germany the transmission power of the HVDC is equal to the total rated generator power of the wind turbines connected to the GCS.

Consequently the converter is an important element limiting the possible transmission power of the GCS by:

1. Maximum admissible current through the IGBTs
2. Maximum voltage in the DC link which is limited by the insulation of the cables and therefore no control parameter to optimise the transmission capacity

Because of the very small thermal capacity of the IGBT the HVDC converters are tripped immediately by the protection system in case of over currents to avoid damages. Overloading
of the HVDC as it could be necessary for operational switching or an interim connection of an OWF is therefore not possible.

To reach an undisturbed operation of the GCS in all operational situations it has to be ensured that the maximum transmission power is not exceeded at any time. The rated current through the IGBT valves is therefore an important constraint for the optimisation algorithm in cases in which it is used to optimise the power transmission of DC cables.

c. Subsea cables
All AC and DC subsea cables in the German North Sea are designed to fulfil the so called 2K-criterion which is part of the permission of the Bundesamt für Seeschiffahrt und Hydrografie (BSH) [1]. The 2K-criterion defines that the soil in a reference point 20 cm under the sea ground directly above the cable must not be heated up more than 2 Kelvin compared to the undisturbed soil.

The calculation of the temperature difference has to be done according to the calculation method described in IEC 60287 [3]. The first step is to calculate the mean value of the temperature difference by using the mean value of the cable losses. In doing this the BSH requires that the fluctuating infeed of the OWF, as well as the concrete design of the GCS (including position and dimensioning of compensation coils) are considered. After this in a second step a transient temperature rise is superposed to the mean value of the temperature difference in the reference point to represent full load phases of the wind farms. Hereby the calculation method described in IEC 60853-2 [4] is used, taking into account a full load phase of seven days followed by a cooling phase of 45 days with mean cable losses.

Because of its large thermal time constant the thermal loading capacity of the cables calculated with this method is not equal to the continuous rated current which could be found in cable data sheets. In contrary to the method mentioned above, the continuous rated current is determined by the maximum admissible temperature of the conductor, which is 90°C for HVAC and 70°C for HVDC XLPE cables.

In consequence technical reserves are available which could not be used for temporary overloading due to permission requirements without methods to ensure that the 2K-criterion is not violated. Essential for developing these methods of transmission capacity management (TCM) is to take into account the thermal preloading of the cable. This means that a higher loading of the cables could be possible after low wind condition without violating the 2K-criterion, whereas after high wind conditions a limitation of the cable current could be necessary.

3 TRANSMISSION CAPACITY MANAGEMENT I
Operational switching and interim connections of OWF at the PCC located on the converter platform could cause operating cases in which the maximum grid connection capacity is exceeded. To prevent this, TCM will be used by the operation management.

As a first step TCM is implemented with the TCM I procedure. The strategy for the grid control with TCM I is an operational management of the number of turbines connected to the GCS. The available transmission capacity of the GCS is dedicated to the regular connected
OWF with first priority. If the regular connected OWF releases parts of its dedicated transmission capacity (e.g. due to planned maintenance of the OWF), the remaining free transmission capacity will be made available for the interim connections.

In consequence of the above described process the TCM I procedure utilises only the reserves which are resulting of the limited availability of the wind turbines (approx. 95 %) connected to the cable. Therefore, TCM I allows only a marginal increase of the energy transmitted to shore.

4 Transmission Capacity Management II

In contrast to the TCM I procedure which is an operational method to ensure that only as much wind turbines are connected to a cable as it is designed for, the TCM II procedure optimises the current through the cable automatically by taking into account the thermal limitations of the cables as well as of the IGBT valves in the converters.

Due to the large thermal time constant of the cable system and the surrounding soil, which is in the range of approximately three years at least, a classic feedback control would not be productive. It would take too long until a stable temperature is reached. Thus, a feed forward approach based on an optimisation algorithm is used. In this case an evolutionary strategy algorithm was applied [5].

Additional to the optimisation algorithm itself the developed approach needs an aerodynamic and electrical model of the wind farms which provides the future cable currents resulting of a wind prognosis for the next days [6]. Further a database with measured cable currents has to be provided by the grid control to be able to calculate the expected time series of the temperature difference at the reference point.

A complete flow chart of the TCM II procedure is shown in figure 2.

![Flow chart of the TCM II procedure](image)

Fig. 2 Flow chart of the TCM II procedure

d. Transient calculation of temperature rise

A very important part of the TCM II procedure is the thermal cable model. In principle two methods for the calculation of the transient heating are available:
1. Analytic calculation methods according to [3] and [4]
2. Finite elements methods

Both calculation methods provide equal solutions in case of homogenous soil and only a few, equally loaded cables. Only in case of complicated laying conditions the more complex finite element methods must be used.

The calculation of the transient temperature rise is calculated acc. to IEC 60853-2 [4] and the cable losses are calculated acc. to IEC 60287-1-1 [3]. Due to the fact that only the temperature rise at 20 cm below surface or 30 cm below surface, depending on the region, must be calculated, some simplifications can be made to speed up calculation and to minimise the influence of environmental effects.

As there is no need to know the actual temperature of the ground when calculating only the temperature rise, the basic simplification is to assume that the cable core has its maximum operation temperature all the time. This gives slightly higher losses but it avoids the calculation of the actual cable conductor temperature, a calculation that is afflicted with several unknowns, such as: the actual temperature of the ground, the thermal resistivity and diffusivity. These variables vary permanently and it is impossible to feed them in an algorithm on a daily basis.

As described above, there are two types of cables used by TenneT: AC-cables and DC-cables. The cable losses of AC-cables are calculated acc. to IEC 60287-1-1 [3], clause 1 and 2:

The electrical losses of the conductor: \[ W_l(t_i) = I(t_i)^2 R'(1 + y_s + y_p) \]
With: \[ R' = R_0(1 + \alpha_0(\theta - 20)) \]

The inductive screen losses: \[ W_s(t_i) = I(t_i)^2 R' \lambda_1 \]

The inductive losses of armor: \[ W_B(t_i) = I(t_i)^2 R' \lambda_2 \]

The dielectric losses: \[ W_d = \omega C U_0^2 \tan\delta \]

For AC-cables the maximum operation temperature of \( \theta = 90^\circ C \) (max. conductor temperature) is assumed, \( I(t_i) \) is the applied current at time \( t_i \).

The constants \( y_s, y_p, \lambda_1, \lambda_2, C, \tan\delta, R_0 \) are given by the manufacturer of the cable. The total loss of a three core AC-cable at each time step \( t_i \) is given by:

\[ W_i = 3(W_l(t_i) + W_s(t_i) + W_B(t_i) + W_d) \]

For DC-cables the loss-calculation can be reduced to

\[ W_i = 2I(t_i)^2 R_0(1 + \alpha_0(\theta - 20)), \theta = 70^\circ C. \]

The transient calculation of the temperature rise for each time step \( t_n \) acc. to IEC 60853-2 [4]

\[ \Delta \theta(t_n) = \sum_{i=1}^{i=n} \frac{\rho_T}{4\pi} W_i \left[ -Ei \left( \frac{-(d_{pk})^2}{4\Delta t_i \delta} \right) - \left[ -Ei \left( \frac{-(d'_{pk})^2}{4\Delta t_i \delta} \right) \right] \right] = \sum_{i=1}^{i=n} W_i \Delta X_i \]

with:

\[ \Delta t_i = t_n - t_i \]
\[ \rho_T \quad \text{Thermal resistance of soil} = 0.7 \text{Km/W or 1.0 Km/W depending} \]
-Ei Exponential integral
d_{pk} Distance of cable to point x
d'_{pk} Distance of cable to reflection point x' (compare IEC 60287-2-1, Figure 1)
δ Diffusion coefficient. An average value that holds for most conditions = 7.0 e⁻⁷ m²/s.

e. Description of the optimisation algorithm
The aim of the TCM II procedure is, to transmit as much power as possible without violation of the 2K-criterion. This means mathematically, to find a current profile with which the areas between the resulting heating function for the reference point and the 2K-line become minimal, as shown in figure 3.

![Diagram showing heating function and optimisation goal](image)

**Optimisation goal:**
\[ \min(T_1 + T_2 + T_3) \]

**Fig. 3 Heating function and optimisation goal**
Because the current through the cable is not independent of the wind farm power, which itself is a result of the wind profile at the site, it could not be used as direct input of the optimisation algorithm. Instead a reduction factor \( K(t) \) is used, which reduces the wind farm power in each time step:

\[ I_{opt}(t) = K(t) \cdot P(t) \]
The optimisation problem described above could not be solved with classic linear or nonlinear optimisation methods. Due to the fact that it is not necessary to find the global optimum exactly, it seems appropriate to use a so called metaheuristic optimisation approach which is a family of methods, which are able to find the solutions of any user-defined problem by approximation. The further requirements on the optimisation algorithm are, that it is able to find a proper solution in an acceptable time and is flexible enough to allow later model extensions.

An optimisation method which fulfils all these requirements and is therefore used in this concept is the so called evolutionary strategy [5] which is a branch of evolutionary algorithms. Evolutionary algorithms in general are stochastic, metaheuristic optimisation approaches and in contrast to the classic exact methods, they therefore do not guarantee to find the exact global optimum.

The basic idea of this type of optimisation algorithm is that a set of possible solution candidates to the optimisation problem is optimised using the Darwinian principles of the biological evolution. Typical mechanisms are reproduction, mutation and selection (survival of the fittest). According to the biological evolution the solution candidates can be seen as individuals in a population. The so-called fitness function (given by the optimisation problem) determines the environment within which the solutions have to survive. Evolution of the population takes place after the repeated application of the above operators.

The optimisation is done by using measurement values of the transmitted energy from the past and a wind forecast for the next 1-2 weeks. The wind forecast is used to calculate the possible generated energy. This calculation requires the geographical position and equipment data of every wind turbine and data of the offshore AC-grid of the wind farm. By using a wake model [7] in combination with a power flow calculation the potential generated power and the losses inside the wind farm are calculated. Taking the forecast as a basis of the generated energy, the
change of the temperature of the sea ground is calculated. If the change of the temperature will be above 2 K, a reduction factor is needed to keep the temperature below 2 K. During the optimisation the reduction factor $K(t)$ is varied, so that the transmitted energy is at maximum and the change of the temperature at every time below 2 K. The chart of the procedure is shown in Figure 1.

5 FIRST RESULTS

The following figures 5 and 6 show the results of a calculation scenario in which a failure of one of two cables connecting a wind farm to the offshore converter is assumed. Hence, the generated power of the wind farm has to be transferred over the residual cable which is consequently overloaded by twice of its designed power. This operational switching scenario is analysed for one year, comparing the temperature differences, cable currents and transmitted energy without any limitation and with the TCM I and II procedures.

Without any limitation the cable current rises to a peak value of approx. 158 % of the rated thermal current. Thus, the temperature difference at the reference point reaches a maximum of about 3.5 K which is 1.5 K higher than the allowed 2 K. Independent of the 2K-criterion this would not be a relevant switching scenario in practise, because the cable would be switched off by the grid protection system due to the violation of the continuous rated cable current.

In contrary with the TCM I procedure which limits the generator power connected to the cable to 50 % of the installed wind farm power, the cable current is lower than approx. 79 % of the rated thermal cable current. This shows, taking into account the scenario being equal to the design case, that the conductor cross section of the subsea cables is from a technical view overdimensioned in order to ensure that the 2K-criterion is fulfilled. The temperature difference in this scenario is 1.94 K in maximum.

Using the thermal reserves resulting from the overdimensioning of the cables the TCM II optimisation algorithm ensures a maximum energy transfer without violation of the 2K-criterion or the continuous rated cable current. As a consequence of this within the considered year approx. 22 % more energy is transmitted to shore than with TCM I.

![Fig. 5 Cable currents without limitation and with TCM I and II](image)
6 CONCLUSION

In this paper, a solution is presented to use the thermal reserves of an offshore cable by regarding the 2K-criterion. To maximise the transmitted energy, a feedback control is not productive as the time constant of the system is too long. Hence, a feed forward approach based on a metaheuristic optimisation algorithm was used.

In an example it is shown, by the using of the presented optimisation procedure it is possible to transmit 22 % more energy compared to a cable system for which the transmittable power is limited to the designed value based on a standard wind profile. Thus, in case of outages or transmission limitations caused by failures or maintenance campaigns of GCS the connected offshore wind farms could partially generate electric power.

7 REFERENCES

REPLACING CONVENTIONAL STEAM POWER PLANTS BY WIND POWER STATIONS
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Keywords: Conventional Power Plant Substitution, Fictitious Synchronous Generator, Steam Turbine Control, Primary Balancing, Instantaneous Reserve, Micro-Grid

Summary: An upgrade of the count of renewable energy resources is forwarded all the time. But the critical aspect isn’t only the count of installed wind parks; here the behavior of each wind power station related to the power grid is important, too. Today the conventional steam power plants are controlling the power grid and realize the fundamental necessary instantaneous reserve by their rotor mass. But if the count of conventional plants decreases more and more their functionalities have to be replaced e.g. by the wind power stations. To serve the purpose a new control method is proposed and explored in a simulation. Here a fully-fed converter based wind power station is used and its complete behavior is changed to a conventional steam power plant. Then primary balancing over droops can be easily realized not otherwise than an instantaneous reserve. There is a fundamental change to conventional control concepts: The control of the dc-link voltage is no longer controlled by the grid-tied inverter, here it is controlled by using the physical generator that is coupled to the rotor in combination with the generator-tied inverter. The behavior is related to a steam turbine. In analogy to the main steam valve the pitch angle is used as the actuator. The power output of the wind can be controlled to keep a primary reserve all the time as demanded by the Entso-E. The grid-tied inverter is controlled by using a fictitious electrical excited synchronous generator with amortisseur. It realizes a dynamic behavior for the controlled grid currents that is equal to a real power plant generator. For protecting the power electronics here it is necessary to implement a limitation for the current outputs of the generator model.

1 INTRODUCTION
The energy turnaround is one of the biggest projects in the 21th century. In that context one important corner pillar is the wind energy technique that leads to new wind parks onshore and offshore at the sea. If the fossil fuel based power plants shall be replaced by ecological preferred wind power stations it is clear that the functionalities for grid control have to move. For a stable power grid it is important to guarantee a quantity of control reserves. An instantaneous reserve and a primary reserve are needed to get a balance between electrical energy generation and consumption all the time. At this point a change in the control concept of wind power stations is proposed to guarantee a stable electricity supply in the future. [1]
For a wind power station it is obvious that the rotary mass of the wind rotor can be used as an energy storage system for the realization of the instantaneous reserve. The wind rotor can be slowed down like a set of a turbine and a generator in a conventional steam power plant during operation. To get an idea for a degree of the instantaneous reserve that has to be decoupled during a frequency disturbance there is an overview in Table 1 about different power plants in the UCTE-grid in comparison to a wind power station with a nominal power of about 5 MW. It is interesting that many energy resources have nearly the same ratio between their decoupled energy out of the rotary mass and their nominal power ($\Delta E_{1Hz}/P_{nom}$) about a value of 200 ms during a frequency step of e.g. -1 Hz. Here the time constant for the UCTE-grid is used to calculate the energy that has to be stored in the rotor of the wind power station as an instantaneous reserve ($\Delta E_{30s}$). As one result only 0.267 kWh are needed for a wind power station with a nominal power of about 5 MW.

Table 1: Comparison of the power reserve ability for different types of power generation [2]

<table>
<thead>
<tr>
<th>Type of generation</th>
<th>$P_{nom}$ (MW)</th>
<th>$E_{total}$ (MWh)</th>
<th>$\Delta E_{1Hz}/P_{nom}$ (ms)</th>
<th>$\Delta E_{30s}$ (kWh)</th>
<th>$\Delta P_{15min}/P_{nom}$ (%)</th>
<th>$\Delta P_{15min}$ (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grid of the UCTE</td>
<td>300,000</td>
<td>400</td>
<td>192</td>
<td>16,000</td>
<td>1.5 – 10</td>
<td>4,500 – 30,000</td>
</tr>
<tr>
<td>Nuclear p. p.</td>
<td>1,300</td>
<td>2.0</td>
<td>222</td>
<td>80</td>
<td>1.5 – 10</td>
<td>19.5 – 130</td>
</tr>
<tr>
<td>Coal-fired p. p.</td>
<td>700</td>
<td>0.8</td>
<td>165</td>
<td>32</td>
<td>1.5 – 10</td>
<td>10.5 – 70</td>
</tr>
<tr>
<td>Wind power station</td>
<td>5</td>
<td>-</td>
<td>192</td>
<td>0.267</td>
<td>1.5 – 10</td>
<td>0.075 – 0.5</td>
</tr>
</tbody>
</table>

The normalized decoupled kinetic energy ($\Delta E_{30s}/P_{nom}$) during a settling of 5 and 10 % of the rotation speed $n_{WEA}$ is shown in Figure 1. It is the drive-train of an Areva M5000 wind power station. The rotation speed is adjusted to an optimal power factor $c_p$ independent of the wind speed $v_{wind}$. For the compare a coal-fired power plant of 700 MW is used. As you can see low wind speed is being enough to realize an equal behavior for the instantaneous reserve as like it can be used by the rotary mass of a conventional steam power plant. [3]

On the other hand there has to be a power reserve of about 1.5 to 10 % related to the Entso-E ($\Delta P_{15min}/P_{nom}$) that has to be kept for supporting the power grid for a time period of about 15 minutes. For each wind power station it means that the power feed during normal operation has to be reduced. And this means if the grid frequency is decreasing a plant of 5 MW has to feed additional 75 to 500 kW as primary power reserve.

The new control concept changes the functionality of the components of a conventional fully-fed converter based wind power station to a steam turbine and an electrical excited synchronous generator with amortisseur. It is a change of the whole behavior of the wind power station. Then it is possible to realize an instantaneous reserve, a primary balancing control and other power plant features that are available in conventional plants. There is also a change in the management from maximum power feed to a reserve based control concept that might be necessary if the count of renewable plants is rising higher and higher.

Figure 1: Mechanical energy stored in the rotor
2 THEORETICAL OVERVIEW

The base components of a steam power plant are the heating circuit including the boiler, the main steam valve and the steam turbine. The steam turbine drives a synchronous generator that is connected to the power grid. The main steam valve adjusts the acceleration torque that influences directly the feed of active power. In a second control loop the feed of reactive power can be controlled by the excitation voltage of the generator. In Figure 2 there is a structural overview and a compare between a fully-fed converter based wind power station (a) and a simplified steam power plant (b). Here are several analogies that can be confronted: The pitch angle can be used to control the power output of the wind related to the main steam valve. The rotor in combination with the generator and its inverter is used as a like as a steam turbine in a conventional power plant that controls the rotation speed of the shaft. Here it is the level of the dc-link voltage that has to be controlled. Both are only able to transport active power. And finally the grid-tied inverter gets the behavior of an electrical excited synchronous generator with amortisseur. All analogies are shown in Table 2 again and therefore a new control concept was developed.

Table 2: Analogies between the components of steam power plant and a wind power station

<table>
<thead>
<tr>
<th>Conventional steam power plant</th>
<th>Wind power station with a synchronous generator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam</td>
<td>Wind</td>
</tr>
<tr>
<td>Main steam valve</td>
<td>Pitch actuator</td>
</tr>
<tr>
<td>Turbine</td>
<td>Rotor, generator and inverter</td>
</tr>
<tr>
<td>Shaft</td>
<td>Dc-link</td>
</tr>
<tr>
<td>Synchronous generator</td>
<td>Grid-tied inverter with a fictitious generator</td>
</tr>
</tbody>
</table>

In the concept the feed of active and reactive power to the grid is automatically adjusted in relation to the implemented droops. Now with the new structure it is possible that the wind power station can support the power grid in the same way as each conventional steam power plant can do. In this case it is possible to guarantee a stable power distribution system during the energy turnaround.

There is an overview of the implemented control concept in Figure 4. As you can see the command value for the active power, here the acceleration torque $\tau_{\text{fictitious}}$ of the fictitious generator is used as the command value $P_{\text{mech,cmd}}$ for the power controller in the turbine control structure. The actuation value is the pitch angle of the rotor blades $\beta_{\text{cmd}}$. The dominant delay in this control loop is the dynamic of the drives for the rotor blades. In a second control loop the dc-link voltage is lifted up to a fixed level by using the generator-tied inverter.
At the opposite direction of the dc-link there is the grid-tied inverter. Here a fictitious electrical excited synchronous generator with amortisseur is used for controlling the grid currents. By using this mathematical model the grid currents get the behavior of a real power plant generator. And there is a second control loop in the power plant structure. It uses the command value for the fictitious excitation voltage $v_f$ to control the feed of reactive power. [4]

Between the wind power station and the steam power plant there is one big difference: The rotation speed of the physical generator of the wind power station is not coupled to the grid frequency. This difference is used in Figure 3 to control the active power by adjusting the pitch angle. The operating point OP2 is used instead of OP1 to drive the plant in a stable area of operation. Here it is the so called spinning reserve what is in that case the instantaneous reserve. [5]

A test stand is constructed for experimental studies. Its structure is shown in Figure 5. An induction motor is used for a wind rotor emulation that drives a permanent magnet excited synchronous generator. It is representing the generator in a wind power station with a nominal power of about 30 kW. For the wind rotor emulation an aero dynamical model is calculated online in Labview. It controls only the torque of the induction motor but measures the actual rotation speed of the drive train that influences the operating point in the characteristic curve. Additionally it has to read the command value for the pitch angle of the rotor blades $\beta_{cmd}$ out of the turbine control. Implemented wind profiles are used for emulating a realistic scenario.
3 SIMULATION RESULTS

For the realization of the fictitious synchronous generator a fast current control loop is used where the outputs of the generator model are the command values. They are compared with the measured grid currents. Its parameters are shown in Table 3. With the implemented amortisseur it is able to run up to the nominal rotational speed if the excitation voltage of the fictitious generator is zero and its inputs are connected to the grid voltages. Then the feed of reactive current is controlled to zero by using a reactive power controller. The polar wheel is then automatically synchronizing to the angle of the grid voltages. [6], [7]

In analogy to a real set of a turbine and a generator the input for the acceleration torque is used to control the fictitious rotational speed. In this point of view the electric torque of the fictitious generator is a disturbance. One important aspect is that the currents have to be limited to a maximum value that is allowed for the inverter. Here a new method is developed that changes the parameters inside the generator model. In that case it is strictly necessary to limit the feed of active power that means a limitation of the acceleration torque. Then the functionality of the amortisseur is available all the time. The current course for a frequency disturbance and a time limited symmetrical voltage drop is shown in Figure 6 and 7. The dashed line shows the currents without any limitation. With the implemented limitation the output currents can be successfully limited to a value of about 2.0 kA. This calibration allows an over-current of about 33 %. The active and reactive power feed to the grid is limited, too. The dynamic behavior is shortly modified but the fundamental functionality keeps working. With the explained method the over-dimensioning of the power electronics can be limited.

Table 3: Parameters of the fictitious synchronous generator

<table>
<thead>
<tr>
<th>type</th>
<th>fictitious synchronous generator</th>
</tr>
</thead>
<tbody>
<tr>
<td>nominal power</td>
<td>5 MW</td>
</tr>
<tr>
<td>nominal rotation speed</td>
<td>3000 rpm</td>
</tr>
<tr>
<td>pole pairs</td>
<td>1 (non-salient pole machine)</td>
</tr>
<tr>
<td>moment of inertia</td>
<td>416.8 kgm²</td>
</tr>
<tr>
<td>terminal voltages</td>
<td>900 V (star connection)</td>
</tr>
<tr>
<td>phase currents</td>
<td>1.85 kA</td>
</tr>
<tr>
<td>excitation</td>
<td>electrical</td>
</tr>
<tr>
<td>excitation voltage</td>
<td>39 V (idle)</td>
</tr>
<tr>
<td>amortisseur</td>
<td>implemented</td>
</tr>
</tbody>
</table>
In an over-all simulation the fictitious generator and the turbine control are tested together. Ramps with high decrease rates are used to change the grid frequency and the voltages. Here the wind speed is constant of about 14 m/s for only having an influence by the power grid.

Within the first 30 s of the frequency disturbance in Figure 8a the primary balancing reserve has to be activated by using the reserve range of the pitch angle in Figure 8e. Only in this way the power grid can be stabilized for the next 15 min after the decoupled instantaneous reserve is consumed. To reach this behavior the fictitious acceleration torque is lifted up for the fictitious generator in Figure 8b. That affects directly an increase of the active power feed to the grid in Figure 8c. There the power feed lifts up from 5 to 5.5 MW during the disturbance. As shown in Figure 8d the turbine control achieves its task to control the dc-link voltage successfully. There are only transient voltages about ±4 % of about the nominal value during the step related changes in the grid frequency.

In the last scenario in Figure 9a the grid frequency is constant and instead of it the voltages are decreased by 5 % of its nominal value. In that case the fictitious excitation voltage in Figure 9b is lifted up to change the reactive power feed to the grid in Figure 9c. Here the grid is supplied with 2.5 Mvar during the voltage fault. The pitch angle in Figure 9e does not change because active power feed is not affected.

4 EXPERIMENTAL RESULTS

The configuration of the test stand is illustrated in Figure 5. In a first step the generator-tied dc-link voltage control is tested under heavy conditions. Here a resistor of about 80 Ω is connected to the dc-link that can be switched on and off. The task of the voltage control loop is to achieve a dc-link voltage of about 650 V all the time. In Figure 10 and 11 it takes about 400 ms after the load of 5 kW is connected or disconnected to get back the nominal value by controlling the transverse component of the generator currents.
Figure 8: Variation of the grid frequency

- Frequency -0.5 Hz in 0.5 ms for 30 s
- Fictitious acceleration torque
- Active power feed to the grid
- Dc-link voltage
- Pitch angle of the rotor blades

Figure 9: Variation of the grid voltages

- Voltages -45 V in 18 ms for 10 s
- Fictitious excitation voltage
- Reactive power feed to the grid
- Dc-link voltage
- Pitch angle of the rotor blades

Figure 10: Ohmic-load switched on (5 kW)

- Dc-link voltage controlled to 650 V
- Power during resistor-load
- Controlled generator currents

Figure 11: Ohmic-load switched off (5 kW)

- Dc-link voltage controlled to 650 V
- Power during resistor-load
- Controlled generator currents
CONCLUSION

A new control concept for wind power stations is illustrated that changes its complete behavior to a steam power plant with a set of a turbine and a generator. In this case they are able to replace the conventional steam power plants.

The simulation results show that the change of the complete behavior is successfully done. During grid disturbances the supervisory power plant control changes the feed of active and reactive power. This is done by adjusting the acceleration torque and the excitation voltage at a fictitious electrical excited synchronous generator with amortisseur that is connected to the grid voltages. The control scheme is the same as in a conventional steam power plant with a rotational speed and a voltage control loop. Here the implemented droops are regarded to the conditions of the network carriers. But the current outputs of the fictitious generator have to be limited to protect the power electronic components. This is successfully done by a new method that changes the parameters of the fictitious generator during operation. In first experimental explorations the generator-tied dc-link voltage control is tested. During an ohmic-load change the dc-link voltage control fulfils the required task and controls the dc-link voltage to its nominal value.

6 SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Name</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{cmd}$</td>
<td>Command value for pitch angle</td>
<td>°</td>
</tr>
<tr>
<td>$I_{gen}$</td>
<td>Currents of the generator</td>
<td>A</td>
</tr>
<tr>
<td>$I_{grid}$</td>
<td>Grid currents</td>
<td>A</td>
</tr>
<tr>
<td>$\omega_{gen}$</td>
<td>Rotational speed of the PMSG</td>
<td>1/s</td>
</tr>
<tr>
<td>$\omega_{st}$</td>
<td>Rotational speed of the SG</td>
<td>1/s</td>
</tr>
<tr>
<td>$\omega_{tu}$</td>
<td>Rotational speed of the steam turbine</td>
<td>1/s</td>
</tr>
<tr>
<td>$P_{el}$</td>
<td>Active power (electrical)</td>
<td>W</td>
</tr>
<tr>
<td>$P_{grid,cmd}$</td>
<td>Command value for active power</td>
<td>W</td>
</tr>
<tr>
<td>$P_{mech,cmd}$</td>
<td>Command value for mech. power</td>
<td>W</td>
</tr>
<tr>
<td>$Q_{ele}$</td>
<td>Power reserve for primary balancing</td>
<td>W</td>
</tr>
<tr>
<td>$Q_{acc,cmd}$</td>
<td>Command value for accel. torque</td>
<td>Nm</td>
</tr>
</tbody>
</table>

7 REFERENCES

Aspects of Wind Turbine Converter related Research at IAL, University of Hannover, Germany

F. Fuchs; A. Mertens

Institute for Drive Systems and Power Electronics
Leibniz Universität Hannover, Germany

09-03-2014

Presentation at IWEC in Hannover, Germany
Overview on Presentation

- Introduction – Overview on Topics

- Example 1: Lifetime of DFIG Windturbine Converters
  - Results
  - Outlook: Fraunhofer Innovation Cluster

- Example 2: Wind Turbine & Grid Impedance Resonance
Introduction – Overview on Topics

GeCoLab Hannover Wind Turbine Research with scale of 1:10

Wind Turbine Research at IAL

Multilevel Converter Concepts for High Power Turbines (Hexverter)

LVRT Control of slow switching converters

Power Control in Microgrid

Grid Current Control and Resonances

Wind Turbine Converter lifetime

New Materials for Wind Turbine Power Electronics

Generator failure prediction methods

Generator design and validation

GeCoLab Hannover Wind Turbine Research with scale of 1:10

Multilevel Converter Concepts for High Power Turbines (Hexverter)

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Power Control in Microgrid

Grid Current Control and Resonances

Wind Turbine Converter lifetime

New Materials for Wind Turbine Power Electronics

Generator failure prediction methods

Generator design and validation
Introduction – Overview on Topics

- High Power Test Bench (scale 1:10)

- GeCoLab (Generator Converter Laboratory) in Hannover going in operation at the end of 2014: 1.2 MW converter generator test bench including a 690 V grid emulator.
Introduction – Overview on Topics

- Multilevel Converters for Grid Applications
  - Modular Topologies of Multilevel Converters: Analysis and evaluation
  - Interior Control of Multilevel Converters (DC link voltages, output currents)
  - Realisation of Modular Multilevel Converters for Experimental Evaluation

Laboratory Setup (690 V / ca. 100 kVA) for Modular Multilevel Topologies

HEXVERTER, a novel Modular Multilevel Converter for grid connection of wind turbines

Modular Multilevel Converter for Battery Energy Storage System
Example 1: Lifetime of DFIG Windturbine Converters
Prolonging Wind Turbine Converter lifetime

- The converter is a weak element in the wind turbine:

Prolonging Wind Turbine Converter lifetime

- One cause: power module damage due to chip temperature changes


Bond wire lift off

Solder Crack

source: senvion.com

source: wikpedia.org

source: woodward.com
Prolonging Wind Turbine Converter lifetime

- Analysis performed for DFIG Wind Turbine System
  - Low fundamental output frequencies (0 - ca. 15 Hz) of rotor side converter are lifetime critical
  - Several actions can prolong the converter lifetime:
    - Higher dimensioning
    - Minimize Temperature swing through
      - Adapt switching frequency / PWM method
      - Avoid critical rotational speeds
      - Intelligent reactive current distribution between stator and rotor
Prolonging Wind Turbine Converter lifetime

Results

• 2 MW turbine; one-year wind profile from German North Sea
• Steady state analysis
• Analysis will be intensified in new Fraunhofer Excellence Cluster
Prolonging Wind Turbine Converter lifetime

- Fraunhofer Innovation Cluster starting 2014
  - Failure analysis of power electronic components
  - Better understanding of the causes of failures by linking them with field measurements (for example: error messages, air humidity, lightning data etc.)
  - Detailed modelling of the complete wind turbine
Example 2: Wind Turbine & Grid Impedance Resonance
Wind Turbine & Grid Impedance Resonance

- Common Approach to model grid impedance for converter issues:
  - inductive

![Diagram of wind turbine and grid impedance resonance]

Graph showing impedance vs. frequency

Institute for Drive Systems and Power Electronics
Leibniz Universität Hannover
Felix Fuchs
Wind Turbine& Grid Impedance Resonance

- not in every case a good approximation:
  - Grid impedance seen by the converter is time varying and not only inductive

- Resonances - within Control Bandwidth!

Large Industrial Distribution system
Case 1: all loads, sources and PFCC are operating at nominal value
Case 2: Some loads, sources and PFCC are not operating
Case 3: Additional 30 MVA and 18 MW converter load connected by long cable
Wind Turbine & Grid Impedance Resonance

- Analysis by control theory
- A better mathematical description is the basis for better control design
Active current incl. transfer function prediction

Reactive current

Next step: Stability analysis with resonances -> will be presented at ECCE 2014
Conclusion

- Short Overview on Wind Turbine Research at IAL, University Hannover
  - Hexverter
  - GeCoLab

- Example 1: Lifetime of wind turbine converters
  - Results for 2 MW DFIG Turbine basis for Fraunhofer Innovation Cluster

- Example 2: Wind turbines and resonances in the grid impedance
  - Influence is analysed analytically and validated by experiment
Thank you for your attention!

F. Fuchs; A. Mertens

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Germany

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INFLUENCE OF THE WELDING PROCESS ON THE FATIGUE RESISTANCE OF THICK STEEL PLATES

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Keywords: Fatigue design, welded joints of higher plate thickness, geometrical notch effect

Summary: The multi-wire submerged arc welding process (SAW) is conventionally used for long welds with large cross sections which are characteristic for the manufacturing of tubular steel towers or monopiles. In this paper analytical and experimental investigations are discussed for predicting fatigue life of welds which are based on the local weld parameters. The fatigue resistance of butt welded connections with higher plate thicknesses and in particular the geometrical notch effect is in the focus of investigations.

1 INTRODUCTION

The dimensions and weight of components increase and the limits of feasibility for manufacturing and installation are reached. Beside cutting, forming and handling, this in special concerns the joining technologies. Welding is the mostly used technique to join steel plates and components and the development of new high performance welding techniques are necessary to join thick plates in faster and in fewer passes. This is also objective of two currently finished joint research projects at the Institute for Steel Construction namely OPTIWELD (funded by the BMU) and DOVOR (supported by the German Federation of Industrial Research Associations “Otto von Guericke” e.V. (AiF)). Within these projects researchers of mechanical, electrical and civil engineering cooperate with industrial partners on the improvement of welding techniques covering the development of the welding process and its effect on the metallurgical and mechanical properties. Extensive welding tests are accompanied by temperature measurements, metallographic analyses, as well as tensile, notched-bar impact and hardness tests. Furthermore, the resistance of butt welded connections under dynamic loading is demonstrated by constant amplitude fatigue testing. Beside normal strength also high strength fine-grained structural steels and here the effect of preheating are investigated. Additional details of the welding process development can be taken from [1], or [2].
2 Fatigue Tests

Within this paper the conventionally used multi-wire submerged arc welding process is in the focus of investigations being a uniform reference for the comparison to the alternative welding procedures. For this purpose welding tests on plates of steel grade S355J2+N with thicknesses of \( t = 20 \) and 30 mm were performed at the tower manufacturer. The welds are realised as a symmetrical double-V butt weld. Beside the extensive documentation of welding parameters, geometrical boundary conditions and manufacturing tolerances, temperatures were measured during these tests in order to get detailed information about thermal cycles. The presented welded joints pass quality level B according to DIN EN ISO 5817 on basis of visual and ultrasonic testing. In addition, the high weld quality is confirmed by tensile, notched-bar impact and hardness tests. Furthermore, samples were prepared for fatigue testing. Figure 1 shows the arrangement of fatigue test specimen in the weld sample and its dimensions.

![Figure 1: Arrangement of the test specimens in the weld sample (left) and specimen dimensions (right)](image)

Due to the limited number of test specimens, which means 15 samples for each S-N curve, the fatigue tests concentrate on the finite fatigue life. Constant amplitude tests with a stress ratio of \( R = 0.1 \) are carried out on three up to five load levels. The specimens are tested axially until total rupture. The fatigue tests are evaluated according to the Background Document of Eurocode 3 [3] and summarized in Table 1. The resulting S-N curve which is presented in Figure 6 on the left is basis for a comparison to the determined fatigue resistance. This structural detail has to be categorised as FAT 90 according to normative regulations. The fatigue resistance of \( \Delta\sigma_{C,95\%} = 149 \) MPa is considerably higher.

<table>
<thead>
<tr>
<th>Test series</th>
<th>variable slope ( m )</th>
<th>mean fatigue resistance ( \Delta\sigma_{C,50%} ) [MPa]</th>
<th>characteristic fatigue resistance ( \Delta\sigma_{C,95%} )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( m ) variable</td>
<td>( m = 3 ) [MPa]</td>
<td>( m = 5 ) [MPa]</td>
</tr>
<tr>
<td>#1 SAW, ( t = 20 ) mm</td>
<td>4.6</td>
<td>173</td>
<td>136</td>
</tr>
<tr>
<td>#2 SAW, ( t = 30 ) mm</td>
<td>4.5</td>
<td>175</td>
<td>152</td>
</tr>
<tr>
<td>#1 + #2</td>
<td>4.6</td>
<td>176</td>
<td>149</td>
</tr>
</tbody>
</table>
3 Fatigue Assessment by Local Approaches

In the following, the fatigue life is determined analytically by applying local fatigue approaches. In contrast to the nominal stress approach as part of the Eurocode 3 [4], where qualities of welded components are taken into account by generalized detail classes, local approaches consider actual qualities of the welded component. Properties influenced by welding which are of importance for the fatigue life can be taken into account individually. This concerns in particular the notch geometry and the strength distribution over the cross-section. The two essential phases of fatigue damage, the crack initiation and crack propagation, can be described by a combination of the notch strain and crack propagation approach, [5].

a. Combination of notch strain and crack propagation approach

Basically, it is assumed that total fatigue life $N_f$ of components consists of two major stages: crack initiation $N_i$ and crack propagation $N_p$. With regard to local effects at the fatigue critical detail, the number of cycles until a small crack with a depth of 0.5 mm and a width of 1.0 mm is initiated may be modeled by the notch strain approach, e.g. according to Seeger. This is presented in [6] and summarized in [5]. Three essential parts can be summed up to predict fatigue life until crack initiation.

1. Cyclic stress-strain behavior of the material
2. Cyclic stress-strain history at the fatigue critical location/ notch
3. Damage mechanism of the material

![Figure 2: Notch strain approach according to Seeger; scheme following [5]](image)

The elastic-plastic strain amplitudes at the failure critical location of the component are compared with the strain-life curve determined on a smooth test specimen under single-axis stress. This is based on the assumption that the local deformation, damage mechanism and crack initiation of the material at the notch root is the same as for the smooth, axially loaded and smaller test specimen, regarding its global deformation, global damage and total failure. In the absence of experimental data the Uniform Material Law [7] is used to generate the
needed input parameters to calculate fatigue life until crack initiation according to the scheme in Figure 2.

The subsequent phase of crack growth is modeled by using linear-elastic fracture mechanics (LEFM), which describes and analyses the relationship between stress, crack and fracture toughness. The basic principle is that the area around the crack tip affects mostly the fracture process. Principles of LEFM are used to relate the stress magnitude and distribution at the crack tip. Because the stresses are of infinitive height, they cannot be used to analyze crack propagation. Therefore a complex parameter is introduced to describe the stress state at the crack tip, being the stress intensity factor K. This factor depends on the geometry of component and the crack length. Under cyclic loading, the stress intensity factor will experience cyclic as well. Among numerous empirical or semi-empirical equations to relate fatigue crack growth rate and the stress intensity factor $\Delta K$, the Paris-Erdogan relationship is widely accepted and commonly used in practice for mode I cracks.

$$\frac{da}{dN} = C \cdot \Delta K^m$$

where $da/dN$ is the crack growth rate per cycle, $\Delta K$ is the range of stress intensity factor, and $m$ and $C$ are material dependent parameters. Approaches for calculating the stress intensity factor K are numerous and depend on the vicinity of the crack tip. Regarding the cracked surface of the failed samples of series #1 and #2 the stress intensity factor is assumed to be a quarter elliptical corner crack with a constant ratio of crack depth $a$ to width $c$ of 0.5 mm, [8].

b. Geometrical notch effect

In addition to the metallurgical discontinuities welds are geometrical discontinuities as well. The weld geometry is a disturbance of the stress distribution and leads to an increase of stresses and the formation of a multi-axial stress state at the weld toe. Under fatigue loading conditions the location of crack initiation is already preprogrammed. Thus, welds belong to the critical details of the fatigue assessment. When applying local fatigue approaches, the notch factor $K_t$ is an essential input parameter to consider the geometrical notch effect of components. This factor puts into ratio the maximum stress at the notch ground $\sigma_k$ and the linear disposed nominal stress $\sigma_n$.

$$K_t = \frac{\sigma_k}{\sigma_n}$$

To get information about the weld geometry, all tested specimens have been measured by two point-type laser sensors (Welotec, Type OWLF 4007 FA S1) which are oppositely arranged with a distance of 120 mm. These sensors record the sample’s bottom and top surface while it is moved with constant velocity through the laser arrangement. Each sample is scanned at least four times with 10 mm space on a length of 190 mm resulting in 40 measured points per millimeter. The measuring arrangement can be seen in Figure 3.

To implement the measured data in a 2D linear-elastic FE-model in ANSYS®, the profiles have to be smoothed by moving average and afterwards have to be compressed to reduce the amount of input points. These operations may not have any effect on the results of the FE-analysis. On the one hand, the stress concentration may not be decreased artificially by
P. Schaumann and M. Collmann

smoothing; on the other hand signal noise has to be removed in order to avoid the appearance of stress concentrations at locations, where they do not intend to be.

Figure 3: Measuring arrangement

In order to obtain more accurate results for the stress increase at the notch region so called submodeling is used. At first, a finite element mesh which is too coarse to produce satisfactory results at the notch region is used. Then in a second step a more finely meshed model of only the region (submodel) of interest has been generated to get accurate results for the stress increase at each notch, see also Figure 4. This means for a plate thickness of 20 mm four submodels are needed and for a plate thickness of 30 mm, where an additional notch exists between the two final passes, up to six submodels are necessary. Each submodel delivers a separate notch factor $K_t$.

Figure 4: Exemplary measured profile of sample #13 SAW 1, $t = 30$ mm, profile 1 (left) and FE-analysis (right)

To take the multi-axial stress state at the notch root into account the evaluation can be based on the first principal stress according to nominal stress hypothesis or the equivalent stress according to von Mises. For the presented investigations the equivalent stresses are applied to calculate the notch factor $K_t$. So, for every tested specimen the stress concentration $K_t$ is determined on basis of laser measured surface profiles. Finally, 144 profiles in total have
been evaluated statistically. The assumption of normal distributed values for $K_t$ has been proved and is adequate, see Figure 5.

Figure 5: Histogram for analysed $K_t$ of SAW samples with plate thicknesses of 20 mm (left) and 30 mm (right)

Table 2 summarises the mean values for $K_t$ and the standard deviation separately for the plate thickness $t = 20$ or 30 mm and the welding test sample to which the investigated profile belongs. Furthermore, the statistical evaluation is made for two variations: Firstly, $K_t$ has been considered over all profiles and all weld toes. And secondly, only the maximum $K_t$ of each specimen is taken into account because fatigue damage is a very local phenomena and crack initiates where the stresses are the highest.

Table 2: Summary of the averaged values for $K_t$ with related standard deviation

<table>
<thead>
<tr>
<th></th>
<th>number</th>
<th>mean over all $K_t$</th>
<th>standard deviation</th>
<th>number</th>
<th>mean of maximum $K_t$</th>
<th>standard deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>SAW 1, $t = 20$ mm</td>
<td>144</td>
<td>1.30</td>
<td>0.17</td>
<td>9</td>
<td>1.56</td>
<td>0.13</td>
</tr>
<tr>
<td>SAW 2, $t = 20$ mm</td>
<td>144</td>
<td>1.46</td>
<td>0.28</td>
<td>9</td>
<td>1.91</td>
<td>0.13</td>
</tr>
<tr>
<td>SAW 3, $t = 30$ mm</td>
<td>216</td>
<td>1.38</td>
<td>0.18</td>
<td>9</td>
<td>1.73</td>
<td>0.14</td>
</tr>
<tr>
<td>SAW 4, $t = 30$ mm</td>
<td>208</td>
<td>1.41</td>
<td>0.28</td>
<td>9</td>
<td>1.85</td>
<td>0.16</td>
</tr>
</tbody>
</table>

Although submerged arc welding belongs to the automated welding processes, random effects of welding parameters and boundary conditions influence the weld appearance. This explains the differences between the investigated samples. However, the scatter is still low. The mean $K_t$, when regarding all determined values, lies in a range of 1.30 up to 1.46. Compared to fillet welds where the notch factor $K_t$ varies between 3 and 4, or also gas metal arc welded butt joints where values up to 2.5 can be reached, the geometrical notch of the regarded welds can be designated as smooth.

c. Input parameters for the calculation

In the following the analytical determination of the S-N curves is shown by combining the notch strain and the crack propagation approach. The Uniform Material Law is applied for structural steel S355 (yield strength $f_y = 355$ MPa, ultimate strength $f_u = 495$ MPa and
Young’s modulus $E = 210'000$ MPa) to estimate the cyclic material properties. The material dependent input parameters for the crack growth equation (1) are also taken from literature. For comparing analytical derived fatigue life to the testing results mean values for $C$ and $m$ are needed. For the following estimations $C = 1.1 \cdot 10^{-13} \text{mm/(MPa}^m \text{mm}^{-1/2})$ with $m = 3.1$ have been applied according to the guideline DNV-OS-J101 [9] for welds in air. In order to calculate the maximum crack length $a_e$ not fracture toughness but the component thickness is considered.

**d. Comparison to fatigue testing results**

Basically, the fatigue tests show that for 22 of 30 performed fatigue tests the prediction of the decisive notch where the crack is initiated corresponds to the location of the maximum notch factor. Especially for higher load levels the agreement is almost 100%. Thus, the application of the notch strain approach is reasonable.

In Figure 6, the right diagram plots the analytically derived S-N of series SAW $t = 30$ mm in comparison to the testing results. For the analytical investigation by local fatigue concepts, two variations have been investigated: First, analysis with a constant notch factor $K_t = 1.79$ as average over all maximum values, second analysis with the maximum $K_t$ related to each specimen. It can be seen when taking the scatter points of $K_t$ into consideration the slope of the S-N curve draws nearer and the prediction of fatigue failure gets better. Besides, it can be observed that the share of the stage of crack initiation increases with decreasing load level and notch factor $K_t$.

**4 Conclusions**

Within this paper it can be shown that the prediction of fatigue resistance can be improved by applying local fatigue assessment, e.g. a combination of the notch strain and the crack propagation approach. The local heat input during welding and the application of weld metal leads to metallurgical and geometrical changes which often decrease the fatigue resistance.
These local effects can be considered by applying local fatigue concepts. Regarding the geometrical notch effect, the notch factor $K_t$ is an important input parameter for the notch strain approach. Random variations of the input parameters improve the prediction accuracy significantly. When using the individual analysed $K_t$, good agreement between the analytical derived S-N curve and the experimental results can be observed. Thus for welded joints, assuming random change of qualities which means not only geometry but also material strength, residual stress state and possible imperfections help to improve the prediction quality of fatigue life considerably.

5 ACKNOWLEDGEMENT
The authors would like to thank the German Federal Ministry for the Environment, Nature Conservation and Nuclear Safety for the financial support to carry out the investigations within the research project “OPTIWELD – Environmental and economical high performance welding techniques for steel tube towers of wind energy converters”.

6 REFERENCES
FREE-FIELD TESTS, LABORATORY TESTS, AND SOFTWARE TOOLS FOR THE OPTIMIZED PREPLANNING OF CORROSION PROTECTION MEASURES AND THE DEVELOPMENT OF OFFSHORE COMPATIBLE SENSORS TO INCREASE THE TECHNICAL RELIABILITY OF OFFSHORE WIND SUPPORT STRUCTURES

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Keywords: Free-Field Tests, Laboratory Tests, Corrosion protection, Software Tool, Offshore-compatible Sensors, Support Structures

Summary: This paper gives an overview about how different methods like free-field and laboratory tests as well as software tools can be used in the development process for corrosion protection measures as well as sensor application systems for offshore support structures and how these different methods and tools can be combined.

1 INTRODUCTION
Against the background of increased reliability of wind turbines (WT) in the offshore sector, the optimized preplanning and application of low maintenance and reliable methods of corrosion protection for steel structures as well as the technical monitoring of WTs by offshore capable sensors is of particular importance (s. Figure 1). Wind turbines in the offshore area are subject to strong mechanical and weather-related stresses. For operators of wind turbines in the offshore area are the reliability and readiness of the facilities of the highest importance, because failure and maintenance can lead to high costs. Monitoring Systems provide the information necessary to assess the actual state of the system and its components, and are thus a basis for planning and implementing condition-based maintenance actions [1]. For wind turbine components, equipped with sensors, the necessary system suitability and reliable long-term stability in the offshore sector, however, are not usually given over a period of up to 25 years [2].

Apart from the need of a reliable sensor system, also for the economic operation of an offshore wind turbine, a possible low-maintenance and easy to repairing corrosion protection of steel structure particularly in the area of splash zone is of great importance, however, since maintenance and repair operations in the offshore area are up to a hundred times more expensive than on land.

The first part of this document is concerned with the optimized pre-planning of resistant corrosion protection and gives an overview about the most important development steps illustrated by some results obtained from the project “BESTkorr”, whereas the second part of
this document deals with offshore compatible fiber optical sensors and gives a description of a first draft for a directive for the application of fiber optic sensors developed for offshore conditions in the project “OFFSHORE-FOS”. Additionally some explanations are given about the development of a laboratory test for standardized environmental tests of offshore suitable sensors and related components like plugs, cables and protection systems.

![Figure 1: Corrosion protection and offshore suitable sensor systems](image_url)

## 2 Preplanning of Corrosion Protection Measures

Corrosion damages at support structures based on coating failures cause an immense cost factor in the offshore wind energy [3]. An example: The costs of repairing one square meter coating of an offshore support structure are approximately 7000 Euro. Innovation is needed on the market, to get suitable products for reasonable prices. Decisions are mostly based on the price and not on the requirements, so the risk for a fail decision is very high. To give new products chances to enter the wind energy marked and avoid fail decisions at present products, software tools should support the decision process to visualize the properties of corrosion protection products and to see advantages and disadvantages of certain products. In the project “BESTkorr” such a software tool was one outcome and will be described in the following.
The main aim of the visualization tool is to support the decision process and underpin the expert arguments, so the number of fail decisions will be reduced in the future. Starting the process, a three level system with the domain of definition (e.g. technical properties, applicability, environmental compatibility, profitability, and reliability) was generated. In the three levels below a list of appending parameters per topic follows (e.g. adherence, UV stability, moisture suction, and impact resistance), which are also split in further lowermost parameters. These data were compiled which should be visualized by the software, shown in Figure 2. The weighting of these parameters was made by a committee of experts. Every aspect was considered, like the increasing adhesive strength, which goes mostly along with an increasing brittleness that leads faster to cracks caused by mechanical loads. Even if the barrier effect is not destroyed, it is evident the first point of attack for marine fouling like barnacles, as shown in Figure 3.

In total eight products were tested in the laboratory and in the field test rigs (shown at Figure 3, 4). Most products are more or less classical heavy corrosion protection coatings, two products were new coating products in the field of offshore wind energy. Especially at the new innovative products the applicability was tested on a typical steel structure surface.
The data of the testing results of many different tests were integrated in the software tool. It was found out that not in all cases high values of the parameters are beneficial, in some cases there exist limits that have to be reached or low values are more advantageous, so there was used a reversed system for positive low parameters to get an uniform appearance in the visualization. After structuring, the information was transferred in the programmed software. The basic software tool visualizes advantages and disadvantages and enables benchmarking of different coating products and hence makes them comparable even with a large number of parameters. Technical properties are listed in different levels and are weighted by expert knowledge (s. Figure 2). The generated tool provides a contribution to plan offshore support structures ahead, to take the corrosion protection more into account and give it the significant value in the cost calculation. Thereby the probability rises to get a functional overall structure, with a lifetime of the announced 25 years or more. At least, the tool can be used to figure out the optimal holistic corrosion protection concept for the support structure out of a multiplicity of individual systems.

3 OFFSHORE SUITABLE SENSORS
The research project “OFFSHORE-FOS” aims to develop guidelines for the use of certified fiber optic sensor technology including its system components to wind turbines in the offshore sector. The guidelines are designed to ensure that for both the manufacturer and the user-side, the sensors will be upgraded for use in this harsh environment and can be reliably applied or installed. This rulemaking is to take potentially interested users an existing uncertainty in dealing with fiber sensors in the offshore sector, raise the quality of the standards in the field of metrology and expand their area of application.

a. Fiber Bragg Grating (FBG) sensors – the basic principle
As a regular inspection, servicing and maintenance of structures on the open sea is logistically, financially and temporally extremely complex, only the standard monitoring using a suitable sensor can enable efficient event- and condition-based maintenance. In order to assess the technical reliability of the entire system - from which finally result the technical
lifetime analysis and the remaining operating lifetime - must be ensured in the first place, the technical reliability of the sensor systems. Only with reliable sensor systems and their data, model simulations can be validated and the structural design be optimized.

Technically reliable sensor systems are therefore an essential requirement for the offshore structures, especially if a measurement period of up to 25 years is estimated, in which the data must be reliably and reproducibly prepared. The fiber optic sensor system takes a special place as a measurement technique account of their safety from lightning and other outstanding features [4]. Base is an optical fiber, whereby the incident light is passed due to the larger refractive index in the fiber core. For the manufacture of such a sensor, into the core periodic reflection points are written to form the Bragg grating in their entirety. The strong light cannot pass by this grating, but will be reflected. Strains or temperature change the distance between the reflection points (the so-called grating period) and thus the wavelength of the reflected light.

b. Offshore loads in different zones

On technical systems in the offshore area acts a complex load spectrum, which consists of dynamic (system dynamics, wind and waves), environmental (water, salt, UV radiation) and marine biological influences [2]. In principle, a distinction is an area of air, the spray and the alternate immersion zone, the static immersion zone and the area of the seabed. The air field is characterized by influences of air temperature, UV radiation, salt air and freezing in winter. The largest part of an offshore wind farm is located in the air region. This is followed by the so-called splash zone. In addition to the impacts from the air field here shock temperature by splashing water (spray by waves) is added. This is followed by the alternate immersion zone. This area is characterized by limited tidal fluctuating water levels, i.e. the surfaces of the offshore wind turbines are temporarily exposed to air or completely covered by water. In addition, the area is characterized by micro-/macro biological vegetation, characterized by alternating mechanical loads due to waves and currents and mechanical stresses due to ice loads. The static immersion zone describes that part of an offshore wind turbine, which is consistently below the water surface. This affects the micro-/macro biological vegetation, the corrosive effect of salt water, mechanical stresses due to waves and currents or ice and abrasive effects of suspended particles (e.g. stones). Finally, there is the region below the seabed. This act of microbial induced corrosion, movements of the seabed as well as flushing by ocean currents.

4 Laboratory Test program

In the past no test method that fully replicates completely the load spectrum by an accelerated laboratory test program was available. Coatings and sensor applications must be designed to withstand the severe conditions of the North Sea climate. Therefore, an accelerated test program for the offshore simulation chamber of the Fraunhofer IWES has been developed for the selection of suitable materials and for the development of robust coatings for surfaces of structures and sensors, which allows for faster degradation of materials than possible in a free field weathering and which especially serves in the future as a test reference for a standardized testing methodology.
a. Worst-case load spectrum

The worst-case load spectrum describes the strongest individual loads (see also 3.b). Table 1 gives an overview of the load components for a worst-case load situation together with the ranges of values of the physical parameters. The numerical values of the load components results from own databases of Fraunhofer IWES based on environmental measurements in the North and Baltic Sea.

**Table 1: Overview load components for worst-case load spectrum in the alternate immersion zone**

<table>
<thead>
<tr>
<th>Loads</th>
<th>Value Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature – Air (TL)</td>
<td>-7°C – 45°C</td>
</tr>
<tr>
<td>Temperature – Surface (TO)</td>
<td>Bis zu 70°C</td>
</tr>
<tr>
<td>Temperature – Water (TW)</td>
<td>0°C (ice) – 28°C</td>
</tr>
<tr>
<td>Maximum value of spectral density of global direct radiation (PSD&lt;sub&gt;glob, direkt&lt;/sub&gt;) (Fino 1, 2004)</td>
<td>1009.3 W/m&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>Maximum value of spectral density of global total radiation (PSD&lt;sub&gt;glob, total&lt;/sub&gt;) (10% reflectivity of sea surface)</td>
<td>1.1*1009.3 = 1110.23 W/m&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>Salinity</td>
<td>Up to 3.5%</td>
</tr>
<tr>
<td>Shock water (ΔT)</td>
<td>± (30…40°C)</td>
</tr>
<tr>
<td>Surface load (F&lt;sub&gt;s&lt;/sub&gt;)</td>
<td>86 kN/m&lt;sup&gt;2&lt;/sup&gt; (Calculation for lighthouse “Alte Weser” according to Sainflow)</td>
</tr>
<tr>
<td>Further effects</td>
<td>Micro-/Macro biological fouling</td>
</tr>
</tbody>
</table>

b. Test program

The test program has been developed in the project “OFFSHORE-FOS”, focuses on the alternate immersion zone and is based on one of the most important standards for offshore corrosion protection: ISO 20340 (performance requirements for coating materials for structures in the offshore sector) [5]. The standard ISO 20340 describes weathering over a total duration of 4200 hours according to 175 days. The total test is run within 25 week cycles. A weekly cycle is composed of a plurality of temporally successive weathering phases UV / condensation (days 1-3), salt spray (day 4-6) and freezing (day 7). The individual phases are applied to the offshore load spectrum in principle, but toughening is specifically introduced. In principle, the toughening of this test program against the ISO 20340 standard results from adding of other load components and targeted modification of individual parameters according to the worst-case load spectrum. The superposition of loads enables - by successive switch-on/-off of load components - detailed investigations of damaging effects through individual loads in comparison to the collective. This is reasonable for comparison or validation purposes. In general, the test program consists of three phases. Overall, these three phases occur within one week according to 168 hours. This week test is repeated 25 times so that a total duration of the test is reached by 4200 hours. The three phases of a week are defined as follows:

- UV / condensation
  - Per Cycle: Sub-phase UV 4 hours, sub-phase condensation over 4 hours
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- Cyclic exchange between the sub-phases
  - Total duration: 9 cycles / 72 hours
- Weathering by salt spray
  - Total time: 72 hours
- Weathering by frost
  - Total time: 24 hours

In parallel phases of a program for shock temperature and mechanical stress on the sample sheets are running.

![Figure 5: Offshore simulation chamber (left), built-in sensor sheets (right)](image)

c. Offshore simulation chamber

The parameterization of the chamber is based on recorded environmental data at the test sites in order to fully grasp the load spectra. The load spectra are described by sets of parameters and used for setting the offshore simulation chamber. Figure 6 (left) shows the offshore simulation chamber in total (left) and sensor plates built in the offshore simulation chamber. The following are the features of the chamber: Corrosion (salt spray), thermal shock (-30 to +100°C, 10-95% RH), temperature shock by torrent water (temperature range +5 to +20°C), simulation of global radiation, mechanical loads (4-point bending test).

d. Test results

In the project “OFFSHORE-FOS” industrial partner were asked to provide sensor applications to be tested in free field trials in the North Sea at Sylt/List for 2.5 years and parallel in the offshore simulation chamber over 3 months (s. Figure 6). As a result of the accelerated life test in the offshore simulation chamber following could be stated:

- The most serious injuries, occurred during the long-term free field experiment in the North Sea, namely mechanical deformation of the sensor sheets by ice formation and corresponding destruction of the optical cable and the sometimes extreme marine biological fouling, especially with barnacles, could not be adjusted in the offshore environment simulation chamber.
- Especially the growth of barnacles, some of which strongly infiltrated the sensor protective layers and pried off or "tattered", caused a shortened lifetime of the sensors by dissolution of the protective layers.
Through the combined load cases and relatively complex load spectra, which can be set in the offshore simulation chamber and selectively controlled, this method is suitable as a test for offshore suitability of the sensors. It could be shown by the accelerated life tests on the sensor sheets that corrosion and corrosive attack in the protective cover of the sensors and in the wider community on the sensor sheet has no effect on the lifetime of the fiber optic sensors and thus plays a subordinate role.

Figure 6: Test results free field

5 CONCLUSIONS
This text gives an overview about the current development status for technically reliable corrosion protection and sensor application in the offshore sector. The development in the area of corrosion and sensor application is just at the beginning and there is a high research need in the development of sustainable materials for sensors and reliable corrosion protection. With the help of the testing equipment, the free field tests and the software tools, the goal is to develop assessment and quality standards for the technically reliable interpretation and application of corrosion protection and sensor materials in the offshore sector and to convert it also reproducible and fast into an industrial use.

6 REFERENCES
DEGRADATION AND CORROSION TESTING OF MATERIALS AND COATING SYSTEMS FOR OFFSHORE WIND TURBINE SUBSTRUCTURES IN NORTH SEA WATERS

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Keywords: Degradation, Corrosion, Offshore Wind Turbines, Coatings, Nanocomposites.

Summary: Degradation due to the environment in marine devices and components is one of the main issues facing the offshore industry, and is an even greater issue in offshore wind turbines (OWT) especially in floating units which require a higher degree of autonomous operation. In this paper, an ongoing research and experimental project is presented. The project CoMaRE - Assessment and mitigation of marine Corrosion in metallic components in Marine Renewable Energy (MRE) devices- is focused in the assessment of the marine corrosion phenomenon on metallic and composite components in MRE devices from a practical point of view and in the evaluation of different candidates for Corrosion Protection Techniques of these components. This project is developed under the MARINET FP7 initiative providing free access to IWES Fraunhofer (IWES Fh.) test facility with an estimated duration of 2 years; therefore 4 phases have been planned to achieve the desired results.

Remarkable results of degradation are also given, for example, in the case of composite material with and without multiwall carbon nanotubes (MWCNT) dispersed into its polymeric matrix.

1 INTRODUCTION

Several degradation and corrosion studies have been carried out for offshore wind turbines [1],[2], determining the effects of environmental exposure over several materials and coating systems. Recently, the addition of nanofillers to the materials has improved the mechanical properties of the materials [3],[4]. In the marine environment, the factors that mainly conditioning the surface damage of marine structures are: wear, corrosion, polymer photodegradation and marine fouling [5]. In the case of marine corrosion, the application of coatings is the most commonly method used for mitigating corrosion and is one of the best
protective methods for marine structures [5]. Marine environment is often classified in atmospheric, splash, tidal and submerged zone [6].

The current status of the CoMaRE project [3] is as follows: phase 1 and 2 has already been finished while phases 3 and 4 are still ongoing. They include an ambitious testing programme: more than 170 coupons in total. The infrastructure user group is formed by three research institutes (CTC, leader; Tecnalia and CENIM). During the latest phases of the CoMaRE project, the main activities have been the installation at Helgoland (IWES Fh. test facility) of almost all the coupons, as well as the preparation of tests in atmospheric exposure at the CTC’s headquarters. The specimens exposed to the marine environment cover a wide range of components used in floating and fixed offshore wind turbines such as mooring chains (with and without a Thermally Sprayed Aluminium coating –TSA), pieces of fibre ropes of different diameters, steel plates (S355 J2+N) and polymer composite specimens. The trials have been developed in the North Sea at the Helgoland Island (IWES Fh., Germany) and at Santander (CTC’s headquarters, Spain), 3 km. from the shoreline.

To assess the atmospheric corrosion, several wire-on-bolt (CLIMAT) specimens [7],[8] have been used. These tests allow determining the Marine Corrosion and Atmospheric Corrosion Indexes (MCI and ACI). The severity classification of atmospheric corrosivity related to MCI index is shown in Table 1.

<table>
<thead>
<tr>
<th>MCI Range</th>
<th>Classification</th>
<th>Significance</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-2</td>
<td>Negligible</td>
<td>Average Habitable Area</td>
</tr>
<tr>
<td>2-5</td>
<td>Moderate</td>
<td>Seaside</td>
</tr>
<tr>
<td>5-10</td>
<td>Moderately Severe</td>
<td>Seaside and Exposed</td>
</tr>
<tr>
<td>10-20</td>
<td>Severe</td>
<td>Very Exposed</td>
</tr>
<tr>
<td>&gt;20</td>
<td>Very Severe</td>
<td>Very Exposed, Wind and Sand Swept</td>
</tr>
</tbody>
</table>

The cumulative environmental loads at an offshore test site differ considerably from the loads on these materials under laboratory tests. The materials are subjected to harsh offshore conditions: temperature fluctuations, increased UV radiation, exposure to seawater, biologically induced corrosion and mechanical loads.

2 EXPERIMENTAL

Under this project, a relevant effort has been made to develop a testing program that covers many different components involved in fixed and floating wind turbines. Blades, mooring systems, support structure (steel plates), and other composite materials that may be used in floating devices has been tested during long exposure periods.

Table 2 summarizes the type, number of tested specimens, and exposed environment.
## Table 2: Summary of exposed and unexposed coupons (laboratory environment).

<table>
<thead>
<tr>
<th>Type</th>
<th>Coating</th>
<th>Environment</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Splash</td>
</tr>
<tr>
<td>GFR UPS</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>GFR UPS with MWCNT</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>Pre-Preg GFR Epoxy</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>Bare Steel</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>R3S chain slide</td>
<td>-</td>
<td>8</td>
</tr>
<tr>
<td>R3S chain slide TSA+topcoat</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Fibre ropes</td>
<td>Zn Galvanized</td>
<td>12</td>
</tr>
<tr>
<td>Wire-on-bolt</td>
<td>-</td>
<td>1</td>
</tr>
</tbody>
</table>

### Specimen preparation

The coupons are classified in the following types:

- Unsatarted polyester composite coupons: manufactured by hand layout and
  composed by a polyester resin and five layers of fibreglass textile. 4 samples were
  obtained, 2 of them added with a 0,1% wt of Multiwall Carbon Nanotubes
  (MWCNT). The three roll mills technique was used to disperse them in the polymeric
  matrix [9]. Samples were elaborated with different glass fiber content (samples a and
  b). They were cut into 280x25x4mm dimensions to be mounted in a Nylon support
  for marine exposure and on a wood frame in the case of atmospheric exposure.

- Pre-preg epoxy composite coupons: composite material of epoxy resin with
  unidirectional glass fibre was manufactured using the pre-impregnated (pre-preg)
  technique. Analogously to polyester composite coupons with 300x25x4 mm size,
  they were prepared for exposure and screwed to frames.

- Bare steel plates: 9 coupons of S355 J2+N structural steel were mechanized to
  400x90x4 mm. No coat was applied.

- Mooring chain slide coupons: 48 coupons were obtained from real offshore
  chainlinks of 157 mm of diameter made of R3S high strength steel. Half of them
  were coated with Thermally Sprayed Aluminium (TSA) of Sultzzer Metco with 99%
  wt content of Al and covered with a few micrometers of a topcoat of epoxy based
  paint. They were also mechanized to be arranged in the test rig.

- Fibre ropes: 3 different diameter fibre ropes (18, 38 and 46 mm), compound of a steel
  wire core and polyester yarns by wrapping. A total of 24 coupons were cut with a
  length of 300 mm approximately.

- Wire-on-bolt coupons are composed of Al wire coiled around 4 screwed rods: Al-Fe,
  Al-Cu, Al-PVC and Al (spiral). Standard dimensions and materials have been used in
  each coupon [8]. This method is only feasible if coupons are not submerged.
Initial appearance of the coupons is observed in Fig. 1.

![Coupons Before Exposure](image)

**Figure 1: Photographs of coupons before exposure:** a) GFR UPS composites with (black) and without MWCNT (white); b) Pre-preg GFR Epoxy; c) Bare steel; d) Mooring chain slide with (yellow) and without TSA (brown); e) Polyester fibre ropes; f) Wire-on-bolt specimen.

b. Environmental tests

For the environmental tests, different exposure times were selected for each coupon:
- Unsaturated polyester composite coupons: 8.5 months in all environmental conditions (atmospheric, splash, tidal and submerged).
- Pre-preg epoxy composite coupons: 12 months of planned exposure (still ongoing). Conditions: atmospheric, splash and tidal.
- Bare steel: 15 months of planned exposure (still ongoing). Conditions: splash, tidal and submerged.
- Mooring chain slide coupons: 6, 12, 18 and 24 months of planned exposure (still ongoing). Conditions: splash, tidal and submerged.
- Fibre ropes coupons: 12 and 24 months of planned exposure (still ongoing). Conditions: splash and tidal.
- Wire-on-bolt: used to determine atmospheric corrosivity, 3 months (90 days) are standardized. Conditions: atmospheric and splash.
c. Characterization tests

Several tests have been carried-out to know the effect of natural weathering (atmospheric and marine) over the different materials, as describe below for each specimen:

The weight of the unsaturated polyester composite coupons was obtained before and after exposure. Tensile tests were made to obtain the variation of the mechanical properties related to each kind of sample and material (with or without MWCNT addition) according to [10]. Glass fibre content of the composite was also measured in unexposed coupons.

Initially, for the wire-on-bolt specimens, the weight of Al wire was measured before exposure. In order to obtain MCI and ACI indexes, the corrosion products over the Al wire were cleaned according to [11] and final weight of uncorroded Al wire was measured [8].

Corrosion rates are aimed to be obtained from mooring slide coupons and bare steel coupons during the next phases of the project. The environmental effect on the tensile properties of pre-preg composite coupons will also be studied.

Additionally visual inspections to all the specimens have been done to know the formation of macrofouling and the corrosion/degradation visual performance.

3 RESULTS AND DISCUSSION

Finally, the degradation of the materials in offshore wind turbine components has been studied by three main methods: mechanical tensile properties, weight variation and visual observations of biofouling and corrosion.

In unsaturated polyester composite coupons, the results showed a better performance of the samples added with MWCNT compared to the same sample without MWCNT. The weight was increased in the marine environment, and on the other hand, weight loss was found after atmospheric exposure (Fig. 2). Fig. 3a shows maximum tensile stress for each environmental condition. Samples (a) without MWCNT only present the following mean property retention in max. tensile stress: 99,8% in splash, 98,4% in tidal and 92,4% in submerged zone. However, samples (a) with MWCNT showed better property retention in max. tensile stress: 103,7% in splash, 100,8% in tidal and 99,8% in submerged zone related to the unaged coupons of the same sample. Samples (b) had a property retention of 88,4% in atmospheric without MWCNT, while the retention with MWCNT in atmospheric condition increased to 93,5%. Tensile strength of each sample was consistent to its glass fibre content; a higher glass fibre content means higher maximum tensile stress (see Fig. 3b). Error bars represent 2 standard deviations from the mean in the figures.
The results of wire-on-bolt tests are condensed in Table 3. Corrosivity is found to be in good agreement with each location, which is highly affected with the distance to the shoreline [12]. A photographic summary of the current appearance of ongoing test and finished test is presented in Fig 3.
Table 3: Percentage Wire-on-bolt Aluminium wire weight loss and MCI Index for both exposure sites.

<table>
<thead>
<tr>
<th>Specimens</th>
<th>Al/Cu</th>
<th>Al/Fe</th>
<th>Al/PVC</th>
<th>Al/Spiral</th>
<th>MCI</th>
<th>Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td>S1 Helgoland</td>
<td>33,27 %</td>
<td>29,42 %</td>
<td>0,58 %</td>
<td>0,29 %</td>
<td>29</td>
<td>Very severe</td>
</tr>
<tr>
<td>S2 Santander</td>
<td>2,83 %</td>
<td>3,10 %</td>
<td>0,05 %</td>
<td>0,14 %</td>
<td>3</td>
<td>Moderate</td>
</tr>
</tbody>
</table>

Figure 4: Photographs of coupons after exposure: a) GFR UPS after 8,5 months in submerged; b) Prepreg GFR Epoxy after 10 months in tidal; c) Bare steel after 1 yr. in splash; d) Mooring chain slides after 1 yr. in splash; e) Fibre ropes after 1 yr. in tidal.; f) Wire-on-bolt specimen after 3 months in splash.

4 CONCLUSIONS

- Composite glass fibre reinforced unsaturated polyester material presents less environmental reduction in the maximum tensile stress values due to the addition of MWCNT. Higher losses of strength were observed in submerged condition.
- The Corrosivity Indexes (MCI and ACI) have been assessed with wire-on-bolt specimens and good agreement with theoretical values of each location has been obtained.
- All the composite coupons in tidal and submerged conditions have been colonized by marine biofouling. The same occurred also for metallic coupons, but in less degree, probably due to the detachment of rust outer layers, which in turn reduced steel thickness. Galvanized steel wire core of fibre ropes was corroded, and biofouling (including green algae, barnacles…) settled over the fibre rope coupons in tidal zone.
TSA coating provided an effective corrosion protection to the mooring slide chain samples, but in tidal and submerged zones was clearly damaged (especially the epoxy topcoat) by foulants, generating a rough surface finish.

5 ACKNOWLEDGEMENTS

The project CoMaRE, including the offshore tests at IWES Fraunhofer facility leading to the results presented in this paper, has received the support from MARINET, a European Community - Research Infrastructure Action under the FP7 “Capacities” Specific Programme. The authors would like to acknowledge H. Schnars and M. Hörnig (IWES Fraunhofer) for their help and support in the marine exposure and are also grateful to Laura Soriano for her work and support in CTC’s laboratory and exposure tests.

6 REFERENCES

DEGRADATION AND CORROSION TESTING OF MATERIALS AND COATING SYSTEMS FOR OFFSHORE WIND TURBINE SUBSTRUCTURES IN NORTH SEA WATERS
D. FERNÁNDEZ, R. RODRÍGUEZ, A. RODRÍGUEZ AND A. YEDRA
The Technological Centre of Components Foundation (CTC) was created in the year 2000 as a non-profit foundation. It is recognized as a Technology Center by the Ministry of Economy and Competitiveness.

Within the various fields of knowledge, the CTC is positioned in Experimental Sciences and Engineering. CTC develops its R. & D. activity in the following fields: Industrial Systems and Nuclear Components, Marine Renewable Energies, Industrial Automation and Robotics and Advanced Materials and Nanomaterials.

CTC has an Office located in Santander, North of Spain. Our headquarters are located in: Scientific and Technological Park of Cantabria (PCTCAN).
0. Index

1. Introduction
2. Experimental
3. Results
4. Conclusions
5. Future work and Acknowledgements
1. Introduction

Surface damage in OWT due to:
- Corrosion
- Polymer degradation
- Wear
- Marine biofouling

Protection techniques for OWT studied:
- Coatings (antifouling and protective)
- Corrosion resistant materials
- Free corrosion
- Improvement by addition of nanofillers

Scope of the study presented herein:

- Mechanical properties of composites environmentally affected
- Wire-on-bolt characterization of exposure sites
- Visual appearance of materials and coatings after exposure (marine biofouling...)

Corrosive severity of the exposure sites with Wire-on-bolt (CLIMAT) specimens (literature-Roberge P.R.) using MCI, Marine Corrosion Index:

<table>
<thead>
<tr>
<th>MCI Range</th>
<th>Classification</th>
<th>Significance</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-2</td>
<td>Negligible</td>
<td>Average Habitable Area</td>
</tr>
<tr>
<td>2-5</td>
<td>Moderate</td>
<td>Seaside</td>
</tr>
<tr>
<td>5-10</td>
<td>Moderately Severe</td>
<td>Seaside and Exposed</td>
</tr>
<tr>
<td>10-20</td>
<td>Severe</td>
<td>Very Exposed</td>
</tr>
<tr>
<td>&gt;20</td>
<td>Very Severe</td>
<td>Very Exposed, Wind and Sand Swept</td>
</tr>
</tbody>
</table>
1. Introduction
CoMaRE

Phases 1,2 completed
Phases 3,4 on going

Exposure sites:
Fraunhofer IWES (Marine)  CTC (Atmospheric)

- Wire-on-bolt coupons
- Mooring systems: chains, ropes...
- Standards coupons with and without coatings.
- Coupons of pre-preg composite used in blades.
- Coupons for accelerated tests.

Project Web pages:
http://ctcomponentes.es/en/comare-2/#/10/10/1/0
http://www.fp7-marinet.eu/access-menu-post-access-reports_comarephase1.html
http://www.fp7-marinet.eu/access_completed-projects_CoMaRE_phaseII.html

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2. Experimental Specimens

Note: Multi-Walled Carbon Nanotubes (MWCNT) were dispersed in the composite matrix using three roll mill technique.
2. Experimental Environmental tests

For the environmental tests, different exposure times were selected for each coupon:

1. **UPS composite coupons**: 8.5 months → atmospheric, splash, tidal and submerged.
2. **Pre-preg epoxy composite coupons**: 12 months (still ongoing) → atmospheric, splash and tidal.
3. **Bare steel**: 15 months (still ongoing). → splash, tidal and submerged.
4. **Mooring chain slide coupons**: 6, 12, 18 and 24 months (still ongoing). → splash, tidal and submerged.
5. **Fibre rope coupons**: 12 and 24 months. (still ongoing) → splash and tidal.
6. **Wire-on-bolt specimens**: ASTM G116 standardized, 3 months.

<table>
<thead>
<tr>
<th>Type</th>
<th>Coating</th>
<th>Environment</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Splash</td>
</tr>
<tr>
<td>GFR UPS</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>GFR UPS with MWCNT</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>Pre-Preg GFR Epoxy</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>Bare Steel</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>R3S chain slide</td>
<td>-</td>
<td>8</td>
</tr>
<tr>
<td>R3S chain slide TSA+topcoat</td>
<td>-</td>
<td>8</td>
</tr>
<tr>
<td>Fibre ropes</td>
<td>Zn Galvanized</td>
<td>12</td>
</tr>
<tr>
<td>Wire-on-bolt</td>
<td>-</td>
<td>1</td>
</tr>
</tbody>
</table>

**Total project: 170 test coupons.**
The results of tensile tests and weight analysis of GFRP (with and without MWCNT):

- Weight gain in marine vs. weight loss in atmospheric
- ↑ mechanical properties retention after exposure for the same sample additivated with MWCNT.
- ↑GFC indicates ↑max. Tensile stress
3. Results

Composites

<table>
<thead>
<tr>
<th>Condition</th>
<th>Without MWCNT</th>
<th>With MWCNT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Splash a</td>
<td>99.76%</td>
<td>103.68%</td>
</tr>
<tr>
<td>Tidal a</td>
<td>98.36%</td>
<td>100.77%</td>
</tr>
<tr>
<td>Submerged a</td>
<td>92.34%</td>
<td>99.76%</td>
</tr>
<tr>
<td>Atmospheric b</td>
<td>88.38%</td>
<td>93.54%</td>
</tr>
</tbody>
</table>

Properties retention in max. tensile stress [%]

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3. Results

Wire-on-bolt and visual appearance

<table>
<thead>
<tr>
<th>Specimens</th>
<th>Al/Cu</th>
<th>Al/Fe</th>
<th>Al/PVC</th>
<th>Al/Spiral</th>
<th>MCI</th>
<th>Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td>S1 Helgoland</td>
<td>33,27 %</td>
<td>29,42 %</td>
<td>0,58 %</td>
<td>0,29 %</td>
<td>29</td>
<td>Very severe</td>
</tr>
<tr>
<td>S2 Santander</td>
<td>2,83 %</td>
<td>3,10 %</td>
<td>0,05 %</td>
<td>0,14 %</td>
<td>3</td>
<td>Moderate</td>
</tr>
</tbody>
</table>

Submerged

Tidal

Splash

Tidal

Splash

Splash
1. **Composite GFR unsaturated polyester** showed ↓ reduction in the maximum tensile stress with MWCNT in each sample. ↑ losses of strength were observed in submerged environment.

2. **Corrosivity Indexes** (MCI and ACI) have been assessed with **wire-on-bolt specimens** and good agreement with theoretical values of each location.

3. **Composite coupons** in tidal and submerged conditions were colonized by marine biofouling. ↓ biofouling has grown over **metallic coupons**, probably due to the detachment of rust outer layers, which ↓ steel thickness. Galvanized steel wire core of **fibre ropes** was corroded, and these coupons showed biofouling (incl. green algae, barnacles...) in tidal zone.

4. **TSA coating** provided an effective corrosion protection to the mooring slide chain samples, but in tidal and submerged zones was clearly damaged (especially the topcoat) by foulants, generating a rough surface finish.
5. Future work and Acknowledgements

Coming up in CoMaRE project during phases 3 and 4:

• Novel AF Coating systems additivated with nanofillers. Results compared with AF Coatings without nanofillers. Surface effects.
• New Corrosion Test Site “El Bocal” in the Northern coast of Spain.
• Laboratory accelerated tests for comparison with field results.
• Further results from exposed coupons

The project CoMaRE, incl. the offshore tests at IWES Fraunhofer facility leading to these results, has received support from MARINET, a European Community - Research Infrastructure Action under the FP7 “Capacities” Specific Programme.

The authors would like to acknowledge H. Schnars and M. Hörnig (IWES Fraunhofer) for their help and support in the marine exposure and are also grateful to Laura Soriano for her work and support in CTC’s laboratory and exposure tests.
Thanks for your attention
FATIGUE BEHAVIOUR OF HIGH-STRENGTH GROUT IN DRY AND WET ENVIRONMENT

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Keywords: Fatigue design, high-strength grout, fracture behaviour

1 INTRODUCTION
High-strength grout is often used in cyclic-loaded connections of offshore wind turbines, e.g. tube in tube connections between tower and foundation, so-called grouted joints [1], [2]. In these connections, the circumferential gap between the outer and inner tube is filled with high-strength grout. According to [3], high-strength grout is a mortar with a maximum grain size of 4 mm, a compressive strength grade of at least C50 and very fluid consistency. The verification at the fatigue limit state might be design-relevant for grouted joints.

The fatigue behaviour of high-strength grout has been very little examined yet [2]. Generally, the transport of water in the microstructure of concrete depends on the amount of capillary pores and defects. There are a few test results for fatigue behaviour under water extant, but many concretes in the area of normal compressive strength and a small number of samples have been investigated. Water-saturated normal-strength concrete failed significantly earlier under fatigue loading. According to [4], the number of load cycles to failure is 10 – 100 times lower than for concrete tested in dry conditions. Based on these tests, it is postulated that saturation leads to pore water pressure in the microstructure of normal concrete. This pressure induces local tensile stresses in the concrete structure, which lead to lower numbers of cycles to failure [5]. In [2] a shorter fatigue life is documented for high-strength grout submerged in water.

The composition of high-strength grout differs significantly from the composition of normal-strength concrete. High fluidity, high cement and powder content and a small grain size is characteristic for high-strength grout. Therefore, common high-strength grouts reach high compressive strength classes in the range of 80 MPa and above. Furthermore, high-strength grout has a very small pore volume, so that the water transport into the microstructure is not directly comparable with normal-strength concrete. Therefore, certain findings for high-strength grout cannot be derived from the literature.

Standards, such as fib-Model Code 2010 [6], cannot be initially applicable for the fatigue design of high-strength grout under water, because the fatigue design in fib-Model Code 2010 is based on test results for normal-, high- and ultra-high-strength concrete specimens in a dry
environment [7]. Therefore, the fatigue behaviour of high-strength grout has to be determined by fatigue tests. In the following, first test results of a common high-strength grout under cyclic loading tested in dry conditions and under water are shown comparatively. The results are discussed in the context of fib-Model Code 2010.

2 FATIGUE BEHAVIOUR OF HIGH-STRENGTH GROUT

Fatigue behaviour of high-strength grout under water under pure compression is intensively investigated at the Institute of Building Materials Science, Leibniz Universität Hannover. Therefore, a special test set-up was developed which allows fatigue tests on cylindrical specimens, both submerged in water and in dry conditions. The test set-up is shown in Figure 1. A servo hydraulic testing machine with a maximum force of 2.5 MN is used for the fatigue testing.

![Test set-up](image)

Figure 1: Test set-up [8]

The test set-up (Figure 1) is filled with water before the specimen is placed into it for tests submerged in water. The water level is about 1 – 2 cm higher than the upper surface of the specimen. The applied force and the displacement of the hydraulic cylinder are recorded while testing. Furthermore, the strain of the specimen is measured by three laser distance sensors, which are placed around the specimen at an angle of 120° to each other.

In the following, selected results of fatigue tests in dry conditions and submerged in water are shown comparatively. Cylindrical specimens with a diameter of 60 mm and a height of 180 mm of standard high-strength grout with a compressive strength grade C80 are used for these tests. The specimens were demoulded after 48 hours. One half of them was stored in a standard climate (20°C, 65% r. H.) until testing after 28 days, and the other half was stored in water (20°C) until testing after 28 days. The water storage prevents the dehydration of the specimen. This is might be important for the fatigue behaviour of the high-strength grout stored and tested under water.
Before fatigue testing, the loaded surfaces of all specimens were plane parallel ground and afterwards fine polished. The static reference strength has to be determined before the fatigue tests. The static reference strength was determined force controlled with 1.41 kN/s in the same conditions as the fatigue tests, that means, three specimens in dry conditions and three specimens submerged. The static reference strength of the submerged specimens was, on average, 7% higher than the static reference strength of the specimens stored and tested in dry conditions. The maximum stress level in the fatigue tests was varied in four steps, whereas the minimum stress level was kept constant. The standard load frequency in the tests conducted was 10 Hz. In case of a high temperature increase of the specimens while cyclic loading at 10 Hz, the following fatigue tests were carried out at a frequency of 1 Hz. There are indications that a high temperature increase leads to a reduction of the number of load cycles to failure of high-strength grout [9].

a. Number of load cycles to failure

The number of cycles to failure (log N) for the tests performed in dry conditions and under water is shown in a comparative manner in Figure 2.

![Figure 2: Number of cycles to failure dry and submerged](image)

The dry specimens reached a higher number of cycles to failure for $S_{c,\text{max}} = 0.85$. Here, log $N_{\text{dry}}$ is in the range of 2.9 – 3.1 and log $N_{\text{submerged}}$ in the range of 2.7 – 2.9. On average, the tests conducted in dry conditions achieved an 8% higher number of cycles to failure.

At a maximum compressive stress level $S_{c,\text{max}} = 0.75$, the influence of water on the fatigue behaviour could not be clearly established. The mean value of log N for both test series is 3.7. Only the range of variation for the submerged specimens seems to be higher, as can be seen
from Figure 2. At this stress level, a test with a reduced frequency (1 Hz) was conducted, because there are indications that the fatigue behaviour of high-strength grout is also dependent on the frequency [9] at lower maximum compressive stress levels. As can be seen from Figure 2, the number of cycles to failure log N of 1 Hz tests is smaller than log N of the 10 Hz tests. The difference is not significant and within the normal range of variation for concrete under fatigue loading.

Figure 2 shows a significant influence on the fatigue behaviour of the frequency for the dry specimens at a stress level of $S_{c,\text{max}} = 0.65$. Here, a temperature increase of the specimens tested at 10 Hz of $\Delta T_{\text{surface, dry, 10 Hz}} = 26$ K could be detected. By comparison, a temperature increase of only $\Delta T_{\text{surface, dry, 1 Hz}} = 2$ K could be detected by testing at 1 Hz. The lower temperature seems to be equivalent to a significant increase of the number of cycles to failure for the 1 Hz test. As can be seen in Figure 2, the enhancement of log N is about 30%. Other tests on high-strength grout in dry conditions under fatigue loading show a similar behaviour [9]. In contrast to that, an influence of frequency on the fatigue behaviour of the submerged specimens could not be detected ($\Delta T_{\text{surface, submerged, 10 Hz}} = 2$ K, $\Delta T_{\text{surface, submerged, 1 Hz}} = 1$ K).

Figure 2 shows, furthermore, that log N of the submerged and dry specimens tested at 10 Hz is in the same dimension. By comparison, it could be seen that log N of the dry specimens tested at 1 Hz and log N of the specimens tested under water with the same frequency is significantly different. In dry conditions, log N is about 20% higher than under water.

The evaluation of Figure 2 shows that differences in the cycles to failure reached between the test results of the dry specimens and the submerged specimens are dependent on the maximum compressive stress level.

b. Strain development

The influence of water on the fatigue behaviour of high-strength grout can also be seen in the strain development. In the following Figure 3, the strain developments of the fatigue tests in dry conditions and under water ($S_{c,\text{max}} = 0.85$, $S_{c,\text{min}} = 0.05$, $f_p = 10$ Hz) are presented with respect to the normalized number of cycles to failure $N/N_f$ (left) and the absolute number of load cycles to failure N (right). The amplitude of $S_{c,\text{max}}/S_{c,\text{min}}$ is generated within 12 cycles; these cycles are not included in Figure 3. Three specimens were tested until failure for both test conditions.

Figure 3: Strain development
Figure 3 (left) shows that the strain development of the submerged specimens has a typical three-phase course. Furthermore, it can be seen (Figure 3, right), that the slope within these three phases is similar for both environmental conditions.

The ultimate strain of the dry specimens at maximum and minimum stress is larger than the ultimate strain of the specimens tested under water. In Table 1, the ultimate strain of the specimens tested under water and in dry conditions are given separately for \( S_{c,\text{min}} \) and \( S_{c,\text{max}} \). The mean value of the ultimate strain of the tests of the submerged specimens considered is smaller than the mean value of the dry specimens in Figure 3 (left). The difference is about 30\%, as can be seen in equation 1.

**Table 1: Example of the construction of one table**

<table>
<thead>
<tr>
<th></th>
<th>dry conditions</th>
<th>submerged</th>
</tr>
</thead>
<tbody>
<tr>
<td>sample number</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>( \varepsilon_{\text{Sc,\text{min}}} ) [%]</td>
<td>-0.702</td>
<td>-0.682</td>
</tr>
<tr>
<td>( \varepsilon_{\text{Sc,\text{max}}} ) [%]</td>
<td>-1.295</td>
<td>-1.290</td>
</tr>
<tr>
<td>( \varepsilon_{\text{mean}} )</td>
<td>-0.705</td>
<td>-1.265</td>
</tr>
</tbody>
</table>

\[
\frac{\varepsilon_{S_{\text{c,\text{max}}},\text{ultimate},\text{sub}}}{\varepsilon_{S_{\text{c,\text{max}}},\text{ultimate,dry}}} = \frac{-0.465}{-0.705} = 0.66 \rightarrow 66\%
\]

\[
\frac{\varepsilon_{S_{\text{c,\text{max}}},\text{ultimate},\text{sub}}}{\varepsilon_{S_{\text{c,\text{max}}},\text{ultimate,dry}}} = \frac{-0.896}{-1.265} = 0.71 \rightarrow 71\%
\]

\( (1) \)

c. Fracture behaviour

The fracture behaviour of the specimens tested under water at low maximum compressive stress levels, e.g. \( S_{c,\text{max}} = 0.65 \), differs from the “normal” failure behaviour of high-strength grout tested in standard dry conditions. That means that spalling of the specimen surface was detected after about 30 – 40\% of the number of cycles to failure (cf. Figure 4, left).
An increasing turbidity of the surrounding water was observed with increasing test duration for the tests with $S_{c,\text{max}} = 0.65$ and 0.55 (Figure 4, right). The turbidity was observed at about $N/N_f = 0.5$. The reason for this turbidity has not yet been discovered. It is assumed that the turbidity is caused by cyclic loading, in terms of water movement in micro defects arising from the spalling, cracks/microcracks and pores. Within the structure of the specimen, the water enters disperses fines of the high-strength grout. The fines are washed out when the water flows out of the specimen. This flow became visible in the form of “fog patches” in the surrounding water, as can be seen in Figure 4 (right).

Shortly before failure ($N/N_f \approx 0.95$), it was detected that large fragments of the specimen broke out in the area of the pressure plates of the testing machine (cf. Figure 5) without leading to the failure of the whole specimen.

By contrast, sudden and explosive failure is characteristic in fatigue tests in dry conditions.

An explanation for this behaviour might be water pressure which can be induced in water-filled cracks/microcracks and pores while fatigue loading. This pressure may lead to tensile stresses in the microstructure of the high-strength grout. When this pressure exceeds the maximum tensile stress of the material, a break out of a fragment occurred without failure of the specimen. A break out of a fragment in dimension, as can be seen in Figure 5, is equivalent to a weakening of the cross-section of the specimen. As a result of this, the normalized stress in the remaining cross-section must be higher than before the break out. This must lead to an accelerated damage progress in the remaining cross-section, which leads to failure. The phenomena observed might be a possible explanation for the lower number of load cycles to failure of the submerged specimens.

At this moment, it is assumed that the phenomena and effects observed are geometry-dependent; because of that, further fatigue tests under water will be carried out with larger cylindrical specimens at the Institute of Building Materials Science, Leibniz Universität Hannover.
3 FATIGUE DESIGN

The new fib-Model Code 2010 is applicable up to and including compressive strength grade C120. This standard refers to the fact that less favourable conditions exist under water, but there is no information about a reduction due to less secure results. The number of cycles to failure of the tests in dry conditions and under water are shown together with the Wöhler curve according to fib-Model Code 2010 in Figure 6.

As it can be seen in Figure 6, the results of the tests performed in dry conditions are well described by the Wöhler curve of fib-Model Code 2010 [6], although some values are below this curve ($S_{c,max} = 0.75$). The number of load cycles to failure of the submerged specimens for $S_{c,max} = 0.75$, 0.65 and 0.55 are below the Wöhler curve, according to fib-Model Code 2010. It could, therefore, be useful to change the slope of the Wöhler curve according to [6], or to shift the curve parallel in the direction of the coordinate origin, but at this moment, an adjustment of the fib-Model code 2010 Wöhler curve for high-strength grout is not useful because of a small number of test results. Further fatigue tests with different high-strength grouts, larger test specimens and a higher sample size have to be carried out before an adjustment can be considered.
4 CONCLUSIONS
The first results of investigations concerning the fatigue behaviour of high-strength grout tested under water and in dry conditions are presented in this paper. The fatigue behaviour of high-strength grout depends strongly on the environmental conditions. Water storage and fatigue testing under water have an influence on the number of load cycles to failure, the strain development and the fracture behaviour. The influence of water on the fatigue resistance of the high-strength grout seems to increase for lower maximum stress levels. Further investigations should be carried out on high-strength grout materials to verify the effects and to get a better understanding of the phenomena observed.

The fatigue behaviour of high-strength concrete is examined at the Institute of Building Materials Science, Leibniz Universität Hannover within the framework of a BMU-funded research project called “ProBeton”.

5 REFERENCES
CLOSED-LOOP PRODUCT LIFECYCLE MANAGEMENT FOR OFFSHORE WIND TURBINES BY UTILISING ENHANCED MAINTENANCE CONCEPTS

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Keywords: Product Lifecycle Management, Operation and Maintenance, Offshore Wind Turbines, Maintenance Concepts, Data Mining

Summary: The paper describes a requirement analysis and a general methodological approach in order to build an information system aiming at efficient and lean processes for maintenance measures of complex technical systems. The methodological approach establishes a closed-loop product lifecycle management in order to create a stronger link between the research and design phase of product development and the use-phase of former product generations. The continuous improved process of technology and technical progress in this field will be actively supported by the methodology presented.

1 INTRODUCTION

Referring to the hype cycles of Gartner, the energy production in far-shore wind parks is in the stage between “Trough of Disillusionment” and “Slope of Enlightenment”. The technical progress in this field will affect the upcoming concrete solutions [1]. Several studies (cp. [2; 3, 4]) show that cost saving potential in the field of offshore wind energy is dedicated to technical improvements. The development of the following product generations is mainly affected by the search for former experience and information as well as best practice examples, which will improve the performance and competitiveness of such products. Offshore wind turbine designers are currently struggling to get and simulate data they need in order to improve the design of components of the entire system. Accordingly, the designers rarely gain their knowledge from real data in the development process for the following product generations but from test runs and experiments under laboratory conditions.

The past technical progress and recent trends illustrate, that the design and construction of wind turbines afford bigger, heavier and more complex systems; especially since the installation of wind turbines was transferred to offshore sites [5]. In general, it is an extensive
organizational and technical task to obtain broad amounts of data of complex technical systems like wind turbines due to a lack of consistent product lifecycle concepts throughout various phases. Understanding the complex situation of collecting and/or getting data and information of wind turbine components will improve the action-oriented knowledge for maintenance of wind turbines itself as well as for the usage in other lifecycle phases.

2 PRODUCT LIFECYCLE OF OFFSHORE WIND TURBINES

In order to achieve the stage of “Plateau of productivity” and obtain lasting renewable energies, a major goal is to achieve dual sustainability, meaning to simultaneously achieve environmental and economical sustainability [6]. Considering wind energy, there is an even weightier interest in offshore wind energy due to higher construction and maintenance costs and related risks [7]. With a focus on the complete lifecycle, the three main phases have to be considered in the product lifecycle management: (1) Beginning-of-Life (BOL) with tasks like imagination definition and realization, (2) Middle-of-Life (MOL) with product use, support and maintenance; and finally (3) End-of-Life (EOL) which includes product retirement disposal and recycling [8, 9]. The MOL is the longest phase of the lifecycle of wind turbines with a lifetime of up to 25 years. The majority of data is created during this phase. An overview of the typical phases of offshore wind turbines respectively farm can be taken from Figure 1.

![Figure 1: Lifecycle of offshore wind turbines](image)

Offshore wind parks require enhanced operation and maintenance (O&M) concepts in order to save costs in the MOL phase. That latter can constitute up to 30% of overall costs for offshore wind turbines [10, 11]. The constraints of accessibility, logistics, weather conditions and human resources lead to a great interest in smart technologies that help to remotely monitor and maintain wind turbines in order to plan all maintenance measures carefully. Currently an important amount of data is already recorded during the MOL phase, while today’s stakeholder is mainly the O&M department. This data has the potential to satisfy further interests in other product lifecycle phases and have an effect on the economic and the ecological performance of offshore wind energy. Information sharing in terms of knowledge management along the product lifecycle promises to reach the stage of “Plateau of productivity” faster and is associated with lower risks [12, 13, 14, 15, 16]. In the following, essentially the case of combining the use-phase with the design of further product generations will be investigated in greater depth.
3 Concept of Closed-Loop Product Lifecycle Management

A knowledge infrastructure consisting of technology, structure, and culture along with a knowledge process architecture consisting of acquisition, conversion, application, and protection of data along the product lifecycle of offshore wind turbines will accordingly support the goal of a closed-loop product lifecycle management [17]. Closed-loop in this sense defined by [18] describes “as strategic approach for the effective management of product lifecycle activities by using product data/information/knowledge which are accumulated in the closed-loops of product lifecycle […]”. Accordingly, the deviated challenge is to transform data into information and action-oriented knowledge. The data from the offshore wind turbines comes from integrated sensors and technical data from the O&M [19]. According to the considered case of improving next generation systems, the data of the individual product lifecycle will be stored and processed in order to get the opportunity to verify the configuration of a product by comparing the calculated data of the design-phase with the real data of the use-phase. As shown in the following Figure 2 the in-use product works as a permanent test rig and shares information all over the product lifecycle. All actors from the entire lifecycle have the possibility to access, manage, and control the related information. The information, of course goes back to the designers and production engineers so that the information flow is closed over the entire product lifecycle [20]. To enable this, a digital or cyber representation of the real product is needed, which means the existing information systems need to be developed further. Following this approach, according concepts of the “Product Avatar” and “Cyber-Physical Systems” will be introduced.

Figure 2: Example of exchange of data and information across lifecycle phases

In order to manage the communication between the intelligent product and the different stakeholders along the lifecycle, the stakeholder-specific digital representation called “Product Avatar” was introduced by [21], which is a valid approach for the challenges addressed by wind turbines according to this paper. In general, the concept and the core idea of the “Product Avatar” imply that each product, in our case the offshore wind turbine or its
components, should have a digital counterpart by which it is represented. Thereby, the different involved stakeholders are able to gain access to the information acquired via the lifecycle and benefit from it through value-added services based on the individual product lifecycle phases [22]. Consequently, a part of the concept implies the establishment of suitable interfaces towards different types of stakeholder. For intelligent products like offshore wind turbines, the interface could, for example, be a common messaging interface, service or agent. For the human actors, such as design engineers or service technicians, these interfaces are for example dedicated desktop applications, web pages or mobile “apps” tailored to the specific information and interaction needs [22].

Acquiring and using data and information essentially in the use-phase of a product requires the enabling of technologies in which the so called “Cyber-Physical Systems” play an important role. The first definition was introduced by Lee [23] and later replenished by Broy [24]. According to this, a cyber-physical system could be understood as a physical object, e.g. a wind turbine, which is equipped with sensors, a data processing module, a communication module as well as actuators, which determine and control physical processes and communicate with other systems [24, 23]. Hence, concentrating on certain points, an offshore wind turbine is a good example for a cyber-physical system. The wind turbines detect their own status and their environment through sensors. The data processing module stores these data and the communication unit transfer the data to the supervisory control and data acquisition system (SCADA). Human actors or other technical systems analyse the data. They are able to effect the environment via actuators, e.g. actuate an axis, reset a system, change parameters, etc.

In the next section, coming back to the challenge of information back flow between early lifecycle phases and the use phase, a more concrete implementation approach of a system will be described.

4 APPROACHES FOR IMPLEMENTATION

For the purpose of collecting data in O&M and the further transfer of information through a system which connects O&M data and their potential usage in other lifecycle phases like design as well as construction and logistics, etc., the following aspects need to be implemented. According to Figure 3, the input part consists of mostly raw data from the turbine itself. Moreover, context-related data has to be part of the input. The following step encompasses the processing of data so as to gain information and action-oriented knowledge. On the output perspective, gained information has to be rehashed for the different stakeholders. These main steps enhance the typical state of practice in handling data from operations and maintenance as follows.
Data acquisition in the use-phase mainly encompasses individual and isolated data sources. The main data that is used today within the use-phase for operations and maintenance purposes consists of technical data from the SCADA system and historical data from past maintenance measures, e.g. orders, checklists, feedback forms, etc. Other known and also isolated data sources, for instance, encompass historical weather data, weather forecasts, general and statistical reliability data, functional interconnections and possible failure modes. In order to answer questions that focus on interlinked aspects of the different data sources, a uniform data access is necessary to enable joint queries. Exemplary questions within the use-phase are for instance:

- Which is the best date for yearly inspection under consideration of weather risk and reliability data?
- In which way do wear-out measurements have an influence on statistical residual lifetime?
- Which component shows irregular behaviour and is likely to fail?
- etc.

Considering the usage of data along different lifecycle phases the possible queries are even more complex. They could, for instance, include:

- Are there some specific patterns in the input data of a wind turbine that announce future failures in advance?
- Are there specific designs that show a significantly higher failure probability?
- Are the simulated or anticipated loads and the derived wear comparable to the measured situations in reality?
Summing up the challenges for the input site of the system, individual data sources have to be transformed for a uniform access and clear presentation. This takes the specific challenges that are derived from sensor technology (e.g. high frequency data of vibration analysis in contrast to temperature measurements), embedded systems or communication technologies into particular account (compare also Figure 3).

b. Processing

Going further from the input side, raw data has to be processed according to the requirements of the individual stakeholders in the different phases. In order to provide and process data, a data cloud concept is proposed according to Figure 3. While the transformation of the input sources according to a uniform data access is the first step that was already mentioned in the part before, the cloud encompasses further processing steps according to the more detailed illustration in Figure 4. In general, transforming raw data to information requires

- methods that interlink data sources in the sense of data mining in order to search for conspicuous patterns,
- methods that predict future developments through an algorithmic processing which encompasses pattern recognition, classification, time-series prediction, statistical residual life prediction, etc. and
- methods that are able to apply above mentioned procedures on continuous data streams again taking the complexity of different data sources into account (i.e. aspects of complex event processing [25]).

![Figure 4: Processing tasks within a data cloud for the use-phase of products](image)

The last processing step within the cloud has to consider that the available information pool is not specific for the individual stakeholders along the lifecycle so that further processing is necessary. Following the aim to provide concrete action-oriented knowledge in the sense of stakeholder-specific information, the previous described concept of the “product avatar” filters information, communication channels and user interfaces according to the stakeholder’s
requirements. Each product avatar of the wind turbine is a tailored virtual representation of the real turbine. In order to integrate these concepts in the social technical systems along the lifecycle, an acceptance to integrate the new concepts and to work with new technologies is mandatory. Essentially with respect to information sharing across companies but also across departments in single enterprises, particular business models are needed. The value of information in contrast to the costs for acquisition and processing is the determining ratio, which leads to win-win-situation between stakeholders, or not.

5 CONCLUSION AND OUTLOOK

The paper describes the lifecycle of offshore wind turbines and reflects today’s state of practice with respect to information usage and information sharing. While the motivation to share information along the lifecycle is clarified, essentially the case of closed-loop product lifecycle management is a valuable concept for supporting the design of next generation turbines based on experiences from the field. Data- and information-driven concepts like the “product avatar” show the applicability to process and document this required experience. Therefore, it is necessary to change the information access and processing according to the stated challenges. The presented architecture is one possible approach for that. Within ongoing projects the technically related challenges (e.g. big data, processing times, etc.) of implementing the described concepts are currently investigated and lead to first prototypes. However, having a functional proofed and performing infrastructure for data processing and information sharing is the first step for further investigations. So as to be able to give an example, great effort is needed for experiments that deal with the tailored integration of all this information in the existing workplaces. To be more concrete, those broader information-sharing concepts will raise the question of how information could be integrated into computer-aided design software in a way that it is stakeholder-specific and context-specific with respect to the engineer’s needs.

6 REFERENCES


Keywords: offshore O&M, adverse weather conditions, offshore operations

1 Introduction

Although modern onshore wind turbines (WT) attain high technical availability of up to 95-98 % [1], their offshore counterparts are still far from these figures. This fact can be mainly attributed to harsh environmental conditions limiting accessibility of the turbines and influencing components failures as well as to logistical issues (e.g. distance from shore). A large part of component failures can be often eliminated before they occur (preventive maintenance) or through regular maintenance works. Nevertheless, methods to predict remaining life expectancy are by for not perfect and component failures take place, prompting corrective maintenance measures. In such cases, large components failures such as the gearbox and rotor blades are of critical importance. The costs for their replacement arise not only from costs of the components but from the deployed vessels, their availability, operational conditions etc. The employed vessel strategy can, therefore, have large implications on the total O&M costs. The goal of this paper is to compare different vessel strategies for the replacement of large components based on their life cycle costs, taking into consideration weather conditions and their seasonal effect on both failure rates as well as maintenance procedures
2 Methodology

The general approach of the paper is shown in Fig. 2.1. The paper will first proceed to mention inputs used during the analysis such as failure statistics, weather conditions, wind farm specifications, etc. In order to account for non-linear effects of adverse weather conditions the seasonal behavior of failures will be taken into consideration. Their output will represent randomly generated events of large component failures, which will be further fed into a WaTSS simulation.

![Fig. 2.1: General approach (left); WaTSS Method basic principle (right)](image)

The WaTSS method is a new approach to evaluate the adverse weather impact on offshore operation. Contrary to the conventional statistics-based approach, which was inherited from the oil & gas industry, the WaTSS method considers the temporal sequence of operations and their restrictions by directly mapping them onto weather time series of multiple years (see Fig. 2.1 (right)). That is, the schedules are virtually executed onto historical weather time series. Doing so, statistical information about their duration can be gathered (e.g. P50, P90 durations) [1].

The durations over the life time of the offshore wind farm originating from the WaTSS analysis are translated into costs by considering vessel rates. These are later used as KPI in order to select the optimal vessel for the wind farm.

3 Scenarios

3.1 Reference Wind Farm

The Fraunhofer reference wind farm represents a virtual (non-existent) wind farm which is representative for current offshore wind farms and serves as a basis for comparison. It is located in the German EEZ at the site of Alpha Ventus (see Fig. 3.1) and consists of 80 NREL reference wind turbines with a rated power of 5 MW each. The turbine power curve (see Fig. 3.1 right) is later on used to calculate losses in production due to maintenance work. The commissioning date
of the wind farm is assumed to be January, 2014, which is considered when calculating the feed-in-tariff. Water depths in the German EEZ reach up to about 90m, whereas the Fraunhofer reference wind farm is located at depths between 30 and 40m. The wind turbine spacing is on average approximately 1.5 km. The service port for large components is defined to be the port of Bremerhaven, which is approximately 140 km away from the wind farm site.

![Fraunhofer reference wind farm location](left) and NREL reference wind turbine power curve (right)

### 3.2 Vessels and component replacement procedures

The paper compares two vessel strategies: jack-up and floating. Since rates and characteristics of vessels can vary significantly, both vessels chosen for the comparison are generic and represent a category rather than specific existing vessels. Table 3.2 lists their characteristics.

<table>
<thead>
<tr>
<th>Type</th>
<th>Deck Space</th>
<th>Payload</th>
<th>Crane</th>
<th>Day Rate</th>
<th>Mob</th>
<th>Demob</th>
<th>max. speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>DP 2 Vessel with AHC Crane (Active Heave Compensation)</td>
<td>3100</td>
<td>13000</td>
<td>1800</td>
<td>38000</td>
<td>65000</td>
<td>65000</td>
<td>12</td>
</tr>
<tr>
<td>2 tugs + jack-up barge medium non self propelled</td>
<td>2500</td>
<td>2300</td>
<td>1000</td>
<td>43000</td>
<td>108000</td>
<td>108000</td>
<td>9</td>
</tr>
</tbody>
</table>
For the sake of simplicity and clarity replacement procedures have been generalized for all large components. Different vessel types, however, require different operation sequences. The analysis, therefore, considers two replacement procedures depending on the vessel used (see Table 3.2 and Table 3.3). The durations for the in- and outbound trips are estimated using 75% of the maximum speed of each vessel in order to account for decreased speed due to tidal currents, wave heights and onboard payloads.

Table 3.2: Jack-up vessel component replacement procedure

<table>
<thead>
<tr>
<th>No.</th>
<th>Task Name</th>
<th>Duration</th>
<th>Restrictions</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>outbound trip</td>
<td>11h</td>
<td>Hs ≤ 1.5m</td>
</tr>
<tr>
<td>2</td>
<td>vessel positioning</td>
<td>1 h</td>
<td>Hs ≤ 1m</td>
</tr>
<tr>
<td>3</td>
<td>jack-up</td>
<td>2 h</td>
<td>Hs ≤ 1m</td>
</tr>
<tr>
<td>4</td>
<td>preloading</td>
<td>3 h</td>
<td>none</td>
</tr>
<tr>
<td>5</td>
<td>crew + equipment transfer</td>
<td>2 h</td>
<td>none</td>
</tr>
<tr>
<td>6</td>
<td>component dismantling</td>
<td>6 h</td>
<td>none</td>
</tr>
<tr>
<td>7</td>
<td>lower component on deck</td>
<td>2 h</td>
<td>U ≤ 7m/s</td>
</tr>
<tr>
<td>8</td>
<td>attach crane gear to new component</td>
<td>1 h</td>
<td>none</td>
</tr>
<tr>
<td>9</td>
<td>install new component</td>
<td>8 h</td>
<td>U ≤ 7m/s</td>
</tr>
<tr>
<td>10</td>
<td>final bolting</td>
<td>2 h</td>
<td>U ≤ 7m/s</td>
</tr>
<tr>
<td>11</td>
<td>crew + equipment transfer</td>
<td>2 h</td>
<td>none</td>
</tr>
<tr>
<td>12</td>
<td>jack down + pulling legs</td>
<td>3 h</td>
<td>Hs ≤ 1m</td>
</tr>
<tr>
<td>13</td>
<td>inbound trip</td>
<td>11h</td>
<td>Hs ≤ 1.5m</td>
</tr>
</tbody>
</table>

Table 3.3: Floating vessel component replacement procedure

<table>
<thead>
<tr>
<th>No.</th>
<th>Task Name</th>
<th>Duration</th>
<th>Restrictions</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>outbound trip</td>
<td>8 h</td>
<td>Hs ≤ 2m</td>
</tr>
<tr>
<td>2</td>
<td>vessel positioning</td>
<td>1 h</td>
<td>Hs ≤ 2m</td>
</tr>
<tr>
<td>3</td>
<td>ballasting</td>
<td>1 h</td>
<td>Hs ≤ 2m</td>
</tr>
<tr>
<td>4</td>
<td>crew+ equipment transfer</td>
<td>2 h</td>
<td>Hs ≤ 1.5m</td>
</tr>
<tr>
<td>5</td>
<td>component dismantling</td>
<td>6 h</td>
<td>none</td>
</tr>
<tr>
<td>6</td>
<td>lower component on deck</td>
<td>2 h</td>
<td>Hs ≤ 2m</td>
</tr>
<tr>
<td>7</td>
<td>attach crane gear to new component</td>
<td>1 h</td>
<td>none</td>
</tr>
<tr>
<td>8</td>
<td>install new component</td>
<td>8 h</td>
<td>Hs ≤ 2m; U ≤ 7m/s</td>
</tr>
<tr>
<td>9</td>
<td>final bolting</td>
<td>2 h</td>
<td>Hs ≤ 2m; U ≤ 7m/s</td>
</tr>
<tr>
<td>10</td>
<td>crew+ equipment transfer</td>
<td>2 h</td>
<td>Hs ≤ 1.5m</td>
</tr>
<tr>
<td>11</td>
<td>ballasting</td>
<td>1 h</td>
<td>Hs ≤ 2m</td>
</tr>
<tr>
<td>12</td>
<td>inbound trip</td>
<td>8 h</td>
<td>Hs ≤ 2m</td>
</tr>
</tbody>
</table>
3.3 Weather Conditions

Weather conditions have a large impact on the duration of repair procedures as well as on component failures which can have high implications on costs. Their seasonal effect causes non-linear behavior in uncertainties and costs. Both significant wave height and wind speed are taken into consideration in the analysis.

The time series for the evaluation in this paper originates from the hindcast data CoastDat v1 by HZG (Hemhholz Zentrum Geesthacht) and encompasses 50 years of data. For the sake of simplicity, the environmental conditions are taken from one geographical point (54N; 6.6E), which is the model grid point in closest proximity to the center latitude and longitude of the wind farm. This assumption can be made since most of the activities during component replacement procedures take place within the boundaries of the wind farm. Moreover, weather conditions far from shore (at the wind farm site) are generally “rougher”. The assumption, thus, would lead to slightly more conservative results, which is a frequently preferred approach.

4 Seasonal Failure Statistics

Fig. 4.1 shows reliability characteristics annual failure frequency and downtime per failure. Obviously, the electrical subassemblies fail more often than mechanical ones, while mechanical subassemblies experience longer downtimes after a failure.

An assumption made in the analysis shown in Fig. 4.1 is that all turbines observed in the Fraunhofer IWES failure database WMEP are weighted equally. No differentiation concerning size of turbine, technical concept, external conditions etc. has been done here. Nevertheless the figure gives a good overview on the relationship between failure frequencies and downtimes. Although large components such as drive train and rotor blades fail very seldom, they account for a large amount of downtimes. The WMEP data is especially concerned with onshore wind turbines but due to lack of databases, in a first approximation, it can be used to deduce failures of components in offshore wind farms.
Hahn [4] has shown that the failure rate increases with higher wind-speeds even when investigating daily data. In his analyses failure rates of subassemblies of the electrical system showed the strongest dependency from wind-speed. For other non-electrical subassemblies the dependency of failure rate on wind-speed, even though weaker, was still present. The dependency of failure rate on wind speed is shown in Fig. 4.1. This correlation further propagates in more failures during adverse weather conditions and influences maintenance costs. It is therefore important to take into consideration the weather conditions and their seasonal effect on both failure rates as well as maintenance procedures. Fig. 4.2 shows the seasonal (monthly) dependences of the frequencies of failure rate for four large components (drive train, gearbox, generator and blades). In general most failure rates occur in winter, but in detail this assumption doesn’t hold for all considered (especially drive train and generator). This could be explained by the fact that the harsh winter weather conditions indeed influence the degradation process of those components but the results of this process take place weeks or even months after the first catastrophic signal. This means that start of the degradation could be in winter but the ultimate occurrence of the failure takes place in the summer.

\[ f(t) = \lambda e^{-\lambda t} \]  
(1)

The cumulative probability function can be described by the following equation:

\[ F(t) = \int_0^t f(\tau) \, d\tau = \int_0^t \lambda e^{-\lambda \tau} \, d\tau = 1 - e^{-\lambda t} \]  
(2)

Whether or not a failure occurs in a particular time step can be simulated by generating a uniform random number and cross-checking it with the calculated cumulative probability for a certain month. If the random number is smaller than the cumulative probability, a failure occurs. In a consequent step this failure triggers a maintenance procedure. The weather impact on the procedure is simulated using the WaTSS method resulting into extended durations of replacement.
procedures. If a second failure occurs while replacement procedure is being performed, it is postponed until the vessel is available and executed sequentially.

The durations of replacement procedures for both jack-up barge and floating vessel are shown in Fig. 5.1. The red lines indicate the 5%, 50% and 95% quantiles. The impact of adverse weather conditions on duration as well as on duration uncertainty can be seen in both vessel strategies. The floating concept, however, requires less time than its jack-up counterpart.

6 Cost Comparison

The costs for both vessel strategies are compared on a monthly basis in order to show seasonal effects. Both costs due to vessel rental and losses in production are considered.

The costs due to losses in production are calculated using the NREL reference wind turbine power curve (see Fig. 3.1 right). According to the German Renewable Energy Act and considering the commissioning data of the wind farm as well as the water depth at the location (see chapter 3.1), the weighted average feed-in-tariff over 20 years of operation amounts to 13.22 Cents/kWh.

Fig. 6.1 shows a comparison in costs of the jack-up (blue) and floating vessel strategy (red). The costs due to vessel rental constitute a major part of the total costs (light blue and red). The overall costs are much higher in winter than in summer due to adverse weather impact on maintenance procedure durations, higher losses in production (dark blue and red) as well as higher failure rates in winter. Overall, given the assumptions in this paper the floating strategy would be the preferred one since it generates considerably less costs over the life cycle of the wind farm.
Fig. 6.1: Cumulated monthly costs over 20 years of operation for jack-up (blue) and floating vessels (red)

7 Summary

A comparison of two vessel strategies (jack-up and floating) for the replacement of large components in offshore wind turbines has been conducted. The analysis in this paper emphasizes on the strongly seasonal behavior of failure rates, maintenance work durations and production losses and thus employs a time-based approach (WaTSS) on multiple-year weather time series. The results confirm a considerable dependence of costs on the season and establish the floating strategy as the clearly less expensive strategy, given the assumptions. The paper shows that a comprehensive analysis is crucial for the selection of an optimal vessel strategy.

8 References


INFLUENCE OF SOIL RESISTANCE APPROACH ON OVERALL STRUCTURAL LOADING OF LARGE DIAMETER MONOPILES FOR OFFSHORE WIND TURBINES

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Keywords: Monopiles, p-y method, Loading, Dynamics, Nonlinear Integrated Simulation

Summary: Recently enhanced manufacturing capacities for large diameter piles strengthen the promising trend of realizing monopiles at sites with water depths of 30m and more. The design of monopiles is anticipated to be driven by fatigue loading or governed by the lower rotor speed boundary in natural frequency analysis. Both criteria are closely related to structural stiffness to which the foundation stiffness contributes decisively. Simultaneously the application of existing p-y approaches to large diameter piles lacks of adequate experimental or practical validation. This paper presents a study on different modeling approaches of soil-structure interaction and its impact on overall structural loading on fatigue load level. Three p-y approaches as well as advanced finite element based calculations are incorporated into an integrated load simulation tool. An efficient coupling approach is applied which empowers the simulation tool to consider complex soil conditions with reasonable numerical efforts. Bedding stiffnesses derived by p-y approaches or sophisticated 3D finite element pile-soil models are thereby taken into account with high accuracy and efficiency.

1 INTRODUCTION

Within the next decades an enormous number of Offshore Wind Turbines (OWT) with capacities larger than 5MW shall be erected in wind farms in the North- and Baltic Sea. Monopiles are currently the preferred foundation concept in water depths up to 30m. Thus, pile diameters up to 8m are necessary. In the design, structural stiffness becomes an important aspect in order to keep the required distance between the eigenfrequencies of the wind turbine to the main excitation frequencies of the operating rotor nacelle assembly and waves. Thereby, the foundation stiffness contributes decisively to the global system stiffness. Conservative stiffness assumptions lead not necessarily to a conservative structural loading. Concept studies show large diameter designs may facilitate monopiles in about 40m water depth at North Sea locations [1, 2, 3]. Recently updates of pipe manufacturing facilities empower this trend. This paper deals with integrated simulations, the subsoil resistances and the soil-structure interactions of a 5MW offshore wind turbine (OWT) supported by a monopile reference structure with 8m base diameter in 40m North Sea depth.
2 DESIGN REFERENCE, APPROACH AND PARAMETERS

a. Design basis and reference design

The considered environmental conditions are typical for the selected location with 40 m water depth at a German North Sea site more than 100 km offshore north of the island Borkum. The wind and wave data was interpolated from a number of meteo-marine hindcast data available to the authors and was lumped to 23 wind bins from 3 to 25m/s with related couples of wave height and period. A constant wind shear exponent $\alpha$ of 0.14 is assumed. The turbulence intensity assigned to each wind-wave bin is determined following the assumptions made for the K13 deep water site in the UpWind Design Basis (cf. [4], p. 111). Table 1 presents an extraction of the pre-mentioned parameters. More comprehensive wind and wave data was used to design the reference monopile presented in this paper. A significant wave height of 11.0m with a zero-crossing period from 12 to 15s was considered. The overall design goal was a very slender monopile design for larger water depths by use of state-of-the-art simulations and design techniques which outlines potential limits of optimization.

Table 1: Extract of the considered wind-wave scatter data

<table>
<thead>
<tr>
<th>$v_{hub}$</th>
<th>$\alpha$</th>
<th>$I$</th>
<th>$H_s$</th>
<th>$T_z$</th>
<th>$h$</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>0.14</td>
<td>0.233</td>
<td>0.50</td>
<td>4</td>
<td>0.090</td>
</tr>
<tr>
<td>5</td>
<td>0.14</td>
<td>0.219</td>
<td>0.75</td>
<td>4.5</td>
<td>0.150</td>
</tr>
<tr>
<td>6</td>
<td>0.14</td>
<td>0.187</td>
<td>1.25</td>
<td>5</td>
<td>0.100</td>
</tr>
<tr>
<td>8</td>
<td>0.14</td>
<td>0.160</td>
<td>1.50</td>
<td>5.5</td>
<td>0.210</td>
</tr>
<tr>
<td>11</td>
<td>0.14</td>
<td>0.148</td>
<td>2.50</td>
<td>6.5</td>
<td>0.260</td>
</tr>
<tr>
<td>13</td>
<td>0.14</td>
<td>0.144</td>
<td>3.50</td>
<td>7.5</td>
<td>0.110</td>
</tr>
<tr>
<td>14</td>
<td>0.14</td>
<td>0.142</td>
<td>4.50</td>
<td>8.5</td>
<td>0.030</td>
</tr>
<tr>
<td>17</td>
<td>0.14</td>
<td>0.137</td>
<td>5.50</td>
<td>8.5</td>
<td>0.050</td>
</tr>
<tr>
<td>19</td>
<td>0.14</td>
<td>0.136</td>
<td>6.50</td>
<td>9.5</td>
<td>0.010</td>
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<tr>
<td>20</td>
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<td>0.134</td>
<td>7.50</td>
<td>10.5</td>
<td>0.002</td>
</tr>
<tr>
<td>22</td>
<td>0.14</td>
<td>0.133</td>
<td>8.50</td>
<td>11.5</td>
<td>&lt;0.001</td>
</tr>
<tr>
<td>23</td>
<td>0.14</td>
<td>0.132</td>
<td>9.50</td>
<td>12.2</td>
<td>&lt;0.001</td>
</tr>
<tr>
<td>24</td>
<td>0.14</td>
<td>0.131</td>
<td>10.50</td>
<td>13.5</td>
<td>&lt;0.001</td>
</tr>
<tr>
<td>25</td>
<td>0.14</td>
<td>0.131</td>
<td>11.50</td>
<td>15</td>
<td>&lt;0.001</td>
</tr>
</tbody>
</table>

$m/s$ [-] $[-]$ $m$ $s$ $rel.$ $[-]$

$V_{hub}$ wind speed at hub height, $\alpha$ wind shear, $I$ turbulence intensity, $H_s$ significant wave height, $T_z$ zero up-crossing period, $h$ relative occurrence

Figure 1: Monopile Model Figure 2: Discretized system

For detailed information on directional distribution, related probabilities, possible misalignments and detailed dimensions of the monopile refer to [3]. The numerical study presented is strictly focused on the influence of different bedding resistance approaches on the overall structural loading. Wind and waves are considered as co-directional. The reference
monopile was developed assuming homogenous, dense sand with an inner friction angle $\varphi'$ of 37.5°. A p-y-based subsoil resistance was modeled following API [5].

b. Reference turbine model with tower, discretization and coupled simulation tool

A nacelle height of 89.0m including a vertical nacelle offset of 2.40m from tower top flange is presumed for the 5MW OWT (cf. Fig. 1). The NREL 5MW baseline OWT is used as turbine model [6]. The tower design has not been optimized for this study. It is rather regarded as a reasonable assumption for the tower stiffness. The integrated load simulations of the operating OWT with support structure and foundation model are performed with the coupled aero-hydro-servo-elastic tool Flex 5 (courtesy to Senvion) and Poseidon (in-house tool of the Institute for Steel Construction). Flex 5 covers the turbine model including the tower. Poseidon manages the monopile model from tower bottom down to the pile base. Two options to model the foundation have been implemented recently which are described in the following. In order to include the nonlinear characteristic of the foundation models, a nonlinear solver in time domain has been added to the Poseidon code.

c. Soil parameter and modeling approach for structure-foundation-interaction

The reference monopile is driven 35m into the seabed (cf. Fig. 1). As first option, the soil response is modeled with nonlinear springs implemented along the pile penetration depth (cf. Fig. 2, 3). The nonlinear stiffness relations are derived from three different p-y approaches. The general relation between lateral pile deflection $y$ and soil resistance $p$ is shown in Fig. 4 (cf. [7]). The initial slope of a p-y curve is represented by the initial bedding stiffness $E_{py}$ which governs the course of the p-y curve, basically described with equation (1) [7].

$$p = A p_u \tanh \left( \frac{E_{py}}{A p_u} y \right)$$

(1)

with calibration factor $A$, maximum bedding resistance $p_u$, initial bedding stiffness $E_{py}$ (in case of API: $E_{py} = k \cdot z$, with $k$ as initial stiffness coefficient and $z$ as depth below ground surface) and pile deflection $y$ at a given depth $z$. 

![Figure 3: Pile with p-y springs](image)

![Figure 4: General comparison of existing p-y approaches](image)

![Figure 5: Integrated Flex 5-Poseidon model with p-y springs](image)
At first, the most common p-y approach according to the recommendations of API [5] is applied. The p-y curve recommended by API were validated against measurements of piles with diameter less than 1 m, which is contraire to nowadays monopile dimensions. As a consequence, two further p-y approaches are considered, which have been developed in 2012 by Sørensen [9] and Kallehave et al. [10]. Both approaches modify the initial bedding stiffness $E_{py}$. Sørensen (Søe) focussed on the pile behaviour under characteristic extreme loads decisive for the SLS design proof (serviceability limit state) and reduced $E_{py}$ compared to API. Kallehave et al. (Kal) evaluated measurements of an OWT monopile under operation and determined higher bedding stiffness than API, adequate for typical fatigue load scenarios. The characteristics of the pre-mentioned p-y approaches are comprehensively outlined in [7]. The first part of the study compares the fatigue loading of the OWT monopile using the three p-y approaches, included within the Poseidon part of the integrated OWT model (Fig. 5).

As second option, the foundation response is derived from numerous pre-calculations with a 3D finite element (FE) pile-soil model (Fig. 6). The stress-dependent and shear-strain dependent soil stiffness are captured by use of the sophisticated HSsmall soil model within the 3D FE-Software PLAXIS 3D. This approach avoids typical shortcomings of p-y approaches, such as inaccuracies in case of layered soil, the limitation to the validity range of pile diameter or load level dependence of a certain approach. For an efficient pre-analysis of the pile response at mudline and a later coupling to dynamic time domain load simulations, the expected load spectra of bending moments $M_{mud}$ and lateral loads $H_{mud}$ at mudline must be known. The results can be introduced to the integrated load simulation tool as discrete coupled translational and rotational springs ($K_{sec}$, $K_{rot}$) at mudline (Fig. 7). The spring stiffnesses are derived from coupled pairs of loads $H_{mud}$, $M_{mud}$ and related deformations of the 3D FE simulation, respectively pile displacement $u_{mud}$ and rotation $\Theta_{mud}$. Flex 5 - Poseidon treats the truncated OWT monopile system with a support node coupled to the nonlinear pile stiffness as nonlinear support node, which saves approximately a third of degrees of freedom.
of the entire load simulation model (Fig. 8). This allows for an accurate FE-based representation of the soil-structure interaction within a nonlinear load simulation while a significant numerical efficiency is achieved at the same time. The second part of the study compares the fatigue loading of the OWT monopile using the widespread API p-y approach with the FE-based pile response, introduced to the truncated OWT monopile system as a database of results from standalone pre-calculation of the pile-soil models.

d. Simulation models and soil parameters

Besides the variation of foundation models, the soil parameters considered are valid for medium dense as well as very dense sand with inner friction angles $\phi'$ of 35° and 40°. The configurations are summarized in Table 2. For detailed input parameters of the FE soil models refer to [8]. The selected soil configurations vary just slightly from the monopile design basis ($\phi' = 37.5^\circ$, see Sec. 2a, Fig. 1) which justifies the use of the same reference design.

<table>
<thead>
<tr>
<th>Soil</th>
<th>Study Part 1 Foundation Models</th>
<th>Study Part 2 Foundation Models</th>
</tr>
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<tr>
<td>Sand</td>
<td>Flex 5-Poseidon</td>
<td>Flex 5-Poseidon with nonlinear support node with API and FE-based stiffness</td>
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<td>$\phi' = 35^\circ$</td>
<td>p-y approaches</td>
<td>API, $S_\phi$, Kal</td>
</tr>
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<td>$\gamma' = 9.75 \text{kN/m}^3$</td>
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<td>API and FE-based stiffness</td>
</tr>
<tr>
<td>Sand</td>
<td>Flex 5-Poseidon</td>
<td>Flex 5-Poseidon with nonlinear support node with API and FE-based stiffness</td>
</tr>
<tr>
<td>$\phi' = 40^\circ$</td>
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<td>API, $S_\phi$, Kal</td>
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<tr>
<td>$\gamma' = 10.31 \text{kN/m}^3$</td>
<td>API, $S_\phi$, Kal</td>
<td>API and FE-based stiffness</td>
</tr>
</tbody>
</table>

3  Simulation Results of Pre-Calculation and Integrated OWT Analysis

a. Modal analysis and fatigue evaluation with p-y approaches

At first, the eigenfrequencies of the entire OWT monopile system including the p-y springs along the subsoil monopile section are analyzed. In order to consider the entire spectrum of fatigue loading, the structure is pre-stressed with a combined wind and wave load level representative for the upper limit of loads under turbine operation. It has been observed that global eigenfrequencies of the OWT are insensitive due to variation of loading within the fatigue load spectrum. More important, Fig. 9 shows the scatter of the first global eigenfrequency (1st Glob. EF) due to the two inner friction angles $\phi'$ of the sand as well as the three p-y approaches. One should notice that the scatter of the 1st Glob. EF due to medium dense or very dense soil is approximately 27 % less than due to different p-y approaches. Overall, the variation of the 1st Glob. EF leads to smaller deviations for a very slender monopile system, considered in this study, than for stiffer monopile designs without applying state-of-the-art simulation and design techniques. A transfer of results to monopiles in general is not possible.

Secondly, the fatigue damage in the monopile has been evaluated for a lifetime of 20 years which is exemplarily shown in Fig. 10. Results are normalized to the API approach. The fatigue damage of the Kallehave-System is merely 2% less than API. Due to its conservative
stiffness estimation on fatigue load level, Sørensen leads to a significant increase of fatigue damage by 24%. For the second part of the study, the API results will serve as reference.

Figure 9: 1st global eigenfrequency of the p-y spring models (API, Kallehave (Kal), Sørensen (Søe))

Figure 10: Normalized fatigue damage at mudline of the p-y spring models with medium dense sand (reference API)

b. Resulting pile stiffness from pre-simulations of standalone pile-soil models

A conservative estimation of the overall load spectrum is made in order to capture all load configurations for a subsequent time domain simulation with a pile-soil representation as a nonlinear support node (cf. Fig. 11). Each nonlinear FE-based simulation has been run increasing the horizontal force with constant vertical eccentricity and is represented by each dotted line in Fig. 11. The resulting coupled spring stiffnesses $K_{sec}$, $K_{rot}$ of the standalone API and FE-based pre-calculations are shown as nonlinear stiffness surfaces in Fig. 12. The green line bounds the combined $H_{mud}$, $M_{mud}$ load level where the FE model leads to higher stiffness which is amongst other effects induced by small strains. In order to maintain the coupling of $u_{mud}$ and $\varphi_{mud}$, the subsequent nonlinear support node simulations are strictly fed with related pairs of $K_{sec}$, $K_{rot}$ for specific combinations of $H_{mud}$, $M_{mud}$.

c. Fatigue evaluation with nonlinear support node approach

In order to increase accuracy and efficiency, the coupled nonlinear stiffness surfaces have been applied for the nonlinear support node model implemented in Poseidon (cf. Fig. 8). A limited number of 16 pre-simulations serves as reasonable number of mesh points for the stiffness surface (positive M) with pairs $H_{mud}$, $M_{mud}$ of the same sign (Fig. 12). $H_{mud}$, $M_{mud}$ with opposite signs lead to a stiffness surface (negative M) with a singularity, supported by 8 additional presimulations (Fig. 14). The smooth stiffness surfaces (positive M) allow for highly accurate interpolations. The interpolation accuracy with the stiffness surface (negative M) is decreasing nearby the singularity. Fig. 13 shows the fatigue damage and damage equivalent loads at mudline. The results of API p-y and API support node simulations differ by 10% in terms of fatigue. This is caused by minor remaining transients due to the imperfect mesh transition in the stiffness surface (negative M). Nevertheless, a good agreement of the
load time series in terms DEL (+/- 3%) between all three cases backs the support node approach as a promising technique to introduce FE-based pile models into nonlinear load simulation models.

Figure 11: Load spectrum for standalone pile pre-simulations

Figure 12: Coupled stiffness surfaces from FE-Model (left) and derived with API p-y approach (right)

Figure 13: Normalized fatigue damage/DEL at mudline of the API p-y spring model vs. the support node approach with API and FE-based results with medium dense sand

Figure 14: Uneven transition in the stiffness surface from lower stiffness to the singularity with extraordinary high stiffness in case of no deformation under load pairs H, -M
4 CONCLUSIONS

The governing first global eigenfrequency of the reference OWT with large diameter monopile scatters at least in the same magnitude of order due to a variation of the p-y approach or a change from medium to very dense sand. The comparison of the p-y approaches within the integrated load simulations showed a good agreement of fatigue loading between the API approach and the one from Kallehave et al.. The slender reference monopile leads to a reduced influence of the foundation model. A 3D visualization of pre-calculated foundation stiffnesses at mudline allows for a better comparison of FE-based and p-y based approaches. The coupling of FE-based pile pre-simulations has been successfully implemented, though the underlying interpolation scheme should be improved in case of singularities in pile stiffness surfaces. In terms of performance the coupling scheme is almost able to outperform load simulations with linearized p-y springs along the pile penetration length. An extended parameter set is required to transfer the presented results to state-of-practice monopile designs. Different water depths, pile diameters and wall thicknesses and the additional consideration of wind-wave misalignment is suggested for future research.

5 REFERENCES

NUMERICAL APPROACH FOR THE DERIVATION OF INTERACTION DIAGRAMS FOR PILES UNDER CYCLIC AXIAL LOADING

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Keywords: Offshore Structures, Finite Element Method, Interaction Diagram, Cyclic Loading

Summary: Offshore wind energy plants promise to become an important source of energy in the near future. It is expected that within 10 years, wind parks with a total capacity of thousands of Megawatts will be installed in European seas. Two foundation concepts which will be used in this field are the jacket and the tripod. Under the wind and waves action, the piles supporting these structures will be subjected to highly cyclic tension or compressive axial loading.

A numerical model of the pile-soil system using the finite element system ABAQUS was developed in order to simulate the pile behavior under cyclic axial loading. In this model the material behavior of the subsoil is described using an elasto-plastic constitutive model with Mohr-Coulomb failure criterion.

The numerical approach was performed in three stages. In the first stage, the static pull-out capacity of the pile was determined. In the second stage, from the first loading and unloading steps, the shear strain amplitude in each element is calculated and the expected volume compaction under the considered number of load cycles is predicted. This volume compaction is applied to the finite element model. The result is a reduction of the skin friction acting along the pile. In the third stage, the (reduced) post-cyclic capacity of the pile is determined.

The calculation results for piles with different dimensions embedded in sand are presented and compared to existing interaction diagrams. These results are used to examine the overall characteristics of tension piles response under cyclic loading.

1 INTRODUCTION

Cyclic behavior of axially loaded piles has been an issue since studies revealed that the capability of transferring skin friction into the ground is highly dependent on the level and the number of load cycles. In many cases, the foundations of offshore wind energy converters will consist of jacket and tripod structures, which are supported by steel pipe piles located at the edges of the foundation area. These piles will have diameters between 1.50 m and 3.0 m and are loaded by cyclic compression and tension loads.
Due to the strong wind and wave actions, the foundation piles are often subject to intensive cyclic loads. Although numerous studies have been carried out regarding pile capacity under cyclic loading, no approved calculation approach exists yet to determine the decrease of pile capacity dependent on cyclic load magnitude and number of load cycles. This paper describes a numerical approach which principally facilitates the derivation of an interaction diagram and the calculation of the pile capacity decrease dependent on the number of load cycles for a certain pile-soil system. The approach is an advancement of the method described before in Abdel-Rahman & Achmus (2011a & 2011b).

2 STATE OF THE ART

Regarding axial cyclic loading, it is known that the pile capacity in general decreases with increasing number of load cycles. This occurs mainly due to a decrease of ultimate skin friction, whereas the tip resistance is believed to remain almost unchanged. Poulos (1988) presented a cyclic stability diagram for axially loaded piles. Based on several model and field tests, the load cycle numbers leading to pile failure are given dependent on the normalized mean load $E_0/R_k$ and the cyclic load amplitude $E_{cyc}/R_k$. A “stable” region is defined here with $E_{cyc}/R_k \leq 0.2$ (i.e. $CLRL \leq 0.2$) for $E_0 \leq 0.6 \ R_k$.

Kempfert (2009) evaluated pile test results in cohesive and non-cohesive soils separately and proposed the following approach to determine the critical cyclic load amplitude:

$$\frac{E_{cyc}}{R_k} \leq \kappa^* \cdot \left(1 - \left(\frac{E_0}{R_k} + 0.65 - \kappa^*\right)^4\right)$$

The factor $\kappa^*$ is dependent on the number of load cycles and the type of soil and is given in the following table:

<table>
<thead>
<tr>
<th>Number of load cycles N</th>
<th>$10^1$</th>
<th>$10^2$</th>
<th>$10^3$</th>
<th>$10^4$</th>
<th>$10^5$</th>
<th>$10^6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\kappa^*$ non-cohesive</td>
<td>0.43</td>
<td>0.38</td>
<td>0.33</td>
<td>0.28</td>
<td>0.23</td>
<td>0.18</td>
</tr>
<tr>
<td>$\kappa^*$ cohesive</td>
<td>0.48</td>
<td>0.43</td>
<td>0.38</td>
<td>0.33</td>
<td>0.28</td>
<td>0.23</td>
</tr>
</tbody>
</table>

The German ‘EA Pfähle’ recommendation (EAP 2012) presents formalized stability or interaction diagrams which can be used in the design, for instance the interaction diagram proposed by Kirsch et al. (2010) shown in Figure 1. Here the normalized mean load is termed $X_{mean}$ and the normalized cyclic load amplitude $X_{cyc}$:

$$X_{cyc} = \frac{E_{cyc}}{R_k}; X_{mean} = \frac{E_0}{R_k}$$
The interaction diagram yields the number of load cycles $N_f$ leading to failure for a certain combination of $X_{mean}$ and $X_{cyc}$. These interaction diagrams do not consider the effect of pile dimensions (length & diameter) or the relative density of the surrounding soil.

3 COMPACTION OF SANDY SOIL DUE TO CYCLIC SHEARING

The main idea of the proposed method is to predict the volume compaction of the soil elements beside a pile induced by cyclic shearing and to consider this volume change in the numerical simulation of pile behavior. The compaction effect due to shear strain was investigated previously by Silver & Seed (1971) by performing cyclic simple shear tests on sand. Their results can be presented as a relationship between the volume reduction due to compaction ($\varepsilon_c$), the load cycle number $N$ and the cyclic shear strain $\gamma_{xy}$ (see Figure 2). It was found that the increase of volume compaction with number of load cycles is dependent on the amplitude of the cyclic shear strain applied.

![Figure 1: Interaction Diagram of Kirsch et al. (2010)](image)

![Figure 2: Cyclic vertical compaction of sand for different shear strain amplitudes and vertical stresses (Silver & Seed 1971)](image)
In the previous tests, the initial relative density of the sand samples was also varied. Evaluating the results of Silver & Seed (1971), the following equations were derived by regression analyses to describe the observed dependency of volume strain on shear strain amplitude, number of load cycles and relative density:

Evaluating the results for N=10, Equations (3) and (4) were derived:

$$\varepsilon_c(N = 10) = 10^{\chi+1.23\log\gamma_{xy}}$$  (3)

Here $\chi$ presents the dependency on the relative density ($D_r$) as follows:

$$\chi(D_r) = 1.71 - 2.82 \cdot D_r + 0.94 \cdot D_r^2$$  (4)

The dependency on load cycle number N can be described by the following equation:

$$\varepsilon_c = \left(10^{\chi} \cdot 10^{1.23 \log\gamma_{xy}}\right) \cdot (0.3 + 0.7 \cdot \log N)$$  (5)

In the ‘small strain’-shear range soil behaves almost like an elastic material, and no volume compaction is to be expected (e.g. Vucetic 1994). Therefore, if the shear strain amplitude $\gamma_{xy}$ falls below a threshold strain $\gamma_{lim}$, $\varepsilon_c$ can be set to zero. The magnitude of the threshold strain is dependent on the soil type (Vucetic 1994). In our investigations, the threshold strain was assumed to be $\gamma_{lim}=5\cdot10^{-5}$. These derived equations are stress independent and do not catch accurately the high deviations obtained in the experimental work done by Silver & Seed.

4 NUMERICAL SCHEME

a. Finite Element model

A two-dimensional (2D-axisymmetric) finite element model to investigate the axial deformation response of a pile in sandy soil was developed. The elements used to model the soil are 4-noded axisymmetric (CAX4) elements. Close to the pile, a very fine discretization was applied in order to get more accurate results. The numerical modelling was performed for a steel tube pile with length of 30.0 m, diameter of 2.0 m and a wall thickness of 3.0 cm. The boundaries of the mesh were at a radius of 40.0 m beside the pile axis and 50.0 m below the base of the pile. With these dimensions, it was verified that the behavior of the pile is not affected by the boundary conditions.

A linear elastic material behavior of the piles was assumed with the parameters $E = 2.10\cdot10^5$ MN/m$^2$ (Young’s modulus) and $\nu = 0.20$ (Poisson’s ratio) for steel. For simplification, a homogeneous solid pile cross section was modeled. The elastic parameters were adapted to ensure axial pile stiffness equal to the stiffness of the steel tube.

To account for the non-linear soil behavior, elasto-plastic material behavior was assumed for the soil elements. A Mohr-Coulomb failure criterion (parameters $\phi'$, $c'$, $\psi$) was considered. The material parameters typical for medium dense sand used here are given in Table 2.

For the contact behavior of the surface between pile (master) and soil (slave) an elasto-plastic model was used. The maximum frictional shear stress is dependent on the normal stress $\sigma_n$ and a coefficient of friction $\mu$. In the numerical simulations presented here $\mu = 0.431$ was assumed ($\mu = \tan (2/3 \phi')$). For full mobilization of the limit frictional stress the relative
displacement (elastic slip) between the pile and the surrounding soil was set to \( \Delta u_{\text{el,slip}} = 2 \text{ mm} \).

| Buoyant unit weight \( \gamma' \) | 10.0 kN/m\(^3\) |
| Oedometric stiffness Modulus | 120.0 MN/m\(^2\) |
| Poisson’s ratio \( \nu \) | 0.25 |
| Internal friction angle \( \phi' \) | 35.0° |
| Dilation angle \( \psi \) | 5.0° |
| Cohesion \( c' \) | 1.0 kN/m\(^2\) |

b. Modeling stages

The numerical modeling was performed in several stages. In each stage, the first step was the initialization of geostatic stresses in the soil mass by activating only the soil elements. Afterwards, the soil elements located at the pile position were replaced by pile steel elements and the pile own weight and contact conditions between the pile and the surrounding soil were activated.

In the first stage, the tensile load was increased gradually until failure occurred. From this stage, the load-heave curve and the static pull-out capacity \( R_k \) were determined.

In the second stage, the pile is loaded up to a load \( F_{\text{max}} \) and then unloaded to a load \( F_{\text{min}} \) which should be lower than the pull-out capacity (\( R_k \)). For each soil element the maximum and minimum shear strains \( \gamma_{\text{max},i} \) and \( \gamma_{\text{min},i} \) are determined. The shear strains \( (\gamma_{\text{max},i}, \gamma_{\text{min},i}) \) are calculated from the principal strain components \( \varepsilon_1, \varepsilon_2 \) and \( \varepsilon_3 \) by means of Equation (7) (see Wegener & Herle 2010):

\[
\gamma = \sqrt[3]{\frac{2}{3} [(\varepsilon_1 - \varepsilon_2)^2 + (\varepsilon_2 - \varepsilon_3)^2 + (\varepsilon_1 - \varepsilon_3)^2]}
\]

The cyclic shear strain amplitude in each element is then obtained by \( \gamma_{xy,i} = (\gamma_{\text{max},i} - \gamma_{\text{min},i})/2 \).

Considering a certain number of load cycles \( N \), the volume compactions \( \varepsilon_{c,i} \) for each element can be calculated using Eq. no. (5).

In the third stage, the pile is again loaded to \( F_{\text{max}} \) and then these volume compactions are applied to the pile-soil system. The desired compaction or shrinkage of the elements was realized by applying temperature differences only in the horizontal direction in the numerical model which induce the desired volume compaction. The result is a redistribution of stresses in the system; in particular a decrease of the horizontal stresses acting on the pile under the applied load \( F_{\text{max}} \). Subsequently, the tensile load is further increased gradually until failure occurs. The cyclic pull-out capacity \( R_{k,N} \) or the cyclic capacity decrease \( \Delta R_{k,N} = R_k - R_{k,N} \) is recorded.
By repeating the third stage calculations for different number of load cycles, the dependence of \( \Delta R_k \) on the number of load cycles is obtained. Furthermore, if \( R_{k,N} \) becomes equal to \( F_{\text{max}} \), the considered number of load cycles is identical to the number of load cycles leading to failure \( N_f \). Thus, by applying series of numerical modeling \( N_f \) can be identified. Therefore, by variation of \( F_{\text{min}} \) and \( F_{\text{max}} \), an interaction diagram for a certain pile system can be derived.

5 Numerical Results

In the following, the first results obtained with the developed approach are presented. Only one pile-soil system is presented here (steel pile \( L=30 \text{ m} \), \( D=2 \text{ m} \), \( t=3 \text{ cm} \)). The static pull-out capacity calculated with the numerical model amounts to \( R_k=4.89 \text{ MN} \).

The following table (Table 3) shows all important data required for the calculation and also the obtained number of cycles leading to failure \( N_f \). The cyclic load amplitude starts at a \( X_{\text{cyc}} \) of 0.2. Underneath this value is the stable zone (refer to Fig. 1). In order to get a complete grid to construct the interaction diagram, both \( X_{\text{mean}} \) and \( X_{\text{cyc}} \) were varied as shown in the table. From these values the \( F_{\text{max}} \) & \( F_{\text{min}} \) can be derived.

Table 3: Abstract of all important data for building a interaction diagram

<table>
<thead>
<tr>
<th>Diameter [m]</th>
<th>Length [m]</th>
<th>( X_{\text{mean}} ) [-]</th>
<th>( X_{\text{cyc}} ) [-]</th>
<th>( R_k ) [MN]</th>
<th>( F_{\text{max}} ) [MN]</th>
<th>( F_{\text{min}} ) [MN]</th>
<th>( N_f ) [-]</th>
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<td>21</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>0.6</td>
<td>0.22</td>
<td>4.89</td>
<td>4.01</td>
<td>1.86</td>
<td>1.02E9</td>
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<tr>
<td>2</td>
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<td>0.6</td>
<td>0.34</td>
<td>4.89</td>
<td>4.59</td>
<td>1.27</td>
<td>22</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>0.65</td>
<td>0.25</td>
<td>4.89</td>
<td>4.40</td>
<td>1.96</td>
<td>6.00E3</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>0.85</td>
<td>0.1</td>
<td>4.89</td>
<td>4.64</td>
<td>3.67</td>
<td>1.00E14</td>
</tr>
</tbody>
</table>

For example, using the normalized loads \( X_{\text{mean}}=0.50 \) and \( X_{\text{cyc}}=0.34 \), a maximum load of \( F_{\text{max}}=0.84\cdot R_k=4.11 \text{ MN} \) and subsequently unloading to \( F_{\text{min}}=0.16\cdot R_k=0.78 \text{ MN} \) was applied. Figure 3 shows the reduction of pile capacity with the load cycle number. An almost linear decrease of \( R_k \) in the logarithmic scale, i.e. with \( \log N \), is obtained. At a load cycle number of about 7,800 the decreased capacity is identical to the maximum load \( F_{\text{max}} \). This means that for the considered load combination the number of load cycles \( N_f \) leading to failure is \( N_f \approx 7,800 \). One point in the interaction diagram of the considered pile system is thus found. This
procedure was repeated for other load combinations listed in the previous table to derive the interaction diagram for this soil-pile-system.

Figure 3: Pile capacity degradation due to cyclic loading (D=2 m, L=30 m) for $X_{\text{mean}}=0.50$ and $X_{\text{cyc}}=0.34$

Figure 4 shows the preliminary results as an interaction diagram obtained using this approach. For each pair of $X_{\text{mean}}$ & $X_{\text{cyc}}$ the corresponding number of cycles ($N_f$) at failure are calculated as shown in Fig. 3. The number of cycles ($N_f$) at failure is decreasing significantly with an increasing cyclic load ratio ($X_{\text{cyc}}$). Simultaneously the number of cycles ($N_f$) decreases moderately with increasing mean load ratio ($X_{\text{mean}}$). The stable zone can be identified and is marked with a dotted line. The results can be considered as preliminary and finer grid points should be analyzed in order to obtain well-shaped interaction diagrams showing clearly the required number of cycles up to failure.

Figure 4: Interaction Diagram for steel tube Pile (L=30.0m D=2m, t=3.0cm)
6 CONCLUSIONS

The presented results show that the proposed method is a promising tool which is capable to derive system-dependent interaction diagrams as well as the decrease of pile capacity for a certain cyclic load configuration. With this method it is possible to study the influence of system parameters on the pile behavior under cyclic axial loading. In its present form the numerical model is only a basic model. In subsequent investigations, further parametric studies will be carried out for different pile dimensions, different relative densities of the sand soil, layered soils and different cyclic load magnitudes. A further refinement of the method is also highly desirable. In particular, consideration of small strain stiffness effects and distinction between first time loading and cyclic un- and reloading is foreseen. Subsequently, a validation of the method by systematic comparison with experimental test results has to be carried out.

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7 REFERENCES

BASICS OF A DESIGN CONCEPT FOR BUCKET FOUNDATIONS OF OFFSHORE WIND TURBINES

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Keywords: Foundation, Bucket, Suction, Monopod, Multipod

1 INTRODUCTION
Within the next decades an enormous number of Offshore Wind Turbines (OWT) with capacities larger than 5 MW shall be erected in wind farms in the North- and Baltic Sea. Currently preferred foundation concepts use driven piles as foundation elements, either one single large diameter pile (monopile) or three or four piles as supports of jacket or tripod structures. However, the installation of piles by driving causes significant noise emission, which harms the environment and in particular mammals living in the Sea. It is thus foreseeable that the allowance for driving at least in the German sea regions will in future be limited to certain time slots. This would of course severely hinder the process and increase the costs of the installation. Due to that, alternative foundation solutions which do not necessitate driving are sought.

In that regard, buckets are a promising foundation solution. Here the foundation element, consisting of a steel cylinder with a steel lid on top, is lowered to the ground by pumping water out of the cavity between sea bottom and bucket lid and thus inducing an underpressure (suction) inside. Such buckets could either be used as a single foundation element (monopod) or as substitutes for the piles of tripods or jackets (multipod, Fig. 1).

Buckets or “suction cans” have already been used in the offshore oil and gas technique, but not for the foundation of wind energy converters with its very special loading conditions. Design recommendations or guidelines for these structures do not yet exist. Therefore, a general design concept describing the design steps and requirements is urgently needed in order to make buckets a feasible alternative to driven piles.

In the scope of the BMU\textsuperscript{1}-funded research project “WindBucket”, the installation process as well as the behavior under operation of buckets were investigated. The results of these investigations are described in the paper in hand. First, methods for the analysis of the installation process are assessed, and recommendations on the analysis procedure are given. Second, the behavior of buckets under combined horizontal and moment loading, as it occurs

\textsuperscript{1} BMU: German Federal Ministry for the Environment, Nature Conservation, Building and Nuclear Safety
for monopods, is analyzed by means of numerical simulations. Third, the behavior of buckets under rapid tensile loading, as it occurs for multipods, is considered. Finally, based on the findings a general design concept for bucket foundations of offshore wind energy converters is outlined. Open questions and from that research needs are identified.

Figure 1: Monopod (left) and multipod (right) foundations with buckets

2 INSTALLATION PROCESS

The forces acting on a bucket during its installation are shown in Fig. 2. The bucket is first lowered to the seabed and sinks a certain depth into the ground due to its own weight. Then a valve in the bucket lid is connected with a pump which pumps water out of the cavity between bucket lid and seabed, hereby inducing an underpressure \( \Delta u \). If the resulting underpressure force \( F_{\text{und}} \) together with the own weight of the bucket \( G \) is greater than the resultant resistance force \( R_{\text{in}} \), the bucket penetrates into the ground:

\[
F_{\text{und}} + G = \Delta u \frac{\pi D_i^2}{4} + G \geq R_{\text{in}}
\]  

(1)

The resistance force results from inner and outer skin friction and tip resistance acting on the currently embedded part of the bucket skirt:

\[
R_{\text{in}} = \int_{z=0}^{H} \tau_{0} \pi D_o \, dz + \int_{z=0}^{H} \tau_{i} \pi D_i \, dz + \pi D_m \tau_{t}
\]  

(2)

In the analysis of the installation process, the development of the resistance force with depth has to be predicted and the – also depth-dependent – maximum allowable underpressure has to be determined in order to prove that the desired penetration depth can be realized.

2.1 Calculation of the resistance force

In principle, two approaches exist to calculate skin friction and tip resistance. CPT-based methods base on the cone resistance \( q_c \) measured in a cone penetration test (CPT). Factors \( k_f \) and \( k_p \) are introduced to calculate skin friction and tip resistance from \( q_c \):

\[
\tau(z) = k_f q_c(z)
\]

\[
\sigma_t = k_p q_c(z = H)
\]  

(3)
Figure 2: Effects and resistances occurring during bucket installation

In a DNV guideline [1], value ranges of $0.001 \leq k_f \leq 0.003$ and $0.03 \leq k_p \leq 0.06$ are recommended for calculation of the penetration resistance of steel skirts in sand.

In contrast, effective stress approaches use the effective vertical stress $\gamma' z$ in a considered depth to calculate skin friction and tip resistance. The factors used are $K \tan \delta$ (earth pressure coefficient times coefficient of friction) for the skin friction and a bearing factor $N_q$ for the tip resistance:

$$
\tau(z) = K \tan \delta \gamma' z
$$

$$
\sigma_i = N_q \gamma' H
$$

Recommended values for $N_q$ and $K \tan \delta$ and limit values for skin friction and tip resistance in sand soil are given in the API regulations [2].

The Equations (3) and (4) do not take the effect of seepage forces on the resistances into account. In a permeable soil like sand, the underpressure inside the bucket induces water seepage towards the bucket’s inside. The inner skin friction is decreased due to the upwards direct hydraulic gradient inside the bucket. Also the tip resistance will decrease, whereas the outer skin friction will in general be increased.

Usually, the ratio of the current underpressure to the critical underpressure causing hydraulic failure ($\Delta u_{hyd}$) inside the bucket is used to quantify the effect of seepage. If a linear dependence on this ratio is presumed, the following equations apply for the CPT-based approach:

$$
\tau_i(z) = k_f q_c(z) \left(1 - r_i \frac{\Delta u}{\Delta u_{hyd}}\right)
$$

$$
\tau_o(z) = k_f q_c(z) \left(1 + r_o \frac{\Delta u}{\Delta u_{hyd}}\right)
$$

$$
\sigma_i = k_p q_c(z = H) \left(1 - r_i \frac{\Delta u}{\Delta u_{hyd}}\right)
$$

(5)
Senders & Randolph [3] proposed to apply $r_i = r_t = 1.0$ and $r_o = 0$. Feld [4] applied a similar approach to account for seepage to an effective stress approach. Houlsby & Byrne [5] developed a method which also bases on the effective stress approach, taking the effect of hydraulic gradients on the effective stresses into account.

It should be mentioned that the installation of a bucket in layered soils consisting of non-cohesive and cohesive soils is even more difficult to analyze. A clay layer embedded in sandy soil interrupts the water seepage and can thus cause a strong increase of the penetration resistance in the underlying sand. If a sand layer is present in a cohesive soil, a plug heave of the overlying clay layer can occur when the skirts reach the sand layer. In such cases, special investigations and considerations are necessary.

Although a number of research works have been carried out, the prediction of resistance forces remains rather uncertain. Uncertainties concern both the $k_F$- and $k_P$-values and the correction for seepage effects, in particular the $r$-factors and the assumption of linear change with underpressure ratio. Therefore, even for penetration in homogeneous soil urgent need for further research must be stated.

### 2.2 Maximum underpressure

The realized underpressure inside a bucket is practically limited by the following limit conditions:

- If the underpressure and with that the upwards directed hydraulic gradients inside the bucket in sandy soil become too great, erosion channels and finally hydraulic failure occurs. This would result in uncontrolled penetration, tilting and soil loosening.
- At a certain underpressure, buckling of the steel cylinder occurs. Therefore, buckling loads have to be analyzed thoroughly. It should be noted that measures to increase the buckling load (increased steel thickness, installation of stiffeners) also affect the penetration resistance.
- The minimum absolute pressure and thus the maximum underpressure is determined by the cavitation limit. Practically, the maximum underpressure is almost equal to the sum of atmospheric pressure ($\approx 100$ kPa) and the water pressure $\gamma_w h_w$ in the current water depth of the lid.

The critical underpressure with respect to hydraulic failure in sandy soils can be determined by means of flow net calculations. For simplification, a stationary seepage state is usually assumed. For each passed depth, the underpressure leading to a critical hydraulic gradient along the inner soil surface of $i_{crit} = \gamma'/\gamma_w$ and thus leading to hydraulic failure can be determined.

Several authors developed analytical equations for simple determination of the critical underpressure in homogeneous soil (e.g. [3], [4], [6]). The ‘suction number’ $S_N$ is used herein as a dimensionless measure for the critical suction. As an example, the approach of Ibsen & Thilsted [6] is given:

$$S_N = \frac{\Delta u_{hyd}}{\gamma H} = 2.86 - 1.20 \arctan \left( 4.1 \left( \frac{H}{D} \right)^{0.8} \right)$$

Houlsby & Byrne [5] also considered an increased permeability inside the bucket due to the soil loosening induced by the seepage forces in their approach. However, also this approach is
only applicable to (initially) homogeneous soil. In general cases with different soil layers, site-specific flow net calculations are necessary to determine critical underpressures.

3 BUCKETS UNDER COMBINED HORIZONTAL AND MOMENT LOADING

Monopod buckets are subject to combined horizontal and moment loading with varying eccentricities of the horizontal force \( h = M/H \) with respect to the bucket lid level and almost constant vertical force. Fig. 3 shows measured load-rotation curves from field tests with relatively small buckets. The first test with a bucket with diameter of 2m and \( L/D = 1 \) in Frederikshavn was reported in [6] and [7], the second test with a bucket \( D = 4m \) and \( L/D = 0.625 \) in Sandy Haven in [5] and [9].

Achmus et al. [10] used these field test results for the validation of a numerical simulation model. An elastoplastic material law for sand soil with a stress-dependent stiffness approach was applied herein. The numerically determined load-rotation curves are also shown in Fig. 3. Evidently, the load-bearing behavior of a monopod bucket can be well reproduced by such a numerical model. In [10] analytical equations are proposed for the estimation of ultimate capacity and initial stiffness of buckets in homogeneous sand. However, for general conditions numerical simulations are required to predict capacity and stiffness of monopod buckets. It should also be noted that it is yet unclear, whether the load-bearing behavior is significantly affected by structural changes of the soil during installation of the bucket.

\[
\Delta \theta_N = \frac{\Delta \theta_N}{\theta_s} = T_c(\zeta_c) T_b(\zeta_b) N^\alpha
\]

Here \( \Delta \theta_N \) is the increase of rotation after \( N \) cycles and \( \theta_s \) is the static rotation under the maximum moment \( M_{\text{max}} \). The parameters \( \zeta_c = M_{\text{min}}/M_{\text{max}} \) and \( \zeta_b = M_{\text{max}}/M_{\text{ult}} \) describe the loading type \( (\zeta_c) \) and the relative load level \( (\zeta_b) \) with respect to the ultimate static moment.
capacity $M_{ult}$. The functions $T_b$ and $T_c$ take the effect of the cyclic load characteristic into account, and $\alpha$ is a cyclic accumulation parameter.

Zhu et al. investigated a model bucket with $L/D=0.5$ in loose dry sand and found an accumulation parameter $\alpha \approx 0.39$, whereas Foglia et al. investigated a bucket with $L/D=1$ in dense saturated sand and found $\alpha \approx 0.184$. The results for the functions $T_b$ and $T_c$ are shown in Fig. 4. Different functions $T_b$ were obtained, which means that this function and also the accumulation parameter are dependent on the system (bucket geometry, soil conditions). In contrast, almost identical functions $T_c$ were found. Therefore, this function seems to be independent of the system features. The same conclusion was drawn in [13] for monopiles. This means that the same methodology as proposed for monopiles in [14] can be used for buckets, presumed that the function $T_b$ and the parameter $\alpha$ for a certain bucket system can be derived. Regarding that, research is necessary. However, an investigation presented in [15] showed that the cyclic behavior of bucket systems seems to be similar to that of monopile systems. Thus, as a first approximation, experiences from monopile might be transferred to bucket foundations.

The tests in [11] showed that the un- and reloading stiffness of a bucket remains almost constant or changes only slightly with the number of load cycles applied.

An open question is how much storm loads affect the capacity of a bucket foundation and how this could be considered in the design. The German BSH standard [16] requires the consideration of a preceding storm in the ULS proof of a OWT foundation. The only way to deal with that problem is to carry out cyclic soil tests in the laboratory and to estimate strength degradation (due to build-up of excess pore pressures) based on the test results. A respective method for gravity foundations is outlined in [17]. However, the method is quite complex. It is highly desirable to develop simpler engineering models to deal with the question of possible strength degradation. Therefore, research regarding that is urgently needed.
4 TENSILE RESISTANCE OF BUCKETS

Multipod buckets of offshore wind energy converters are mainly subject to compressive and tensile loads. Due to the relatively small own weight of an OWT structure, significant tensile loads occur under extreme loads, and these loads are usually design-driving regarding the bucket geometry.

The calculation of the drained tensile capacity of a bucket, which consists of outer and inner skin friction forces, is rather simple. However, in reality the extreme loads from waves are transient loads which build-up fast and last only a few seconds. Therefore, the soil behaves partially drained or even undrained, i.e. excess pore pressures generate. In the case of tensile loading, a suction is induced below the bucket lid, which can – dependent on the loading rate – significantly increase the pull-out capacity.

Thieken et al. [18] established a hydraulic-mechanically coupled numerical model which enables a quantification of the favorable effect of a high pull-out rate on the tensile capacity. Fig. 5 shows the results for an exemplary bucket geometry in dense sand. It can be seen that for large pull-out rates and thus undrained behavior the pull-out capacity is increased by a factor of 5 and more compared to the drained case. However, large heave values are necessary to mobilize the large capacity.

![Figure 5 Interaction diagram for the tensile resistance of a bucket (D=10m, L/D=1, dense sand) [18]](image)

Under a constant load greater than the drained capacity, a constant heave rate of the bucket occurs, with the actual rate dependent on the magnitude of the tensile load (see also [18]). This indicates that under cyclic loads significant heave accumulation can occur, if the drained capacity is exceeded too much or too often. However, it is an open question how such heave accumulation under cyclic tensile loads can be accurately or even approximately predicted.

5 BASIC DESIGN CONCEPT

The main geotechnical design proofs which have to be fulfilled for foundations of wind energy turbines are:

- **Bearing capacity**: Sufficient safety against bearing failure under extreme loads – usually extreme wave loads are decisive herein – has to be secured.
Serviceability: Settlements and in particular tilting of the tower has to be limited. An important requirement is that a certain permanent tilt angle (in most projects 0.5° including installation tolerances) must not be exceeded.

Stiffness during operation: It must be proved that the natural frequency of the OWT system does not coincide with one of the main excitation frequencies of the structure. This is particularly important for monopod foundations, since here the natural frequency of the system is highly affected by the stiffness of the monopod-soil system.

In all these design proofs a possible effect of cyclic loading on the soil or foundation behavior must be taken into account. Regarding the bearing capacity proof, the BSH guidelines [16] require consideration of a certain design storm in the determination of effective soil strength. Fig. 6 summarizes recommendations regarding the necessary design steps for monopod and multipod buckets, based on the state of knowledge described above.

Generally, 3D finite element simulations with appropriate soil material laws are necessary to determine the bearing behavior of buckets under horizontal and combined loading. Calculations with lower bound and upper bound soil parameters should be carried out in order to cover uncertainties. Possible changes of soil parameters due to the foregoing installation process should also be taken into account.

The behavior of monopod buckets under cyclic loads can be assessed either based on existing model test results or based on numerical simulations (e.g. with the stiffness degradation method, see [15]). A considerable change of operation stiffnesses due to cyclic loads must not be expected according to the existing test results. However, regarding strength degradation for the bearing capacity proof, an assessment based on cyclic laboratory test results must be done. There is yet no ready-to-use design method to do this, but at least with conservative assumptions an acceptable proof should be possible.

The calculation of static resistances of a monopod bucket under axial load can be done analytically. However, the calculation of the behavior under transient short-term loads and under cyclic axial loads above the drained capacity is not yet state-of-the-art. Based on present knowledge, only a prediction of the possible bandwidth of the foundation behavior is possible by parametric studies. Therefore, the observation method has to be applied for multipod buckets. This means that a comprehensive monitoring program has to be realized to control the structure’s behavior. For the case that the range of predicted deformations or stresses is exceeded, countermeasures must be foreseen.

6 Conclusions
The design of bucket foundations for OWTs requires at the time being elaborate finite element simulations. Based on such simulations, a safe design of monopod buckets is possible, if parameter bandwidths are considered and cyclic laboratory tests are taken into account. However, for multipod buckets which are subject to large cyclic tensile loads, the uncertainties even in a design with numerical simulations are still too great, which means that the observation method has currently to be used in the design. Basic research as well as demonstration projects are urgently needed to improve the understanding of the complex behavior of bucket foundations and also to develop simpler calculation methods for practical design purposes.
Figure 6 Design steps for monopod and multipod buckets

7 REFERENCES


THE GICON®-TLP FOR WIND TURBINES – WIND AND WAVE TANK TEST RESULTS

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Keywords: GICON®-TLP, Floating Foundation for Wind turbines, Tank tests at MARIN

Summary: This document summarized the preliminary status of the research project regarding the so call GICON®-TLP. It was not possible for to submit a full paper for the IWEC2014 in Hannover. The reason for was a lot of work deduced from the pilot plant design and certification of the GICON®-TLP.

1 ABSTRACT

A great potential for offshore wind farms off the coast of the European Union is in the range of 30 m up to 200 m water depths. These offshore wind resources are driving the solution requirements for new offshore wind power platforms, due to the fact, that fixed foundations are not any more economical in this water depth.

The paper will describe the preliminary design of the so called GICON®-Tension Leg Platform (TLP) as an innovative foundation concept for offshore wind turbines. The TLP-structure is characterized by an open relief design with a specific configuration of the mooring line system. The GICON®-TLP is meant to provide a solution for harnessing the power of offshore wind at depths of water between 20 and 350 m. Also a design for 700 m is still in the developing phase.

Furthermore preliminary results from model basin tests will be described. Therefore the currently ongoing research, by comparison of calculated and experimental data obtained by extensive wave tank experiments with a scale model of an offshore wind turbine at the Maritime Research Institute Netherlands (MARIN) in summer 2013, will be presented. These tests have provided insights regarding the dynamic characteristics of the GICON®-TLP by analyzing the systems response to different load cases.

The experiments included wind and wave loads, which represent three different sea states, each with three different directions of inflow. The chosen load cases represent the proposed location in the German Baltic Sea where the full scale prototype will be erected.

Furthermore the results of the scaled test at MARIN have confirmed that a superposition of the internal forces of the moored structure in operation for wind and wave loads can be
assumed for the structural design. This deduced from the stiffness of the mooring line system and the innovative mooring line configuration. Data and video from the tank tests will be shared as part of the presentation. Based on these tank test results the basic and detail design for the prototype will be finished. Finally, the erection of a full scale prototype in the Baltic Sea is planned for 2014/15.
TESTING OF THE SLIP JOINT CONNECTION IN A SCALED MODEL

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Keywords: Support Structures, Slip joint, Testing, Extreme loads, Dynamic behaviour

Summary: The slip joint connection is an alternative connection between the monopile and transition piece or tower, which circumvents the drawbacks associated with grouted and flanged connections. Within the FLOW research programme, a numerical and experimental study was done on the behaviour of the slip joint. The tests concern mounting, extreme loads and dynamic behaviour. The joint exhibits benign behaviour. Under dynamic loading, the behaviour improves over time, as the contact improves due to settling.

1 INTRODUCTION

Offshore wind turbines have a transition piece which facilitates the installation of the turbine tower on top of a support structure, often a monopile. Difficulties have risen in grouted connections [1], so the industry is interested in alternative connection methods. One of these is the slip joint connection. This connection consists of two conical tubes that are mounted on top of each other (see Figure 1), without further welding or grouting. The load is transferred by friction and normal forces in the mating surfaces.

The study of this type of connection is one of the projects within the FLOW research programme [2]. The slip joint is already being applied in an onshore wind turbine [3]. The offshore wind turbine application involves larger bending moments. To test the mechanical behaviour under these higher loads, scaled models have been modeled and tested at Delft University of Technology [4] and at WMC [5]. This paper presents results of modeling and testing of the behavior of this structure under extreme and dynamic loads at WMC.

2 TEST DESCRIPTION

The modeling and testing is based on the OC3 [6] monopile design with the 5 MW NREL reference turbine [7]. The test specimens are scaled 1:10.

Two specimens are tested. One where the top and bottom cone angles are 1°, and one for 2° cone angles.

The test setup is shown in Figure 1, and features two cylinders for actuation. Depending on the test, the cylinder control is based either on force or on displacement. The cylinders are equipped with sensors to measure forces (F1) and displacements (S). Strain gauges (G) are distributed over the specimen, and laser transducers (D) are mounted to measure the displacement (‘sliding’) of the top cone bottom rim over the bottom cone.

---

1 This letter is used as a reference for the sensors, and is used in the graphs. E.g. F1 + F2 denotes the sum of the forces in cylinder 1 and 2; S_avg denotes the average cylinder displacement.
Using the integrated design software FOCUS6 [8], the extreme loads and the fatigue loads are determined by simulation. These loads are scaled to match the specimen size and test duration.

Finite element calculations using MSC Marc [9] have given initial insights in the slip joint behavior and provided points of attention for the test.

The following tests are performed on the specimens:

- Mounting and unmounting
- 100% of the extreme load
- 200% of the extreme load
- A dynamic test

a. Mounting and unmounting test

During mounting, the top cone is pressed onto the bottom cone. The required force is necessary to overcome the elastic deformation and friction. The following formula describing the force displacement relation is derived as:

\[ F = 2\pi E \frac{t h}{D} (\tan^2 \alpha + \mu \tan \alpha) d \]  

(1)

Here, \( F \) is the applied axial force, \( E \) is Young’s modulus, \( t \) is the cone wall thickness, \( h \) is the height of the overlap between the top and bottom cone, \( D \) is the average diameter of the overlapping surface, \( \alpha \) is the cone angle, \( \mu \) is the friction coefficient, and \( d \) is the displacement, measured from the point of first contact. The first term between parentheses, \( \tan^2 \alpha \), is related to force balancing the elastic deformation, and the second term, \( \mu \tan \alpha \), is related to the friction force.

A force displacement graph obtained by finite element calculations is depicted in Figure 2. Starting from X=0 mm, going down, the top cone meets the bottom cone at -2.3 mm. Going further down, the required force pushing down increases. The rate of increase matches Equation (1). Pulling the cone up, the force first reverses sign. The displacement is due to the
specimen going from a compressed state to a tensioned state; the cones do not slide, until the pulling force overcomes the friction. Then the pulling force decreases, as the cone rises, relaxes, and the friction force decreases accordingly. The rate of decrease also matches Equation (1), taking the change of direction in account by changing the sign of $\mu$.

Figure 2: A typical force displacement graph. Node 244 is located on the top of the model. X is in vertical direction, pointing upward.

The numerical results match the equation, and suggest that the force increase and decrease rates can be used to determine the friction coefficient. This motivates the mounting and unmounting test.

b. Extreme load tests

The extreme loads consist of an axial load and a bending moment found by simulation. These are applied simultaneously to the test specimen. The extreme axial load is composed of two parts: first, the reference load due to the weight, and second, the load increment due to the extreme load events. The moment due to the weight is small compared to the moment increment due to the extreme loads and is ignored. The loads for the 100% extreme load case and for the 200% load case are summarized in Table 1.

Table 1: Extreme loads.

<table>
<thead>
<tr>
<th></th>
<th>Axial load [kN]</th>
<th>Bending moment [kNm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference load case</td>
<td>57.6</td>
<td>0.0</td>
</tr>
<tr>
<td>Extreme load increment</td>
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<tr>
<td>Extreme load 100%</td>
<td>60.6</td>
<td>107.6</td>
</tr>
<tr>
<td>Extreme load 200%</td>
<td>63.6</td>
<td>215.2</td>
</tr>
</tbody>
</table>

c. Dynamic tests

To obtain a load sequence for the dynamic test, the following procedure is followed:

1. Simulation of the wind turbine yields a time sequence of loads that is representative for real life loads. This involves standard IEC 61400-3 class I-B wind conditions [10], and wave conditions based on measurements as reported in the Upwind project [11].
2. The fatigue characteristics, expressed by a Markov matrix, are determined using rain flow counting.
3. Create a variable amplitude time series corresponding to this Markov matrix, reducing the number of cycles such that it fits in 2% of the available testing time. Increase the loads so that 50 repetitions of these time series match the fatigue damage determined using the Markov matrix obtained in step 2.
4. Scale the loads to match the scaling of the dimensions.
The dynamic test is performed by applying the time series 50 times. The repetitive structure allows to compare the results over the course of the test, while the random variable amplitude within each time series preserves the characteristics of the real life simulation results.

3 RESULTS
The tests are performed for the two specimens. The nature of the results is discussed.

Table 2: Tests performed on the specimens

<table>
<thead>
<tr>
<th>Specimen</th>
<th>4710</th>
<th>4700</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cone angle</td>
<td>2°</td>
<td>1°</td>
</tr>
<tr>
<td>Mount &amp; unmount (friction)</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>Extreme 100%</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>Extreme 200%</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>Settling / Dynamic test</td>
<td>Settling</td>
<td>Dynamic</td>
</tr>
<tr>
<td>Additional test</td>
<td>Extreme 200%</td>
<td></td>
</tr>
</tbody>
</table>

a. Mounting and unmounting test
The mounting and unmounting test is performed for both specimens. Graphs of the imposed displacements and the resulting force-displacement are shown in Figure 3. Multiple mounting and unmounting cycles are performed.

Figure 3: Repeated mounting and unmounting by prescribed displacements (left). The corresponding displacement - force graph for the 2° test specimen (right). F is positive for pulling down.

The shape of the displacement – force graph does not match the graph shown in Figure 2. It is assumed that this is due to the imperfect shape of the specimens [12]. This leads to partial contact instead of the perfect contact assumed in the derivation of the formula and the
numerical simulation. In the course of mounting, the conical shapes bend elastically to get matching surfaces. The stiffness in axial direction is lower than predicted, and cannot be used to infer the friction constant. Notice however, that the stiffness increases as the top cone slides further onto the bottom cone.

b. Extreme load test

The 100% load test and 200% load test show that the slip joint connection can transfer the extreme loads. Repeated application of the 200% load shows a decreasing amount of settlement with each application. The results of three 200% load tests are shown in Figure 4. Two tests are performed before the dynamic load test discussed in the next section, and the third is performed after.

![Figure 4: Three executions of the 200% Extreme load test. Between the second and the third test, the dynamic test is performed (omitted in the graphs). The graphs depict the applied loads (top left), the top cone rim displacements (top right), strains along the bottom rim of the top cone (bottom left), with strain gauges (G) and laser transducer placements in a cross section view (right).](image)

The results show that the top cone first settles under the reference load. Then the extreme load increment is applied twice. The effects of the moments show in the divergence of the displacement of D3 to those of D1 and D2. The divergences are similar in the first two tests; in the third test, after the settling caused by the dynamic test, the divergence is less.

A similar effect shows in the strain gauges on the bottom rim, where the range of the cycle reduces after the dynamic test.
c. Dynamic load test

The results of the dynamic load test are shown in Figure 5. After the application of the reference load, the bending moment series are applied 50 times. The average of the three laser transducers is shown. With each series application, the displacement settles in a decreasing rate. By averaging the displacement over each series, this settlement is shown more clear. The variance in the signal is also evaluated, and shown to decrease.

![Figure 5: The dynamic test. The variable amplitude time series (top left) is applied 50 times (top right). The average displacement per series shows slow settling (bottom left), and the decrease in variance shows stiffening (bottom right).](image)

The strain gauges at the bottom rim of the top cone show the same stiffening behavior, as can be observed in Figure 6.

![Figure 6: Strain gauges 28 to 30, at series 0 (left), 9 (center) and 49 (right). Notice the decrease in the range of the strains.](image)
M. J. de Ruiter and T. Westphal

4 CONCLUSIONS

The mounting and unmounting tests show that the axial load is supported before reaching the full contact state. The partial contact state at the start of the test is due to shape inaccuracies of the specimens. The partial contact state results in axial stiffnesses that are not as high as predicted.

The specimens performed well under extreme loads.

Under dynamic loading, the slip joint settles, effectively stabilizing the system over time. With increasing settlements, the strain and displacement ranges in the joint reduce. The results of an additional extreme load test at the end of the dynamic test demonstrates that the slip joint can withstand extreme loads also at the end of its lifetime.

5 REFERENCES


Improved Tank Test Procedures for Scaled Floating Offshore Wind Turbines

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Keywords: Floating Offshore Wind turbines, tank testing, scaled models, rotor model, representative aerodynamic forces, Software-in-the-Loop

Summary: This study collects issues from previous tank test campaigns of scaled Floating Offshore Wind Turbines (FOWT), compares the different scaling methodologies, points out critical aspects and shows possible alternatives and recommendations for future tests depending on the specific objective. Furthermore, it gives practical recommendations for the modeling and construction of scaled rotors. The presented scaling procedure will be applied in tank tests within the EU Seventh Framework Program InnWind (ENERGY.2012.2.3.1 “Innovative wind conversion systems (10-20MW) for offshore applications”).

1 Introduction

The numerical simulation tools for calculating motion and dynamics of FOWT are still under development. Many existing codes of the offshore and oil & gas industry provide approved and validated routines describing the dynamics and loading from waves and currents to floating structures. But they lack a sufficient consideration of the aerodynamic loads on the rotor caused by turbulent wind, complex rotor aerodynamics and an own control system on top of the structure. This introduces a fully non-linear loading source to the entire system. Additionally, the influence of second order hydrodynamics and mooring line behavior is not always modeled in sufficient detail.
Currently many research programs like OC3/OC4 [1] compare the results of different simulation codes together with measurement data from model tests. Today only a few full scale prototypes are in operation and measurements of existing floating concepts are rare. Therefore experiments with scaled floating models in wave tanks in combination with wind modelling are a cost effective approach to gather data for referencing and validating the dynamics and loads of FOWT. This might also be a necessary step for the certification of a prototype.

2 LITERATURE REVIEW
For more than twenty years the idea of installing wind power generators offshore on floating foundations is pursued in research and industry. Within many of the past and present projects the challenge of experimental tests with combined wind and wave loading has been addressed. Overviews of these projects can be found in [2], [3], [4] and [5]. Useful practical information on scale model building can be found in the theses of the University of Worcester, see [6] and [7] and [8].

Wave tank tests of FOWT can be conducted with two major purposes: Firstly for validation of numerical models and secondly in order to obtain the full system dynamics in all occurring load situations for full-scale prototypes, which are going to be built. There have been various commercial projects for the implementation of floating wind energy systems. The projects Hywind [9], [10] and WindFloat [11], [12], [13] and [14] were among the first to gain commercial experience in the years 2009 and 2011, respectively. Gicon [15], [16] and GustoMSC with their “Tri-Floater” [17], [18], [19], [20] followed in the years after with multiple test campaigns. In Japan the Fukushima project [21] with various phases includes model and prototype tests for a conventional spar concept, a “hybrid spar” and a semi-submersible. The wave tank tests for the spar concept are documented in [22]. Innovative large-scale offshore floating structures are investigated by Kyushu University, [23]. Ishihara has developed an innovative multi-turbine foundation and presented model tests in [24] and [25]. The “Japan Marine United Advance Spar” was scaled for wave tank tests, [4]. The large-scale prototype projects in Europe, of which tank tests are known, are HiPRWind with a semi-submersible [26], BlueH with a two-bladed rotor and semi-submersible foundation [27] and the Norwegian Sway, which is a tension-leg spar [28]. A comparable concept is the Nautica wind power layout, for which some information on testing can be found online in [29]. The multi-turbine concept WindSea offers some data on the wave tank tests online, see [30] and [31]. Tests were also conducted for the multi-purpose foundation Poseidon, see [32]. This concept was tested in various wave tanks with and without wind and also on site with different prototype scales. The Pelastar TLP by Glosten Associates [33] was tested but very little information is published. The Vertiwind concept by Technip with a vertical-axis wind turbine was tested in a scale of 1:2, see [34]. Recently, the Nautilus semi-submersible concept, which has also been tested in the Cork wave tank in Ireland, was published in [35]. Iberdrola is developing a TLP concept and has presented the wave tank test layout in [36] with a publication on the most recent test campaign, see [37]. The University of Maine is involved in a project building the first scaled FOWT prototype in the US called VolturnUS, see [38]. All of the described model tests have used a scale between 1:130 and 1:2.
Various publicly funded research projects have included FOWT wave tank tests. As part of the Marinet project [39], state of the art scaling routines were revised and the dynamic-elastic scaling proposed as adequate for scaled FOWT-tests. The DeepWind project aimed at an integrated research on vertical-axis floating wind turbine (VFOWT) systems including tests [40]. A numerical model development of VFOWT together with a wave tank validation is given by [41].

Within the Winflo project [42], scaled models were developed and tested in order to understand the dynamic behavior of the concept design and to validate the floater and mooring design. The models included an adapted rotor design. The DeepCwind [43] project to this point delivers the most throughout insight available on the capabilities of numerical and experimental investigation of the three major FOWT systems: tension leg platform (TLP), semi-submersible and spar buoy. Major parts of the research were the assessment of a proper scaling methodology [44] and its numerical verification [45], the realization of a model wind turbine for wave basin testing [46] and the development of thrust scaled blades [47]. Also in the focus were the analysis of the interaction of the mooring dynamics with the global response of the FOWT [48] and the calibration and validation of a full scale simulation model within the simulation software FAST [49]. In 2013 a new test campaign with the same semi-submersible platform has been performed under improved testing conditions in the MARIN facility in the Netherlands, see [50]. The results together with a comparison to the previous DeepCwind campaign and a model of a smaller scale (1:130) for a reliability assessment can be found in [51]. A code validation of the MLTSIM hydrodynamics code coupled with the FAST software with the same test data can be found in [52]. A discussion on the differences of the behavior of the three different platform concepts is available in [53].

A detailed test session description aiming at the realization of a full scale turbine has been given by Utsunomiya. His tests cover simplified scale model tests for extreme condition dynamic analysis, see [54], and simulation validation, see [55]. They reach towards scaled prototype tests in full-scale at-sea-environment [56] and [57]. The derivation of the numerical model is given in [58]. A semi-submersible with a single-point mooring has been tested by Osaka University see [59]. Shin et al. published test results to study the motions of the OC3-Hywind spar [60] and a new spar-type FOWT model in 2D and 3D wave tanks, see [61], [62]. Another investigation of a “stepped spar”-model in a curved wave tank has been presented by Sethuraman et al. with a focus on the validation of the software OrcaFlex and nonlinear mooring line behavior comparing the RAOs for pitch, surge and heave motion in regular and irregular waves, see [63]. Philippe et al. [64] tested a model of the Dutch Trifloater in order to validate their numerical model. The influence of storm waves on a simplified TLP model was analyzed by Wehmeyer et al. [65] with varying tower stiffnesses, putting large effort in high wave modeling quality and showing that the inclusion of second order hydrodynamics is essential, especially regarding TLP-specific responses such as ringing and slack line events. Another validation of numerical results for a TLP has been done by Ren et al., [66] and Olinger et al. [67], who collected information on the dynamic behavior in pitch, surge and heave direction using simplified models of TLP and spar type FOWT. They found that surge motion of the platform dominates over other motions for TLP & Spar and that varying tether
pretension has little effect on RAO values. Within the AFOSP project a monolithic concrete spar design of a FOWT has been developed and wave basin tests presented in [68].

The impact of the rotor dynamics of the wind turbine rotor or the gyroscopic effect on the coupled system dynamics has been studied in [69]. In [70] an instability of floating wind turbines has been reported, yielding large amplitudes in yaw together with large amplitudes in pitch-motion coupled by the gyroscopic effect.

The presence of two different phases of the surrounding medium (air and water) poses a general problem to the geometrical scaling of the considered system. A description of the general loads acting on offshore structures within classical marine technology and the underlying physics can be found in [71] and [72]. For hydrodynamic similitude, Froude scaling needs to be applied, but inevitably leads to an error in the scaling of the Reynolds number for both hydro- and aerodynamics. While testing experience indicates that on the hydrodynamic side, the KC number is more significant [73], the aerodynamic Reynolds number is of high importance due to its influence on the flow condition around the airfoil. The geometric downsizing of the wind turbine effectively alters the aerodynamics of the scaled turbine, i.e. the Reynolds number and the associated forces on the turbine [74]. In order to keep a correct relationship of the forces, different solutions have been used in the past. Rotor models range from concentrated masses with added point forces [55], over drag disks with a rotating body [12] up to the redesign of the rotor blades in order to correctly scale the aerodynamics of the rotor, see [50], [75] and [76]. More optimization of this state-of-the-art scaled blade design through an alternative pitch angle distribution is possible [77]. In order to avoid the discussed scaling problem, a part of the environment can be imitated by real-time controlled actuators as has been done for hydrodynamics [78], [79] and [80] and for aerodynamics [81].

Building on the given experiences, two alternative procedures for rotor modeling are presented and evaluated below. The aims are to optimize the quality of experimental results when using an adjusted blade design and to reduce the costs of future experiments.

### 3 Aerodynamic Forces from Re-Designed Model Scale Rotor

The scaling of the aerodynamic forces is sketched in the following. Further the establishment of these loads through direct wind forcing is discussed and the requirement of a tailored model scale rotor design is detailed, following the work of [39], [44], [51], [77] and [76]. As the wind generation system may limit the range of wind speeds that can be generated, a free parameter is introduced, which modifies the scaled inflow speed at the expense of changing the tip speed ratio. This approach is denoted the Froude-\(\beta\) scaling, see [82]. The limitations of the redesigned rotor approach and the open questions subject to current research are outlined.

#### a. Froude scaling and the need for a redesigned rotor

As mentioned above, testing floating wind turbines in wave tanks leads to application of Froude scaling in order to fulfill the requirement of a true scaling of the wave force. This follows from the role of gravity as the governing restoring effect for the waves. Given a length scale ratio of

\[
\lambda = \frac{L_p}{L_m}
\]

(1)
the conservation of gravitational acceleration $g$ between prototype and model scale locks the time scale ratio to be

$$\frac{T_p}{T_m} = \lambda^2. \quad (2)$$

Here $(L, T)$ are representative length and time scales and subscripts $(p, m)$ denote prototype and model respectively. Due to the requirement of a conserved ratio between structural mass and water mass, mass $M$ scales according to

$$\frac{M_p}{M_m} = \rho_w \lambda^3, \quad (3)$$

where $\rho_w$ is the density of water. The above scaling of length, time and mass leads to a scaling of water particle velocities as

$$\frac{u_{wp}}{u_{wm}} = \frac{1}{\lambda^2}. \quad (4)$$

Hence, if the ratio of air and water velocities is to be preserved from prototype to model scale, a similar relation for the air-velocities $u_a$ follows

$$\frac{u_{ap}}{u_{am}} = \lambda^2. \quad (5)$$

Further, forces, $F$, scale as

$$\frac{F_p}{F_m} = \rho_w \lambda^3. \quad (6)$$

Rotor thrust is given as

$$F_T = 0.5 \rho_a C_T A u^2 A \quad (7)$$

where $C_T$ is the thrust coefficient, $A$ is the rotor area and $\rho_a$ is the air density. Given that the rotor area scales as $\lambda^2$ and that the air velocities scale by $\lambda^2$, the requirement of force to scale as

$$\frac{\rho_{wp}}{\rho_{wm}} \lambda^3 \quad (8)$$

yields the desired scaling of the thrust coefficient as

$$\frac{C_{TP}}{C_{TM}} = \frac{\rho_{wp}}{\rho_{wm}}. \quad (9)$$

Although this result of a conserved thrust coefficient may suggest that the prototype rotor design can be geometrically down-scaled to lab scale, the lowered lab-scale Reynolds number which results from the scaling of air-velocity leads to insufficient thrust levels. At the smaller Reynolds number, the lift-to-drag ratio is reduced and an increased chord-length is therefore needed for the model-scale blade to deliver the same thrust-coefficient as at full scale [39], [44], [51], [77] and [76]. For this reason a re-design of the blades at model scale is needed. In this context it is natural to base the design on low-Reynolds number air foils. The redesigned model scale rotor will thus have blades with increased chord-length relative to the prototype scale blades, see e.g. [77].

b. Adjustment of air-velocity scaling

The changed Reynolds number and blade geometry implies that the similarity of the aerodynamic flow between prototype and model scale is broken. Further, for the above scaling, a large scale ratio leads to very small air-velocities at lab scale – in some cases so
small that perturbations from heat convection and other motion in the lab can cause significant distortion. On the other hand, the wind generation system may also have limited capacity such that at larger scale ratios, certain wind speeds cannot be obtained. Note here that the wind generation is very different from the controlled conditions in a usual wind tunnel. The model scale rotor area may be 3m in diameter and the wind must be generated over water. For the above reasons, an adjustment of air-velocity scaling may be needed. This can be described through a free parameter \( \beta \), such that

\[
\frac{u_{ap}}{u_{am}} = \lambda \beta^\frac{1}{2}.
\]

For \( \beta = 1 \), Froude scaling is obtained, while \( \beta < 1 \) implies larger air velocities than for Froude scaling. With this modified air-scaling, the thrust coefficient must scale as

\[
\frac{C_{T_p}}{C_{T_m}} = \frac{\rho_{wp}}{\rho_{wm}} \beta^2.
\]

Further, the tip speed ratio (TSR) will not be conserved since

\[
\frac{TSR_p}{TSR_m} = \beta.
\]

Although conservation of TSR is generally preferred, the lack of aerodynamic similarity is likely to justify such a change. Note that with the factor \( \beta \), the Reynolds number scales as

\[
\frac{Re_p}{Re_m} = \frac{3}{\lambda \beta^\frac{3}{2}} \nu_m
\]

where \( \nu \) is the kinematic viscosity. The Froude-\( \beta \) scaling is detailed in [82].

c. Pros and cons of the scaled rotor approach

The approach of a re-designed rotor at model scale is able to deliver the correct thrust, see e.g. [51], [76]. Further, given that the mass distribution and rotational speed are scaled correctly, the correct gyroscopic forces and 1P, 3P forcing frequencies will be reproduced [39], [77]. If also the structural stiffness is scaled correctly, the structural frequencies and deflection to the loads will scale correctly [39], [77]. Further dimensionless numbers which will be conserved are the Keulegan-Carpenter number (KC) and the Lock number. This is conserved in the Froude-\( \beta \) scaling, both for the aerodynamic and hydrodynamic forces. Other dimensionless numbers are not conserved. These are the Reynolds number in the air and in the water which leads to changed hydrodynamic force coefficients and the need for a re-designed rotor; the Weber number which measures the balance of hydrodynamic surface tension to inertial loads (not expected to have importance, except at very small scales); the Strouhal number in water and air - for preserved Strouhal number and strict Froude scaling \( \beta = 1 \), though, the vortex shedding frequency will scale as \( \sqrt{\lambda} \) which is consistent with the time scaling; the Mach number in water and air (not considered to be important); and the tip speed ratio, which is only conserved for \( \beta = 1 \). Besides the dimensionless numbers, a couple of effects will also not scale automatically:

- the aerodynamic torque
- the aerodynamic power
- the generator torque and its contribution to roll-forcing
- the magnitude of the 3P forcing from the tower shadow
However, a careful rotor and nacelle design may help on this. An accurate reproduction of the above mentioned properties is part of current active research.

4 Aerodynamic Forces from Software-in-the-Loop Method

Another method to include a realistic force to represent the aerodynamic thrust in combined wind and wave scaled tests is based on the use of a ducted fan substituting the wind turbine scaled rotor. The fan thrust is controlled by the fan rotational speed set by the controller, which again depends on a computer real time simulation of the full scale rotor in the wind field. The real time simulation considers the platform motions measured in real time in the wave tank test. Therefore, the aerodynamic damping is modelled by the fan force. We refer to the described method as Software-in-the-Loop (SIL).

The layout of the SIL system is shown in Figure 1. The left side describes the simulation part of the system, which works in full scale, and the right side represents the wave tank scaled test. The different magnitudes that are interchanged between both blocks are transformed by the appropriated scaling laws based on the factor scale $\lambda$.

![Software-in-the-Loop Method Diagram](image)

The simulation tool provides the total aerodynamic force on the shaft $F_{\text{aero}}$ from integration of all the aerodynamic loading at the blade elements. This force in full scale is transformed to the model scale ($f_{\text{aero}}$) and the pulse width of the PWM (Pulse-Width Modulation) signal needed to produce the force in the ducted fan is provided by a calibration curve previously obtained. The control system regulates the fan speed that introduces the desired force at the model’s hub height. The waves produced by the wave maker are also acting over the platform and, together with the aerodynamic thrust, inducing motions. The acquisition system measures the positions and velocities for the 6 degrees of freedom of the platform at a certain sampling period. These measurements are sent to the simulation tool that is waiting for the data to advance one time step and calculate the new value of the aerodynamic thrust. For this
reason, the sampling period $\Delta t$ and the simulation time step $\Delta T$ have to be set accordingly (with a factor $\lambda^{0.5}$).

The methodology is explained in more detail in [83], where some experimental results obtained in a test campaign of a 6 MW semi-submersible platform at the ECN (École Centrale de Nantes) wave tank are also presented. These tests results using the SIL method are compared with computer simulations to show the performance and validate the method.

a. Pros and cons of the SIL approach

The SIL approach can provide a realistic aerodynamic thrust on the scaled model. As the computation of the force takes into consideration the motion of the platform, the effect of the aerodynamic damping is included. In addition, the control actions, the different types of wind (turbulent, constant, gusts) and the operating condition (idling, power production, etc.) are taken into account for the calculation of the thrust. Conditions where the waves and wind are misaligned are sometimes not easy to be reproduced in wave tanks with wind generation systems. With this method, they can be easily achieved by changing the fan orientation at the tower top. Furthermore, the SIL system allows performing test cases including wind at wave basins where the wind generation system is not available. In addition, the simplicity of the method makes it cost effective and flexible because the material is not specific for a certain wind turbine model and it could be used in different tests for different models.

Effects that are not scaled correctly with this procedure are:

- the aerodynamic torque
- the gyroscopic momentum.

Alternatively the use of a rotating scaled mass to represent the rotor inertia can be used to match the gyroscopic effects. Active research on the response of different fan units depending on the size of the wind turbine, scale factor, etc. is being conducted with the aim to explore the limits of the methodology.

5 CONCLUSIONS

To this point, numerous model scaled tests have been run and extensive experience on the dynamics of FOWT systems could thus be collected and have been summarized in this paper. The complex physics of the coupled wind-wave environment pose a strong challenge toward the scaling procedures. Past research projects have shown that various approaches with different levels of complexity for the combined Froude-scaled model test exist. As testing procedures are likely to become part of the certification process, two alternatives to the state-of-the-art procedure with blades modified according to Froude-scaling have been presented here in order to reduce the complexity of future test procedures. Whereas the SIL approach reduces the aerodynamics to a correctly reproduced thrust force by a fan the re-designed rotor aims at a match of power and thrust coefficients. Thus, the first solution will answer questions related to the full system dynamics whereas the latter allows also more detailed studies of the rotor aerodynamics and special effects therein. Eventually, the testing procedures for FOWT are significantly more complex than the common tests of naval architecture. It will be therefore essential to thoroughly select the adequate setup for each test and exclude features and effects where possible.
6 Bibliography


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**Wind Farm Active Power Control Based on Energy Storage System**

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**Keywords:** wind farm, energy storage system, active power control, rotor control, pitch control, real-time simulation

**Summary:** Wind farms are more and more required to act as a conventional generator with its penetration growing high in power system. First of all, the fluctuations rate of wind farm power output should be smoothed to a reasonable level. Secondly, wind farm should have the ability to response system frequency. Based on the technical characters of wind turbine rotor control, pitch control and Energy Storage System (ESS), a novel Active Power Control (APC) method is proposed in this paper, which utilizes the potential energy in wind turbines and ESS effectively. An APC control platform and the corresponding control algorithm of wind farms/ESS system are proposed, which coordinate the above control methods to accomplish the frequency regulation function or the fluctuations suppression function according to frequency deviation. The proposed method is verified through a real-time digital simulation system based on RTDS.

1 **Introduction**

Wind farms are more and more required to have the ability to control their active power output with the penetration in power system increasing[1]. To act as a conventional generator will be especially important for the oncoming large wind farm bases in China, where wind farms are constructed at large scale and transmitted through long distance. Wind farm Active Power Control (APC) will be more important for better performance of power system. APC includes the suppression of wind power fluctuations to limit ramp rate, and participating in primary frequency regulation to maintain system stability[2-4]. APC can make the operation of power system easier and of high efficiency especially for high wind penetration.

Wind power fluctuations suppression can be realized through the coordinated control of wind farms cluster, or the adoption of energy storages[5]. With the power output ramp rate maintained in a reasonable range, the requirement for the spinning reserve or backup power can be declined.

As to wind power frequency regulation, the wind turbines/farms control have not historically been required either for double feed induction wind turbine or direct driven wind turbine.
Moreover, the above variable speed wind generators do not intrinsically participate in system inertial response and frequency regulation automatically as would traditional generators since they are decoupled from grid frequency through power electronics[6]. With the increasing requirements for wind power participating frequency responses, various APC have been developed for both individual turbine and wind plant. Some wind turbine manufactures such as Siemens[7], Mitsubishi[8] and Vestas[9] have patented methods for dynamically modifying an individual turbine’s active power output though wind turbine rotor or pitch control. While wind plant control have focused on monitoring and forecasting the available power in wind turbine to control the total power aggregated in a wind plant[10-13]. Peng Li et al proposed a wind farm APC control strategies which employ energy storage to restrain wind power fluctuations when its generation is small, while participate in system frequency modulation and peak shaving when its generation is high[14]. Due to the stable and flexible operation of ESS, it can compensate the stochastic and intermittent power from wind farm and make the active power reserve more creditable.

In this paper, wind power APC with both fluctuations suppression and the frequency responses are presented to obtain a better performance for wind farm. It combines the control of wind turbines/farms and that of ESS[15]. Due to the high cost of ESS, it is located as the complementary of wind turbine control to achieve good APC technical economy performance. A novel control strategy is proposed which coordinate the operation of frequency regulation function and fluctuations suppression function according to the conditions of system frequency deviation and wind speed. Additionally, it can limit the high ramp rate when frequency deviation is small and frequency response is disabled to maintain frequency stability. A real-time digital simulation system based on RTDS is established to verify the proposed coordinated control method.

2 Proposed APC Platform Based on ESS

Fig.1 Wind farm/ESS APC control platform

Fig.1 shows the block diagram of the proposed wind farm APC control platform. It is top-level controller connected with the Transmission System Operator(TSO) or grid operator, utility grid, and the individual wind turbines. It comprises of power system status sample and assessment unit, wind farm operation status estimation unit, and wind/ESS coordinated control unit. It monitors the operation status of power system, the output power of the Point of Common Coupling (PCC), the operation status of individual wind turbines and ESS. The individual wind turbine can follow the power reference from wind farm controller by altering generator torque and/or blade pitch angle. ESS is enabled when the total power reserved shortage or pitch control delay. Through proper control, the instructions of operation adjustment of certain wind turbines and ESS are distributed to them instantaneously. As a
result, the output power of wind farm at PCC presents a better performance either for the output power rate or the frequency response ability.

3 APC CONTROL METHOD COMBINED WITH WIND FARM AND ESS

The proposed wind farm APC based on ESS are shown in Fig.2 where $\Delta f$ is frequency deviation. Power system frequency is sampled as a condition to determine the operation of the wind/ESS system. There is usually a frequency dead band for frequency regulation. If system frequency is within the dead band, the wind/ESS system frequency regulation function is disabled while the fluctuations suppression function is enabled. In case that frequency exceeds the dead band, the frequency regulation function is enabled and then the fluctuations suppression function is disabled. In this way, the transition between frequency regulation function and fluctuations suppression function can be implemented.

The frequency regulation function is enabled when system frequency deviation is beyond the dead band, as shown in Fig.3 where $P_{ESS}$ is power of ESS, $P_{overspeed}$ is the power obtained through rotor over-speed control, $P_{pitch}$ is the power obtained through pitch angle control, $v_{wind}$ is wind speed, $v_{wind,max}$ and $v_{wind,min}$ are the upper and lower wind speed set in the control respectively, and $\Delta P_{freq}$ the total power needed for frequency regulation.

Firstly the frequency deviation direction is judged. If under-frequency occurs, it means the wind/ESS system should increase active power output. Then the wind speed is used to determine the operation of wind rotor control, pitch control or ESS control. In low wind speed condition, since the potential power of wind rotor control and pitch control is limited, ESS is controlled to provide the required active power incremental. In mediate wind speed condition, both the wind rotor control and pitch control can be active through load shedding control such as rotor over-speed control and pitch angle control. ESS is controlled in this condition to complement the mismatch between wind turbine power output and the required. While in high wind speed condition, since the rotor over-speed control is disabled due to the mechanical stress limit, pitch angle control can provide the required power together with ESS eliminating its time delay. It is the same when over-frequency occurs.
The fluctuations suppression function is enabled when frequency deviation is within the dead band, as shown in Fig.4 where $\Delta P_{flu}$ is the power needed for fluctuations suppression. As we know, the smooth wind power profile will be helpful for system frequency stability and quality. Since frequency cannot always be controlled in its nominal value, it will swing along the nominal value continuously. Firstly frequency deviation direction is judged. If frequency is below the nominal value, the wind speed is then judged to determine the control process of wind/ESS system. Since the active power output increasing of wind power is helpful for system frequency moving to the nominal value from a lower state, the up-stream fluctuations of wind power can be neglected. However, the down-stream fluctuations should be taken care in this condition for a better active power balance of the whole system.

The objective of wind power fluctuations suppression can be obtained through a one-order low passing filter algorithm[10]. Wind speed condition is sampled to determine the operation of wind rotor control, pitch control and ESS control. In low wind speed condition, since the potential power of wind rotor control and pitch control is limited, ESS is control to provide the required power. In mediate wind speed condition, both the wind rotor control and pitch control can be active through load shedding control such as rotor over-speed control and pitch angle control. ESS is controlled in this condition to complement the mismatch between wind power output and the required. While in high wind speed condition, since the rotor over-speed control is disabled, pitch angle control provides the required power together with ESS to eliminate its time delay. It is the same when the frequency is above the nominal value.

Fig.3 Flow chart of wind/ESS frequency regulation function
4 SIMULATION AND DISCUSSION

A real-time simulation platform of wind farms with ESS is established in RTDS simulator. The WSCC 3-machine 9-line power system is taken for case study with wind farm and ESS integrated. To verify the effectiveness of the proposed APC method, three control schemes are taken in the real-time simulation, including scheme 1: wind farm do not participate in system frequency regulation and fluctuations suppression; scheme 2: wind farm participate in system frequency regulation and fluctuations suppression; and scheme 3: wind/ESS participate in system frequency regulation and fluctuations suppression.

a. Simulation with load changed and constant wind speed
In this condition, a stable wind speed of 10 m/s is given in the simulation. A load increment of 30 MW occurs at t = 4.1s. As can be seen from Fig. 5, system frequency decline to 49.7 Hz quickly when no wind/ESS control taken. It is improved through wind and wind/ESS frequency coordinated control respectively. The minimum value is 49.8 Hz and 49.9 Hz respectively and the stable state frequency deviation is reduced. It can also be seen that the insufficient power reserve of wind farm can be complemented by ESS in time.
Fig. 5 Frequency regulation process with load variation

b. Simulation with variable wind speed

Fig. 6 illustrates the simulation results under variable wind speed and different control schemes. In time area 1, the frequency deviation upward is rather high due to the increasing wind speed. The wind/ESS frequency regulation function is enabled under scheme 3 control framework. It can be seen that wind power output decreases, the fluctuation rate of the feed-in power at PCC decreases, and ESS is recharged. In time area 2, due to the frequency deviation is small, the wind/ESS frequency regulation function is disabled and the fluctuations suppression function is enabled. It can be seen that the feed-in power at PCC is quite smooth. In time area 3, the frequency deviation downward is rather high due to the decreasing wind speed. In this situation the wind/ESS frequency regulation function is enabled. It can be seen that the wind power output decreases, the fluctuation rate of the feed-in power at PCC
increases, and ESS is discharged. As also can be seen that scheme3 can be more effective with smaller frequency deviation and smoother feed-in power compared with that of scheme2.

Fig.6 Wind/ESS operation process with variable wind speed

5 CONCLUSIONS
With the development of wind power and its penetration in power system growing higher, better performance of wind farm is more and more required. Based on the technical characters of wind turbine rotor control, pitch control and energy storage, a novel APC method is proposed in this paper. It combines the utilization of the hidden energy in wind turbines and ESS. The APC control platform and the corresponding control methods of wind farm/ESS are introduced which coordinate the frequency regulation function and the fluctuations
suppression function according to frequency deviation. A real-time digital simulation system was setup and verified the feasibility of the proposed method. This study can be helpful for wind farms performance improvement, and the scalable development of wind power.

6 ACKNOWLEDGEMENT
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7 REFERENCES
Inertia emulation capability
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INERTIA EMULATION CAPABILITY FOR WIND POWER PLANTS HELPING THE ELECTRICAL GRID TO RECOVER FROM A TRANSIENT ON GRID FREQUENCY

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Keywords: wind turbine, synthetic inertia, grid frequency.

Summary: Frequency transients due to disconnection of power plants, load or part of the transmission systems causes a sudden change on grid frequency. There are several ways the systems correct those frequency perturbations. The first one is the natural inertial response from conventional synchronous generators, which deliver energy from rotational inertia when grid frequency changes. Therefore, this energy damps the frequency change until other regulation methods take action. However, variable speed wind turbines have not this natural inertial response. So increasing levels of variable speed wind generation reduce total system inertia and then severity of frequency perturbations can increase. This work presents a new wind turbine and wind power plant technology for emulation of this “inertial response” with variable speed wind turbines without a “recovery period”, characteristic of the response of other “synthetic inertia” proposals.

1 INTRODUCTION

Nowadays, the impact of wind turbines on the grid is no longer negligible. Therefore, network operators are making stricter wind turbine grid connection requirements. Some of these requirements focus on the transients in frequency that the wind turbines must be able to withstand without disconnecting from the grid and even react in order to help the system to recover the grid frequency to its rated values.

Particularly, transmission system operators are concerned about the frequency transients, generally caused by disconnection of big generators, loads or even part of the transmission system which causes a sudden change of the grid frequency out of its rated values, due to the mismatch between generated and consumed active power. This change on the grid frequency needs to be corrected quickly, so as to avoid that it gets out of its maximum range, which could imply a cascaded disconnection of generators and electrical consumers.

There are several layers of response of the generators and the grid in order to correct any sudden drop of the grid frequency. The first one is the natural response of the conventional generators. These generators can automatically deliver an instantaneous power, thanks to the rotational energy stored in their shafts. This is called inertial response. These synchronous machines are rigidly connected to the grid. This means that a change of frequency leads to an immediate and proportional change on the turbine rotational speed. The mechanical inertia of the generator plays an important role: inertia in a synchronous machine can be seen as a
conversion of kinetic energy of the rotating mass to electrical energy delivered to the grid, helping the grid to reduce the severity of a frequency drop.

This conversion of kinetic energy into electrical energy can be expressed as a source of active power generation. Since rotating energy is proportional to the square of the turbine rotational speed, which is proportional to the grid frequency, power delivered due to the change of frequency becomes negatively proportional to the rate of change of frequency.

When variable speed wind turbines are considered, this “natural” response to grid frequency is not present anymore. The electronic converter controllers used in these turbines decouples the grid frequency from the generator rotating speed [1]. Thus, wind turbines do not naturally modify their speed according to the grid frequency, absorbing or delivering energy from it moving rotor. Consequently WTGs do not damp the frequency drop [2].

Taking into account that wind power is starting to have an important penetration rate in the electrical system, the inability of delivering an inertial response leads to a reduction of the total inertia of the system. As a consequence, the severity of frequency drops can increase considerably.

2 SYNTHETIC INERTIA

As a solution for this lack of inertia, variable wind turbines controllers can be adapted to deliver an extra power when the frequency changes. This extra power $\Delta P_{\text{inertia}}$ can be a fixed value of power [3], a power proportional to the deviation of grid frequency from its rated value or to the rate of change of the frequency $df/dt$ [4], [5]. This last option matches with the emulation of a synchronous generator, which change its rotational speed the same amount of grid frequency. This implies that when frequency changes, rotational speed changes and the energy stored is delivered as a proportion to rate of change of frequency $df/dt$ according the following formula:

$$\Delta P_{\text{inertia}}=-2H\frac{df}{dt}$$

With $H$ is the energy stored in the kinetic energy stored in the rotation mass divided by generator rated power:

$$H=\frac{1}{2}J\omega^2/S$$

With $J$ the moment of inertia of the synchronous turbine, $\omega$ rotation speed and $S$ rated power.

Variable speed wind turbines have not this natural response because, in contrast with synchronous generators, power converters isolate grid frequency change from wind turbine rotational speed change.

This kind of wind turbines are composed of two controllers: wind turbine controller and converter controller:

- Wind turbine controller tries to keep the wind turbine at its optimal rotational speed in order to produce maximum power and secure operation. This controller is divided into a “pitch controller” and “torque controller”. Below rated power pitch controller operates pitch angle of the blade in order to extract maximum energy from wind and torque controller sends power references to the converter controller in order to assure that turbine speed is kept at its optimal rotational speed. At rated power operation, pitch controller operates blade angle to keep turbine rotational speed at its rated value.
while torque controller sends a power reference of rated power to converter controller.

- Converter controller acts electrically on converter and generator to generate the power reference from torque controller.

Wind turbines controllers can emulate synchronous generator inertial response by means of calculating this extra power and adding to the reference power from torque controller to converter controller (Figure 1). This is straight-forward when wind turbine operates at its rated operation because there is enough energy from wind to deliver this extra power. However, below rated operation power depends on wind speed. Then, the addition of this extra power involves a challenge in terms of wind turbine controller.

![Figure 1: Synthetic inertia addition to power reference to converter controller](image)

One approach to deliver synthetic inertia consists on calculating an extra active power negatively proportional to the rate of change of frequency. Then it is added to the wind turbine controller active power reference to the converter. However, power feed-in drops quickly thereafter [6]. This is due to the observed turbine rotor deceleration which provokes a decrease of the wind turbine controller active power reference due to the fact that the wind turbine controller controls the power according the deviation of turbine speed from a reference speed. As the turbine decelerates it deviates from reference speed and thus power should be reduced to recover to the reference speed. This is what some grid codes as [7] call “recovery period” after inertia is delivered. It introduces uncertainties on wind power plant behaviour in terms of frequency response and even can cause an undesired drop in frequency after this inertial response.

In a second approach, the wind turbine controller active power reference to the converter is hold to the value previous to the transient during a predefined time matching the expected duration of the frequency transient. An extra power is also added to this power reference. This extra power is calculated either as negatively proportional to the rate of change of frequency or proportional to the frequency deviation from rated frequency. Afterwards, it is added to the power which was hold previously [8]. However, during the time the over production is delivered turbine decelerates up to the point that the minimum speed is reached and then inertia delivery is stopped and power reference hold released. Then, there is a drop in production which is kept until turbine recovers its initial speed. This is again a “recovery period” which could cause an undesired drop in frequency after this inertial response.

In all explained cases there is a “recovery period”, which is dependent on the performance of the wind turbine before this period starts. Although during the inertia delivery time the grid frequency drop is damped, when the recovery period starts, the wind power plant drops its
power production below the value previous to the event. Consequently, during this “recovery period” the wind power plant can cause a second drop in grid frequency which has to be solved by another generation plant.

Another important issue regarding “synthetic inertia” is the calculation of the rate of change of frequency $df/dt$, because of the noise increase inherent to a derivate. In fact, frequency is a derivate of the grid voltage angle and therefore $df/dt$ is a second derivate.

3 CONTROLLER FOR SYNTHETIC INERTIA

The new controller faces the challenge of the recovery period associated to synthetic inertia as well as the noise on calculation of $df/dt$ and the physical limitations of the wind turbine. The technology developed emulates the natural response of the generator without the above mentioned recovery period issue. It is based on emulating through wind turbine controller performance of the synchronous generators based on the following main design concepts:

- $df/dt$ calculation is inherently noise. Direct derivation of the frequency would imply from equation (1) that there will be a noisy extra power generation when there is not a real frequency change. This issue is avoided, firstly, using low pass filter for the derivate and, secondly, using a dead band (Figure 2.a) which output a zero $df/dt$ when filtered $df/dt$ is lower than dead band (Figure 2.b).

![Diagram](https://via.placeholder.com/150)

**Figure 2:** Calculation of $df/dt$: (a) block diagram, (b) filtered value of $df/dt$ simulated inside dead band when no frequency changes

- Deliver an extra power negatively proportional to the rate of change of frequency according the equation (1). This extra is added to the power reference from torque controller to converter. In addition, equation (2) is substituted by (3) in which $H$ can be changed with a factor $K_{inertia}$

$$H = K_{inertia} \cdot \frac{1}{2} \cdot J \cdot \omega^2 / S$$

(3)

With $\omega$ is in this case the turbine rotational speed reference before the change of frequency.

In addition, this inertial extra power $\Delta P_{inertia}$ is higher and lower limited to a value $\pm \Delta P_{inertia,max}$ due to converter limitations. This would imply that the inertial energy would not be delivered in case of limitation but an “energy recovery” function is employed to deliver this energy after the limitation finishes.
Turbine rotational speed reference is updated in proportion to grid frequency change for pitch and torque controllers (Figure 3), according the equation (4). Therefore, turbine speed will change with frequency as in fact is done in a synchronous generator. This way, when the wind turbine delivers the extra power, turbine speed is changed in the same proportion than kinetic energy is delivered throughout the extra power production. In addition, this speed reference is modified with a transfer function $G(z)$ which models the system delays: mainly from $df/dt$ calculation, converter power generation time delay and speed measurement filters.

$$\frac{\Delta \omega_{ref}}{\omega_{ref}} = G(z) \cdot K_{inertia} \cdot \frac{\Delta f}{f_{rated}}$$

(4)

With $\Delta \omega_{ref}$ the turbine rotational speed reference change, $\omega_{ref}$ the turbine speed reference before the transient, $\Delta f$ grid frequency change and $f_{rated}$ rated frequency. Moreover, this reference speed change is limited according the minimum $\omega_{min}$, as seen in Figure 3, and maximum speed.

Figure 3: Turbine rotational speed reference change

4 RESULTS

4.1 SIMULATIONS

This controller is firstly validated by means of simulations with a wind turbine model comprised of a wind turbine controller with the same level of detail as in reality, a simplified model of converter controller and the physical models of the electrical subsystems and mechanical subsystems. The model is fed with a constant wind speed and a change on frequency using a look up curve.

Results are shown in Figure 4. Simulated frequency (Figure 4.a) is employed, firstly, to calculate the $df/dt$, which is shown slightly delayed in Figure 4.b due to the low pass filter and dead band. Secondly, the change on turbine speed reference is calculated according equation (4). This involves that turbine speed $\omega$ divided by initial value of turbine speed $\omega/\omega_0$ in Figure 4.c is delayed from frequency evolution $f/f_{rated}$ as seen in Figure 4.c.

The $df/dt$ calculation is employed to calculate, according equation (2) and (3), the $\Delta P_{inertia}$. Afterwards, it is applied added to the power reference in order to deliver the extra power generation as seen in Figure 4.d. Generated power differs slightly from reference due to the drive train oscillation and the delay of power converter for the following the referenced value of power.
4.2 TEST IN FIELD

This controller has been tested in field in a 2.0 MW rated power wind turbine. The grid frequency change is simulated by means of injecting a frequency curve, which bypasses the frequency measurement, into the WTG controller as explained in [9].

The first test result shown in Figure 5 in which wind turbine was operating at partial power and with $K_{inertia}=1$. A 1 Hz temporary drop in frequency is applied during 40 seconds (Figure 5.a) and $df/dt=±0.2$ Hz/s (Figure 5.b). The matching $\Delta P_{inertia}$ shown in Figure 5.d is followed by the generated power (Figure 5.e) and turbine speed (Figure 5.c) is changed in the same proportion as frequency (Figure 5.c). However, in contrast with the simulation, wind speed is changing. Therefore, during the 30 seconds that frequency is at 49 Hz turbine speed and power changes due to change on wind speed.
The second test result shown in Figure 6 in which wind turbine was operating at rated power, but limited its rated power to 1920 kW due to specific site limitations, and with $K_{\text{inertia}}=1$. Again, a 1 Hz temporary drop in frequency is applied during 40 seconds (Figure 6.a) with a higher $df/dt=\pm 0.8$ Hz/s (Figure 6.b).

The matching $\Delta P_{\text{inertia}}$ (Figure 6.d) is limited to the maximum $\Delta P_{\text{inertia max}}=\pm 200$ kW. Due to the “energy recovery” function the $\Delta P_{\text{inertia}}$ pulse lasts more than the 1.25 seconds in which $df/dt=\pm 0.8$ Hz/s. As wind turbine is operating at rated power turbine speed does not need to change according grid frequency and therefore it varies only due to wind speed changes (Figure 6.c). Finally, power (Figure 6.e) is increased and decreased according $\Delta P_{\text{inertia}}$ reference.

**Figure 6: Field test at rated power**

5 CONCLUSION

The controlled presented in this paper emulates the inertial performance of a synchronous generator, which delivers the energy from its rotational movement when a grid frequency change happens. This performance is critical to keep system stability when a transient unbalance between generation and consumption in the electrical system happens.

This controller emulates this inertial response by means of delivering a controlled extra power negatively proportional to the rate of change of frequency. This performance faces several challenges. The first one is the inherent noise on the measurement of the rate of change of frequency, which is solved with filtering and a dead band. The second challenge is the electrical limitations, solved with the “energy recovery” function which delays the delivery of the energy not delivered due to limitation. And finally, the main one is the “recovery period” which is presented in [6][7][8] and is solved by means of adapting in a controlled way the turbine speed according to the change of frequency.

Moreover, this inertia emulation controller has been also tested in simulation and with field tests showing that it complies with the expected performance.

Therefore, this controller overcomes the challenges of variable speed wind turbine technology to emulate inertia. Then, this inertia emulation has been proved to deliver inertia...
response equivalent as a conventional generator and, then, supporting the grid to damp the grid frequency change.

6 REFERENCES


BEHAVIOUR AND CONTROL OF DOUBLY FED INDUCTION GENERATORS IN WIND TURBINES DURING GRID FAULTS.

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Keywords: Grid Fault, Generator, Converter,

Summary: This paper describes the influence of grid faults on a doubly fed induction generator and the attached converter. It also presents a control strategy to minimize these effects without using a crowbar. Therefore keeping the converter generator system controllable.

1 INTRODUCTION

Doubly fed induction generators (DFIG) are commonly used in wind energy turbines. While the mechanical loads introduced in the drive train due to wind turbulent have been examined (e.g. in [12]). The influence of grid faults on the generator and the drive train is not very well understood. This paper examines the behaviour of the DFIG in case of grid faults, the effects of these grid faults on the wind turbine as well as possible control strategies.

2 MODULAR SIMULATION MODEL

In order to examine these effects, a modular matlab simulink model of the electrical system of a wind turbine has been developed at the IALB. The model can work in concert with a simplified matlab model of the mechanical part of the drive train developed at the IALB as well as a much more detailed multi body model of the mechanics. The modular model of the electrical system shown in Fig. 1 consists of a simplified mechanical model, a detailed model of the generator, a converter model, a detailed representation of the cable, the transformer and the grid.
a. Generator Model

A standard model of a DFIG [4] neglects iron-saturation and skin-effect. Some models consider exclusively skin-effect [2] or iron-saturation [9]. In fact both saturation and skin effect are affected in case of transient events. Depending on the shape of the rotor slots the rotor current is influenced by the skin effect. The amplitude of the currents affects the magnetization, in case of saturation the effective main inductance is reduced and therefore cannot be described as a constant parameter [5]. A mathematical model of the DFIG oriented on the main flux axis, describing the physical relationship between the magnetizing current $i_m$ and the main inductance in a direct way was presented in [10].

In the equations (1.1) to (1.4) $u_{Sx}, u_{Sy}$ and $i_{Sx}, i_{Sy}$ describe the direct and the quadratur part of the stator voltage and stator current in main field coordinates, $u_{Rx}, u_{Ry}$ and $i_{Rx}, i_{Ry}$ describe the direct and the quadratur part of the rotor voltage and rotor current in main field coordinates. $R_S$ and $R_R$ represent the resistances of the stator and the rotor windings, $i_m$ the amplitude of the magnetisation current, $L_h$ the main inductance; $L_{Sσ}$ and $L_{Rσ}$ the stator-, rotor- leakage inductance, $ω_m$ the main flux angular velocity, $ω$ the mechanical angular velocity and $p$ the number of pole pairs.

As can be seen in the equations (1.1) to (1.4) the description of the saturation is separated from the description of the skin effect. This allows an easy calculation of the dynamic behavior because saturation and skin effect are described independently. The new model allows the simultaneous use of the well known methods for calculation the skin effect [2], [13] and the main flux saturation [5], [9]. With these results the influences on the torque and the resulting dynamic loads on the drive train of the wind turbines can be described in a higher level of detail while still retaining easy to use methods.
b. Converter Model
The modelling of the converter was based on the PhD thesis of H. Raffel[8] and has been extended by pulse-width-modulated signals both for the grid side as well as the generator side of the converter. Depending on the configuration of the complete system model it is possible to vary the influence of the converter on the whole simulation. At the lowest detail level, only the time delay is considered. At higher levels the switching behaviour of both the grid side and the generator side converter are taken into account. Based on the standard control in stator flux coordinates optimized controllers for both converters were developed.

c. Cable and transformer Model
For the modelling the cable was divided into multiple parts of equal size. Each of these parts has the basic structure of a series oscillator circuit. These parts or segments interact via action and reaction. Each Segment comprises an inductance \( L_C \), a resistance \( R_C \), as well as a capacitance \( C_C \) and its parasitic resistance \( R_{PC} \). The length of the simulated cable can be varied by the number of segments. By the parameterization of each segment an aerial line can be simulated as well as an underground cable. In a first step for the simulations the three-phase cable was approximated by three screened cables. In order to expand the model to the simulation of an unscreened three-phase cable the inductance, resistance and capacitance in each segment can be replaced by matrices also comprising the coupling factors between the phases. Additionally a transformer model was developed. An important challenge was to ensure the compatibility of this model with the rest of the separate models as well as the complete model as a whole. Starting from the basic model of a transformer[3] the model was extended in order to consider additional effects. As with every other sub-model here also the needed computing capacity increases with higher complexity of the model and a balance has to be found between the required complexity and the computing capacity. With the available computational power not all details (e.g. the capacitive couplings between phases) could be simulated. However all significant aspect up to ca. 1 kHz signal frequency are represented by the model[3].

d. Grid Model
Furthermore a compatible simulation model of a three-phase grid was derived. The model can be used to simulate symmetric as well as asymmetric grid conditions. As is shown in [15], [16] an asymmetric system can always be split into three symmetric systems. These are the direct system, rotating in the same direction as the original three-phase system, the inverse system, rotating in the opposite direction and the zero system, which does not rotate. In a three-phase grid the zero system is not present[15], [16]. Also a symmetrical system can be considered a special case, where there is only a direct system.
The sub model of the grid allows for the simulation of different grid fault and the introduction of harmonics at an arbitrary point. The grid model comprises sinusodial voltage sources for the fundamental frequency as well as higher frequency sources, which can be added with different amplification to the fundamental frequency, thereby simulating harmonic contents with different amplitudes. Furthermore the three fundamental phase voltages can be attenuated in order to simulate voltage drops.
3 GRID FAULTS
For this work the consideration of the many possible grid faults was reduced to some
significant and characteristic cases. Among these are the single-phase and the three-phase
voltage drops shown in Fig. 2. At a specific point in time the voltage of one or all three phases
drops to a certain level and rises again to the original level after a predetermined amount of
time (in the example in Fig. 2 after 200ms).

![Fig. 2 exaplay grid fault](image)

4 EFFECTS ON THE GENERATOR
In order to evaluate the effects of grid faults on the generator, both simulations with the new
modular model and measurements on a test stand were conducted.

a. Simulation
In this section the results of simulations of both a symmetrical and a asymmetrical voltage
drop to about 60 % of the original value are shown. The influence of the grid fault can be seen
in the following figures.

![Fig. 3 Simulation of a symmetrical grid fault](image)

![Fig. 4 Simulation of an asymmetrical grid fault](image)
In Fig. 3 a) and Fig. 4 a) the time signals of the grid voltage, the rotor current and the shaft torque of the generator are shown for a symmetrical and an asymmetrical grid fault respectively. Fig. 3 b) and Fig. 4 b) show the spectrum of the rotor current both before (blue) and during the fault (red).

From the equations of the induced open-circuit voltage ($u_{R0}$) (1.5), (1.6) (cf. [14]) during the transition form one voltage level to the other the behaviour of the rotor current under load can be derived. Eq. (1.5) describes the rotor open-circuit voltage during a symmetrical grid fault.

Besides a stationary part comprising the slip frequency ($\omega_m - p\omega$), an additional, transient component with the mechanical frequency of the rotor exists,

\[
u_{R0} = \frac{L_{h(i_m)} p\omega}{L_{S(i_m)} \omega_m} U_2 e^{j(\omega_m - p\omega)t} - \frac{L_{h(i_m)}}{L_{S(i_m)}} \left(1 - \frac{p\omega}{\omega_m}\right) \left(U_1 - U_2\right) e^{-j\omega t} e^{-\frac{1}{\tau_S}}
\]  

(1.5)

In case of an asymmetrical grid fault, using the description by the symmetric components of the asymmetric fault, in addition to the stationary part of the direct system at the slip frequency ($\omega_m - p\omega$) and the transient part at the mechanical frequency, there appears a stationary part of the inverse system at the frequency -($\omega_m - p\omega$). Under load transient multiples of the grid frequency occure in the stator current, caused by retroactive effects [7]. This in turn causes transient components at frequencies ($n\omega_m - p\omega$). In case of asymmetrical grid faults also at frequencies -(n$\omega_m + p\omega$) where n=2,3,

\[
u_{R0} = U_{s,+} \frac{L_{h(i_m)}}{L_{S(i_m)} \omega_m} e^{j(\omega_m - p\omega)t} + U_{s,-} \frac{L_{h(i_m)}}{L_{S(i_m)}} \left(2 - \frac{p\omega}{\omega_m}\right) e^{-j(\omega_m + p\omega)t} 
- \Psi_{S,A0} \frac{L_{h(i_m)}}{L_{S(i_m)}} \left(1 + jp\omega\right) e^{-j\omega t} e^{-\frac{1}{\tau_S}}
\]  

(1.6)

b. Test Stand

The test stand emulates the behaviour of a wind energy plant. An induction motor driven by a frequency converter provides the torque and simulates the wind driven rotor blades. The mechanical to electrical energy transformation is done by a 22 kW doubly fed induction generator. Additionally the drive train comprises a single stage spur gear box. Additionally a replication of the connecting cable was developed and implemented at the IALB [12] At the machine lab of the IALB the dynamic of the voltage drop is limited to a fall time of about 15 ms. In order to compare the measurements additional simulations were done with this reduced dynamic. These are shown together with the measurement in the following section.

c. Measurements

Fig. 5 shows an exemplary result for a symmetrical voltage drop to 60 % of the original voltage. Shown are the measurements (red) in comparison to the corresponding simulation (black). As can be seen, both fit quite well and confirm the usability of the simulation model. At the machine lab of the IALB the dynamic of the voltage drop is limited to a fall time of about 15 ms. Additional faults with a higher fault dynamic in the falling flanks of the voltage drop not obtainable at the test stand were therefore examined in simulation. Fig. 5 shows in black a simulation of the same case as above but with a fall time of 15 ms.
5 CONTROL STRATEGY

As has been shown, the grid faults lead to high current peaks in the generator. Because
the converter is usually not designed for such high current peaks, the converter has to be
protected. This poses a challenge as switching off or using a crowbar gives up the control
of the system. According to the grid codes of the energy supply companies, depending on
the level of the voltage drop faults of 150 ms up to 1.5 seconds duration must be ridden
through and the system must stay controllable [11]. A novel control strategy in order to reduce
the current peaks and allow the converter to stay in active control of the system is presented
here.

By means of an FFT-Analysis of the rotor currents during a grid fault the relevant harmonics
were identified and dedicated controllers were developed in order to compensate for these.
The basic controller concept is shown in Fig. 6. Shown in blue is the standard current
controller. Drawn in red is the new control concept to eliminate the higher harmonics excited
by the fault.
Thereby the fault ride through capability of the system is reached. A continuous control of the DFIG is possible for symmetrical and asymmetrical grid faults. This forms the basis for all further load reducing concepts, as it is necessary for the converter to stay active in order to implement any kind of control of the system.

6 RESULTS

Fig. 7 a) and Fig. 8 a) show the effects of the control in case of a symmetrical and asymmetrical voltage drop to 60% of the original voltage with a fall time of 5 ms, respectively. Fig. 7 b) and Fig. 8 b) show the spectrum of the rotor currents both before (blue) and during (green) the fault. Comparing these to the uncontrolled results for the same fault shown in Fig. 3 and Fig. 4 it can be seen, that the additional frequencies induced by the voltage drop were successfully reduced by the new control. The time signals of the rotor current show significantly lower peaks in both cases.

7 CONCLUSION

In this paper a modular simulation model of the drive train of a wind turbine including the generator and converter with grid connection was presented. This model was used to examine the effects of grid faults on the converter / generator sub-system. Furthermore a new control strategy was developed in order to reduce these effects and thereby keep the generator controllable and connected to the grid during grid faults.
8 REFERENCES


NEW OFFSHORE SUPPORT STRUCTURES
INTERNATIONAL WIND ENGINEERING CONFERENCE IWEC 2014

EKKEHARD OVERDICK *, REINER KLATTE *
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Offshore Engineering Consultants
Cremon 32, 20457 Hamburg, Germany
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Keywords: Support Structures, Offshore Design and Offshore Transportation and Installation,

Summary:

This document summarizes trends for the design of new offshore support structures, their transportation and installation offshore proposed by various engineering companies as well as contractors.
Moreover actual significant offshore wind farm projects are highlighted by focusing on construction details, transportation and installation methods. This overview does not reflect to all new design proposals and is therefore limited to structures to be founded on the ocean floor, thus being bottom-fixed. A short outlook over current research programs is given as well.
1 **INTRODUCTION**

As evaluated by various studies carried for the identification of cost saving potential for the construction and installation of Offshore Wind Farms the main head line or findings are significantly clear:

All costs related to the complete installation of a wind park have to be reduced. As far as the support structures are concerned, these study requirements might only be reflected by introducing so-called “cost saving designs”. Some of these are highlighted within this paper.

2 **SHORT OVERVIEW, REVIEW AND COMMENTS ON NEW CONCEPTS**

Support structures and foundations for offshore wind turbines are designed in broad spread of solutions and party installed in open sea offshores areas. Designers are forced to consider a comprehensive set of boundary conditions which finally will rule each individual design proposal. Those may be but are not limited to:

- Operational water depth at individual site within a wind farm offshore
- Seabed geology and subsoil conditions
- Environmental impact by wave action, wind and ice, also acting in combination
- Construction requirement wrt to local facilities and construction sites
- Transportation and installation requirements in line with weather risks
- Environmental restrictions reflecting ocean born wild life
- Decommissioning after field life fulfillment
- Material supply and selection due to local facilities of production or recovery

Following definitions will be used within this paper:

- **Support structure**: defines the type of foundation and its method of fixing to the seabed
- **Foundation**: part of the support structure, considered to be the structural member from the transition piece or tower down to the ocean floor
- **Tower**: The tower is that structural part, which is part of the support structure carrying the nacelle on top of it. Due to load transfer requirements the tower may be connected by a transition piece
As shown below new concepts for support structure may be grouped or categorized as follows:

Category 1: Innovative Jacket Designs – piled/vibrated and suction can options
Category 2: XXL-Monopiles – piled or vibrated options in combination with piling
Category 3: Tri/Quadpods: Piled/vibrated or Suction Bucket designs
Category 4: Gravity Type Structures
EKKEHARD OVERDICK, REINER KLATTE

Fig. 1 Various design proposals for New Support Structures
Fig. 2 Proposed Classification of New Offshore Support Structures

<table>
<thead>
<tr>
<th>Category</th>
<th>Type of Support Structure</th>
<th>Foundation design</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Jacket / Lattice Type</td>
<td>Piling/ Suction Cans</td>
</tr>
<tr>
<td>2</td>
<td>XXL-Monopiles</td>
<td>Piling /Vibration or Combination</td>
</tr>
<tr>
<td>3</td>
<td>Tri / Quad-Pods</td>
<td>Piling /suction Cans</td>
</tr>
<tr>
<td>4</td>
<td>Gravity Structures</td>
<td>By Gravity</td>
</tr>
</tbody>
</table>
2.1 Innovative “Twisted Jacket” Design

The “Twisted Jacket” design as introduced by KEYSTONE Engineering [1], also named as IBGS denoting the abbreviation for INWARD BATTERED GUIDE STRUCTURE, has been selected to demonstrate its performance in US waters. Battered piles are twisted around a central column, to be treated as a predrilled structural member of jacket. Designer announced this type of structure may be manufactured easier and installed than conventional steel jackets. Dominion Virginia Power and Fishermen’s Energy will use this type in US-department of Energy projects:

Up to now two similar IBGS foundation concepts have been installed to support drilling platforms in the Gulf of Mexico in 2005:

- ExxonMobil West Delta 30 Field Redevelopment Project
- West Delta 30 BB

Fig.3 Twisted Jacket for Offshore Wind [1]  Fig.4 FE-model of IGBS [1, 2]
Fig. 5    Installation Method of IBGS-Jacket as proposed by KEYSTONE Engineering [2]
Fig. 6 Comparison of combined stresses: twisted type vs. un-twisted. [2]
It should be mentioned that within an F+E-analysis carried out by Annabel Lush, Durham University, UK [2] and simulating loading behavior derived for offshore wind power application the advantage of the twist design could not clearly be shown. Removing the twist in the IBGS increases the stiffness under analyzed load cases, as performed by Lush.

Actually, Keystone Engineering's 'twisted jacket', one of four finalists from the Carbon Trust's Offshore Wind Accelerator (OWA) foundations competition, has become the first design to be installed through the OWA in a demonstration project jointly funded by Mainstream Renewable Power, DONG Energy and the Carbon Trust to be used as a Metocean Tower at Location Horns Rev, North Sea.
The demonstration is a significant step on the route to commercializing the structure for use with turbines. The project incorporates a comprehensive monitoring program and will provide valuable insights into optimizing the structure for fabrication and installation.

The 'twisted jacket' uses less steel than a conventional jacket and is easier to fabricate, using fewer components. Installation costs will also be lower because more units can be fitted onto an installation barge due to its small footprint.

The Offshore Wind Accelerator foundation competition was launched in 2009 to find cheaper foundations for 30-60m water depths and attracted 104 entries from all over the world. Over the last 18-months, the Carbon Trust and the eight OWA members: DONG Energy, E.ON, Mainstream Renewable Power, RWE, Scottish Power Renewables, SSE, Statkraft and Statoil have invested in the conceptual design and de-risking of four finalists, and undertaken a program of work to assess the impact on the cost of energy of each design.

The foundation supports the first met mast to be installed for Round 3 and the data collected will inform the wind farm design and energy production estimates. The 4GW Hornsea zone is being developed by SMart Wind, the joint venture between Mainstream Renewable Power and Siemens Project Ventures.

2.2 XXL-Monopiles

Being motivated by a successful installation of 22m in diameter steel tubular piles in China in 2011 in line with foundation works for the Hong Kong-Zhuhai-Macau Bridge by vibratory hammers [3] this technical success was picked up for a feasible installation method by offshore construction contractors as well as by offshore wind farm developers.

Fig.9 Vibratory driving of a 22m diameter pile in Hongkong, China
Consequently the Riffgat Offshore Wind Farm, German North Sea, was erected using steel monopiles of a so-called XXL-category with varying diameter from 4.7 m at top and max. 6.5m at pile tip. Corresponding weights range from 480 to 720 mt.

Fig. 10 Windfarm Riffgat Monopile Installation

It should be noted that due to concern of a soil strength degradation by vibratory driving, the last 10m of the monopoles still have to be impacted by hammers. At Riffgat an IHC S 1800 hydraulic hammer went into operation.
2.3 Suction Pile Foundations

Actually there are some promising new designs under evaluation and verification as well. DONG Energy may be seen as a pioneer since their prototype, a jacket type structure, founded by 3 suction buckets, shall be installed in mid 2014 in 23m water depth. Location foreseen is the Borkum Riffgrund 1 wind farm location in the German sector of the North Sea. A SIEMENS 2.8 MW turbine shall be fixed to the supporting structure.

![Fig.11 DONG Energy’s prototype jacket with suction buckets](image)

A joint industrial research project is actually ongoing, conducted by IWES-Fraunhofer Society and participation of relevant institutes of LEIBNIZ University Hannover, joined by some industrial partners like SENVION, OVERDICK and others. The jacket designed is foreseen to operate in 40m of water depth and shall be equipped with an 8 MW turbine. Structural design engineering works are nearly completed whereas geotechnical engineering is still under evaluation. Geotechnical large scale model tests are foreseen to be started in the end of 2014. New test facilities at Fraunhofer Test Center near Hannover shall be used.
Fig 12 Design of a Quadjacket with suction buckets in 40m of water depth, 8 MW turbine application, designed for North Sea conditions, R+D-project
2.4 Reinforced Concrete Gravity Types

Looking at new ideas in line with gravity type structures a so-called cranefree gravity foundation is offered by SEATOWER, a Norwegian company. In fact the idea is not new to have a gravity design to be installed without any specialized ships or crane vessels. Running a short review on relevant installations e.g. in the North Sea or Baltic Sea all these platforms were installed cranefree, using built-in buoyancy creating sufficient hydrostatic stability during offshore transportation and touch down sequences at location. Of course, the advantage of this technique is that conventional tugs can be used equipped with relevant pumping power packs and other auxiliary equipment. This certainly will have a positive impact on T & I costs for investments of Offshore Wind Farms.

![Fig.13 Concrete Gravity Platform in 1983, Deutsche Texaco, Germany](image1)

![Fig. 14 Seatower Design, 2014](image2)

3 Design Outlook and Recommendations for Further Research

Future design works will be influenced by focusing all efforts for the reduction of overall cost, especially on the CAPEX side. So, pile driving by vibration technique seems to have a promising cost saving potential, in line with the upcoming use of jackets with suction bucket foundations and their geotechnical verification for use in line with Offshore wind turbine support structures; it should be noted
that up to now the installation of transformer or converter stations has to be treated as “state of the art”.

Fig. 15 Transformer Station GLOBAL Tech 1, based upon MOAB-Design with Suction Cans

Further research shall be concentrated on installation methods of XXL- monopiles by vibrating and the use of suction pile foundation in any combination with support structures.

4 REFERENCES

[1] KEYSTONE Engineering, Houston, USA, The twisted jacket design, inhouse publication, 2014


Opening of
TEST BENCH FOR GENERATORS AND CONVERTERS (GeCoLab)
DO WE NEED RESEARCH?
September 2014 / Matthias Deicke
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   Concept

4. HybridDrive Project
   How we address LCoE

5. Prototype HybridDrive
   Prototype testing

6. Prototype HybridDrive
   Field measurement

7. Development
   Next steps

8. Summary
   Outlook and Questions
**Mission**

“Winergy will maintain its position as leading supplier of reliable wind drive train components. This will be accomplished through long-term partnership with our customers.”

---

**Market Position**

- Global market leader for drive train components
- 100 GigaWatt installed base
- Every 3rd wind turbine is equipped with Winergy components

---

**Customer Base**

- Vestas
- Siemens
- GE Wind Energy
- ALSTOM
- Acciona
- Repower
- Envision
- Kenersys
- Suzlon
- Nordex
- Gamesa Edifa
- Goldwind
- Sinovel
- BHD
- Gamesa Edifa
- Suzlon
- Goldwind
- Sinovel
- BHD

Trustworthy partnership with all major wind turbine OEMs since 1981

---

**Global Footprint**

Winergy has production and service facilities in all major markets around the globe.

[Map showing global footprint with major markets such as USA, China, India, and Europe marked.]

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<table>
<thead>
<tr>
<th>Drive train concept</th>
<th>Technical facts</th>
<th>Facts of interest</th>
</tr>
</thead>
<tbody>
<tr>
<td>Planetary stages drive train</td>
<td>Two-/three-stage gearboxes&lt;br&gt;Winergy set standards in wind industry&lt;br&gt;500 kW – 7.5 MW&lt;br&gt;i = 90 … 110&lt;br&gt;Average: 1.5 – 2.5 MW</td>
<td>Proven technology&lt;br&gt;Highest market share&lt;br&gt;Extensive track record (&gt;70,000 gearboxes supplied)</td>
</tr>
<tr>
<td>Multi Duored</td>
<td>High power density&lt;br&gt;Great serviceability&lt;br&gt;Up-tower service and maintenance&lt;br&gt;Two generators for variable output power and speed&lt;br&gt;8-fold power splitting&lt;br&gt;i = &gt;&gt;100&lt;br&gt;5 MW – 12 MW</td>
<td>First installation of two prototypes in Bard wind turbines&lt;br&gt;Duored technology known for many years in industry</td>
</tr>
<tr>
<td>HybridDrive</td>
<td>Highest overall drive train efficiency&lt;br&gt;Modular set-up for easiest serviceability&lt;br&gt;Super compact for flexible concepts&lt;br&gt;3 MW – 7.5 MW&lt;br&gt;i = 35 … 42</td>
<td>First two prototypes with Fuhrländer/ W2E Wind to Energy GmbH&lt;br&gt;3.0 MW prototype installed in a wind turbine in Sep. 2013</td>
</tr>
</tbody>
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Stand: Nov. 2013

Stromgestehungskosten [Euro\textsubscript{2013}/kWh]

- Photovoltaik: PV klein bei GHI = 1000 kWh/m\textsuperscript{2}/a, PV frei bei GHI = 1200 kWh/m\textsuperscript{2}/a, PR = 85%, mittlere Marktentwicklung
- Wind Offshore: VLS von 2800 bis 4000 h/a, PR = 95%, mittlere Marktentwicklung
- Wind Onshore: VLS von 1300 bis 2700 h/a, PR = 97%, mittlere Marktentwicklung
- Biogas: VLS von 6000 bis 8000 h/a, PR = 100%
- Braunkohle: Vollaststunden, Brennstoffkosten, Wirkungsgrade, CO\textsubscript{2}-Preise abhängig von Betriebsjahr, vgl. Tabelle 4-7
- Steinkohle: Vollaststunden, Brennstoffkosten, Wirkungsgrade, CO\textsubscript{2}-Preise abhängig von Betriebsjahr, vgl. Tabelle 4-7
- GuD: Vollaststunden, Brennstoffkosten, Wirkungsgrade, CO\textsubscript{2}-Preise abhängig von Betriebsjahr, vgl. Tabelle 4-7

Source: IWES / Fraunhofer
**LCoE formula:**

\[
LCoE = \sum_{t=0}^{n} \frac{Investments_t}{(1+WACC)^t} + \sum_{t=0}^{n} \frac{O \& M \_ Costs_t}{(1+WACC)^t} + \sum_{t=0}^{n} \frac{Fuel \& CO2 \_ Costs_t}{(1+WACC)^t} + \sum_{t=0}^{n} \frac{Energy \_ Output_t}{(1+WACC)^t}
\]

LCoE considers costs and electricity generation over the whole lifetime of a power plant.

**WACC:** Weighted Average Capital Cost

**O&M:** Operation and Maintenance

**n:** lifetime of power plant
LCoE formula (simplified):

\[
LCoE = \frac{\sum_{t=0}^{n} Investments_t}{(1 + WACC)^t} + \sum_{t=0}^{n} \frac{O & M \_ Costs_t}{(1 + WACC)^t} + \sum_{t=0}^{n} \frac{Fuel & CO2 \_ Costs_t}{(1 + WACC)^t} - \sum_{t=0}^{n} \frac{Energy \_ Output_t}{(1 + WACC)^t}
\]

For renewables NO costs of Fuel have to be considered.
LCoE formula (simplified):

\[
LCoE = \frac{\sum_{t=0}^{n} Investments_t}{(1+WACC)^t} + \frac{\sum_{t=0}^{n} O \& M \_ Costs_t}{(1+WACC)^t} - \frac{\sum_{t=0}^{n} Energy\_Output_t}{(1+WACC)^t}
\]
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## HybridDrive Drive Train Matrix

<table>
<thead>
<tr>
<th>Speed Category</th>
<th>Fixed speed</th>
<th>Variable speed</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>High speed</strong> 700 – 2000</td>
<td>Asynchronous</td>
<td>Asynchronous</td>
</tr>
<tr>
<td></td>
<td>Squirrel cage</td>
<td>Squirrel cage and converter</td>
</tr>
<tr>
<td></td>
<td>Pole changing</td>
<td>DFIG and converter</td>
</tr>
<tr>
<td></td>
<td></td>
<td>DFIG and resistor in rotor circuit</td>
</tr>
<tr>
<td><strong>Intermediate speed</strong> 100 – 700</td>
<td></td>
<td>Electrically excited generator and converter</td>
</tr>
<tr>
<td><strong>Low speed</strong> 10 – 100</td>
<td></td>
<td>PEM generator and converter</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Electrically excited generator and converter</td>
</tr>
<tr>
<td></td>
<td></td>
<td>PEM generator and converter</td>
</tr>
</tbody>
</table>
HybridDrive
General Concepts

2-stage gear technology

PMG or EESG medium speed generator

Winergy HybridDrive

- Low weight
- High efficiency
- Serviceability
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Great Serviceability through a Modular Design

- A design comprising three modules allows the individual elements to be simply removed/installed:
  - 1st gear stage
  - 2nd gear stage
  - generator
- The internal service crane of the nacelle can be used to transport these modules instead of employing an external crane.

Benefits

Reduction of service complexity and costs for service work
HybridDrive
Rolling Bearings or Journal Bearings

or

Gearbox with journal bearings
- in operation
- promising results
- customer benefits
- field test running
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HybridDrive 3D-Model

Concept 2011

Prototype 2013
HybridDrive Test Procedure/General Set Up

- Certification guideline GL2010
- Component Certification
- Type test before Sep. 2012

INVERTER NOT WINERGY’s SCOPE of SUPPLY
Full system test:
- 2 HybridDrives
- Back-to-Back
- Full Load
- Heat Run
- Vibration
- Noise
- Functional Test
- Start-Up Procedure
HybridDrive
Test Procedure/FULL System Test

Efficiency Measurement
Noise/dB(A)
Temperature Rise
Vibration/Speed
Noise 1/3 Octave
Conclusion
HybridDrive
Temperature Rise Test

- Different load points
- All temperatures below specified limits

<table>
<thead>
<tr>
<th>Nr./No.</th>
<th>Vorgang / Item</th>
<th>Laststufe / Load stage</th>
<th>Probelauf / Test run</th>
</tr>
</thead>
<tbody>
<tr>
<td>490</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td></td>
<td>7.x</td>
<td>280</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>8.x</td>
<td>415</td>
</tr>
</tbody>
</table>

Different load points and all temperatures below specified limits.
• System efficiency @ rated point >96 %
• Very high efficiency @ part load

TEST passed
HybridDrive
Vibration Test/T1-T4/Acceleration [m/s²]

<1 m/s² in all conditions

TEST passed
HybridDrive
Noise Measurement

LESS noise
than other
systems

Rotor speed (vs. Torque)

TEST
passed
## Noise Measurement

### Schallintensitätspektrum

**Sound intensity spectrum**

1/3 Oktave

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>dB(A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>20</td>
</tr>
<tr>
<td>125</td>
<td>40</td>
</tr>
<tr>
<td>160</td>
<td>60</td>
</tr>
<tr>
<td>200</td>
<td>80</td>
</tr>
<tr>
<td>250</td>
<td>100</td>
</tr>
<tr>
<td>315</td>
<td>120</td>
</tr>
<tr>
<td>400</td>
<td>140</td>
</tr>
<tr>
<td>500</td>
<td>160</td>
</tr>
<tr>
<td>630</td>
<td>180</td>
</tr>
<tr>
<td>800</td>
<td>200</td>
</tr>
<tr>
<td>1000</td>
<td>220</td>
</tr>
<tr>
<td>1250</td>
<td>240</td>
</tr>
<tr>
<td>1600</td>
<td>260</td>
</tr>
</tbody>
</table>

**Max. Value**
- **Spektrum Analysis**: 76.8 dB(A)
- **L_SM**: 14.6 dB(A)
- **L_WA**: 91.4 dB(A)

**Next Value**
- **Messabstand**: 0.10 m
- **Höhe**: 2.08 m
- **Breite**: 1.98 m
- **Tiefe**: 2.78 m

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>n_1</td>
<td>12.76 min^-1</td>
</tr>
<tr>
<td>n_2</td>
<td>453.30 min^-1</td>
</tr>
<tr>
<td>Ölzviskosität</td>
<td>Ölviskosität</td>
</tr>
<tr>
<td>Belastung</td>
<td>Load</td>
</tr>
<tr>
<td>320</td>
<td>VG</td>
</tr>
<tr>
<td>2987.0</td>
<td>kW</td>
</tr>
<tr>
<td>Drehrichtung D1</td>
<td>Direction of rotation</td>
</tr>
<tr>
<td>links (cw)</td>
<td></td>
</tr>
<tr>
<td>rechts (cw)</td>
<td>X</td>
</tr>
</tbody>
</table>

**TEST passed**
Conclusion:

- HybridDrive tests finalized
- Test results as expected or slightly better
- Heat run test ➔ o.k., as calculated
- Efficiency ➔ slightly better than calculated
- Noise ➔ better than calculated
- Vibration ➔ better than expected

- HybridDrive system test successfully completed.
- First prototype has been installed and commissioned.
- Second prototype is under commissioning.
- Field measurements have been started.
HybridDrive
First Prototype/Facts

<table>
<thead>
<tr>
<th>Project name:</th>
<th>T10x</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type:</td>
<td>W2E-120/3fc</td>
</tr>
<tr>
<td>Design:</td>
<td>IEC 61400-12 IEC 2a</td>
</tr>
<tr>
<td>Power:</td>
<td>3MW</td>
</tr>
<tr>
<td>Rotor diameter:</td>
<td>120 m</td>
</tr>
<tr>
<td>Tower:</td>
<td>Steel 100 m</td>
</tr>
<tr>
<td></td>
<td>Concrete hybrid 140 m</td>
</tr>
</tbody>
</table>

Source: W2E Wind to Energy

Statement of Compliance
Design evaluation of HybridDrive PZFG 2630

Name and address of the manufacturer:
Winergy AG
Ase Industriepark 2
46452 Verden
Germany

Component type:
Drive train HybridDrive PZFG 2630

Scope of evaluation:
The Statement of Compliance attests compliance of the design with the following standards:
"GL Guidelines for the Certification of Wind Turbines", Edition 2015, concerning the design.
The conformity evaluation of the A-Design was carried out according to the referenced standards listed in the relevant certification reports mentioned below.

Certification reports:
- Load Assumptions (1891003668805.4 2013-05-05)
- Machinery Components (1891003668805.7 2013-05-05)
- Electrical Equipment (1891003668805.3 2013-05-05)

Validity:
Any change in the design has to be approved by TÜV Rheinland. Certification Body for Wind Turbines.
Without approval, the Statement of Compliance loses its validity.

Cologne, 2013-04-02
LV. Karl Friedrich
LV. Frank Witter
TÜV Rheinland
Verden, 04.04.2013

www.w2e.com
HybridDrive Development Process

- Wind turbine requirements/specification
- V-model
- product

OEM prototype test

Prototype AND serial test
- e.g. generator test IEC 34
- e.g. Gbx, Gen
- e.g. mainbearing
- e.g. steel

IDS

- component*)
- part
- material
- component wind
- component C

IDS: integrated Drive System

*) main component or sub system

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Uptower verification of parameters

- Input torque
- Electric power
- Efficiency
- Vibration/noise
- Temperatures
- Displacement

Remote system has been installed / Verification is on-going.
1. **Company Profile**  
   Facts and figures

2. **LCoE**  
   Design Drivers

3. **HybridDrive Design**  
   Concept

4. **HybridDrive Project**  
   How we address LCoE

5. **Prototype HybridDrive**  
   Prototype testing

6. **Prototype HybridDrive**  
   Field measurement

7. **Development**  
   Next steps

8. **Summary**  
   Outlook and Questions
HybridDrive
... a modular drive train concept

Same color = same component, gearbox remains identical.
**Some differences:**

- EESG 20% heavier than PMG
- EESG slightly “bigger”
- PMG water-jacket cooled
- EESG with top-mounted cooler
- EESG has a higher short-circuit current / short-circuit torque compared to PMG
- EESG needs excitation system brushless: exciter machine or with brushes: inverter
- Time constant of EESG is longer compared to PMG
- PMG uses rare earth magnets, EESG does not
- EESG can be operated with 2Q inverter system.
Possible configuration:
- Medium-speed EESG with rectifier up-tower
- DC link in the tower
- Grid-side converter and transformer down-tower

Benefits:
- Compact nacelle
- Less power electronics
- Reduced cable costs

Challenges:
- Control
- Drive train damping
- “Constant” DC voltage
1. **Company Profile**  
   Facts and figures

2. **LCoE**  
   Design Drivers

3. **HybridDrive Design**  
   Concept

4. **HybridDrive Project**  
   How we address LCoE

5. **Prototype HybridDrive**  
   Prototype testing

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   Field measurement

7. **Development**  
   Next steps

8. **Summary**  
   Outlook and Questions
HybridDrive
First Prototype / T10x / 3 MW / 120 m

Source: W2E Wind to Energy
<table>
<thead>
<tr>
<th>Speed Range</th>
<th>Fixed speed</th>
<th>Variable speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>High speed</td>
<td>Asynchronous Squirrel cage</td>
<td>Asynchronous Squirrel cage and converter</td>
</tr>
<tr>
<td>700 – 2000</td>
<td>Squirrel cage Pole changing</td>
<td>Electrically-excited generator and converter</td>
</tr>
<tr>
<td>Intermediate speed</td>
<td>Electrically-excited generator and converter</td>
<td>PEM generator and converter</td>
</tr>
<tr>
<td>100 – 700</td>
<td>DFIG and resistor in rotor circuit</td>
<td>DFIG and converter</td>
</tr>
<tr>
<td>Low speed</td>
<td>PEM generator and converter</td>
<td>PEM generator and converter</td>
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<tr>
<td>10 – 100</td>
<td>Electrically-excited generator and converter</td>
<td>Electrically-excited generator and converter</td>
</tr>
</tbody>
</table>
HybridDrive
Project Review

2009
- Start of internal R&D project / ENTSO-E Network Code

2011
- First Order: 3 MW customer Fuhrländer
- Inhouse prototype testing

2012
- Delivery of first prototype
- First IQPC presentation
- Launched into the market at the Hanover Fair

2013
- Commissioning (October)
- Component certificate
- Husum Wind Fair: Ready for delivery
- Fuhrländer bankruptcy

2014
- 2nd prototype
- Frost & Sullivan
- EESG as alternative
- Journal bearings

2015
- 0-Series

FUTURE
HybridDrive: System Features and Benefits

Outline
(new, with integrated oil tank)

Features / Customer benefit

HybridDrive: Torque in -> Electricity out
incl.: Gbx, Gen, OSS, Sensors

Main data:
- Torque: 2688 kNm
- Electric power: 3600 kW @ 720 V
- Length: 2935 mm
- Weight: ~ 34 tons (79 Nm / kg)
- Efficiency: >96.5% @ rated power

Benefits:
- Completely tested
- System responsibility
- Improved logistic concepts possible
First prototype installed in September 2013

First kWh produced in October 2013
HybridDrive
Inauguration of Prototype 12.10.13

Prototyp einer 3-MW-Windenergieanlage

Mit 10.000 MWh/Jahresertrag versorgt die Anlage 1.615 Vier-Personen-Haushalte.

Robordurchmesser: 120 m
Nebenhöhe: 100 m
Gesamthöhe: 160 m

Entwickler der Technologie: W2E Wind to Energy GmbH
Betreiber: W2E Windpark UG (haftungsbeschränkt) Kankel KG

Verbundpartner:
Lehrstuhl für Technische Mechanik/Dynamik
 der Universität Rostock
Thank you for your Attention!

Matthias Deicke
Winergy – Head of Electrical Systems

Siemens AG
Am Industriepark 2
46562 Voerde

Phone: +49 (2871) 92-1054
Fax: +49 (2871) 92-2487

E-Mail: matthias.deicke@siemens.com
Erndtebrücker EisenWerk

Key supplier for offshore wind foundations
Unique Product Range
ensures best solutions for our customers

Global Player
we are always close to our customers worldwide

more than
75 years of experience
EEW Group Organisation

EEW Erndtebrück

EEW SPC Rostock

EEW Korea

EEW Malaysia

Global Pipe
Ownerschip Structure

100% Family Owned
Erndtebrücker Eisenwerk GmbH & Co. KG

50% Family Dietze

50% Family Schorge

50% Christina Dietze

25% Christoph Schorge

25% Joerg Schorge
History

1936  Foundation
1974  Start SAW Pipe Production
2001  EEW Korea
2008  Foundation of EEW SPC
2009  Foundation of WELDEC
2009  EEW Pickhan joins the Group
2010  Start of EEW Malaysia
2013  Foundation of (EEW) Global Pipe Company
Revenues by Industrial Sector

Average of the last three Fys

- Offshore Wind: 39%
- Plant Construction: 26%
- Offshore Oil & Gas: 23%
- Pipelines: 8%
- Power Plant Construction: 2%
- Other: 1%

Total revenue EEW Group: € 606 Mio. in BY 12/13

Source: EEW-E Controlling
Mill in Rostock

- Property: 239,000 m²
- Production floor: 29,600 m²
- Production capacity: 170,000 MTPA
- Employees: 430
Technical Capabilities

Total Length < 120 m

Outside Diameter
0.4 m – 10 m

Wall Thickness
8 mm – 150 mm

Total Piece Weight < 1.500 t
# Products of EEW for Offshore Wind

<table>
<thead>
<tr>
<th>Product</th>
<th>Supply Chain</th>
<th>Production site</th>
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<td>MP‘s/TP‘s</td>
<td>Utilities/EPCI‘s ↓ Jacket Fabricator ↓ EEW</td>
<td>Rostock</td>
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<td>Jackets</td>
<td>Utilities/EPCI‘s ↓ Jacket Fabricator ↓ EEW</td>
<td>Erndtebrück Korea</td>
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<tr>
<td>Substations</td>
<td>Utilities/EPCI‘s ↓ Jacket Fabricator ↓ EEW</td>
<td>Erndtebrück Rostock</td>
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</table>
# EEW Group – OWF Projects

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<thead>
<tr>
<th>Year</th>
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<th>Rostock</th>
<th>Jackets</th>
<th>Erndtebrück Korea Rostock</th>
<th>Substations</th>
<th>Erndtebrück Rostock</th>
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<tr>
<td>2009</td>
<td>Belwind</td>
<td>20.640 t</td>
<td>Ormonde Pin Piles</td>
<td>7.350 t</td>
<td>Alpha Ventus</td>
<td>3.620 t</td>
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<tr>
<td></td>
<td>Baltic I</td>
<td>8.860 t</td>
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<tr>
<td>2010</td>
<td>Walney I + II</td>
<td>86.650 t</td>
<td>Thornton Banks Jacket &amp; Piles</td>
<td>22.720 t</td>
<td>BorWin/ HelWin</td>
<td>7.200 t</td>
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<tr>
<td></td>
<td>London Array</td>
<td>69.840 t</td>
<td>Nordsee Ost Jacket &amp; Piles</td>
<td>34.600 t</td>
<td>Borkum Riffgrund</td>
<td>2.730 t</td>
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<td>2011</td>
<td>Meerwind TP’s</td>
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<td>SylWin Alpha</td>
<td>4.940 t</td>
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<td>Dan Tysk MP’s</td>
<td>14.500 t</td>
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<td>Amrumbank</td>
<td>1.360 t</td>
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<td></td>
<td>Gwynt y Mor</td>
<td>110.020 t</td>
<td></td>
<td></td>
<td>Butendiek</td>
<td>710 t</td>
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<td>2012</td>
<td>Butendiek</td>
<td>73.000 t</td>
<td>Baltic II Jacket &amp; Piles</td>
<td>22.510 t</td>
<td>HelWin II</td>
<td>2.490 t</td>
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<td>Northwind MP’s</td>
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<td>Baltic II Pin Piles</td>
<td>19.580 t</td>
<td>Humber Gateway</td>
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<td></td>
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<td>DanTysk</td>
<td>770 t</td>
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Revenues OWF

<table>
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<tr>
<th>Country</th>
<th>Wind Farms</th>
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<tbody>
<tr>
<td>EEW SPC</td>
<td>Bard I, Belwind, Baltic I+II, Walney I+II, London Array, GT 1, BW II, Nordsee Ost, Meerwind, Dan Tysk, Gwynt y Mor, Butendiek, Northwind</td>
</tr>
<tr>
<td>EEW KG</td>
<td>Thornton Banks, Amrumbank, Westermost Rough, Butendiek, Baltic II (Jackets), Humber Gateway, DanTysk, Ormonde</td>
</tr>
<tr>
<td>EEW Korea</td>
<td>Ormonde, BW II, Thornton Banks</td>
</tr>
<tr>
<td>EEW Malaysia</td>
<td>GT 1, BW II</td>
</tr>
</tbody>
</table>

![Bar chart showing revenues for different periods and countries]
Fabrication Key Figures

Fabrication Capacity Rostock Factory

Fabrication and assembly of longitudinally welded, thick-walled carbon steel pipes:

- Can fabrication
  - Hall 1 & 3
- Pipe assembly
  - Hall 2, 3 & 5
- Pipe diameter:
  - 1.5 m to 10 m
- Pipe length:
  - up to 120 m
- Steel plate thickness:
  - up to 170 mm
- Single pipe weight:
  - up to 1,500 t
- Total capacity/year:
  - approx. 170,000 t

Offshore Coating (company Krebs)

- 2-3 MP’s/TP’s per week
Aerial Photo of EEW SPC

- Production space: 32,000 m²
- Storage incl. sec. steel & coating: 228,000 m²
- Length of quay: 194 m
- Draft: 9.5 m
Dimensions MP’s/TP’s

Monopile

Dimensions:
- Diameter: 1.5 m to 7 m
- Length: up to 120 m
- Weight: up to 1000 t

XL Monopile

Dimensions:
- Diameter: up to 10 m
- Length: up to 120 m
- Weight: up to 1500 t

Transition Piece (incl. XL TP’s)

Dimensions:
- Diameter: up to 6.5 m
- Length: up to 25 m
- Weight: up to 300 t
Offshore Wind Energy in the EU

Development installed offshore wind power capacity in MW

Cumulative offshore wind capacity in GW (2005-2020)

Historical data:
- Actual 2013: 5.2 GW
- CAGR +23.5%

Projection:
- Actual estimate 18.3 GW
- CAGR +36.6%
- Total actual 5.2 GW
- CAGR +23.4%

Installed capacity: share of installation in MW

- Actual 2013: 5.2 GW
- Estimated 2017: 18.3 GW
- Others
- 7281; 40%
- 4971; 27%
- 1193; 7%
- 976; 5%
- 823; 5%
- 1141; 6%
- 976; 5%
- 823; 5%
- 1921; 10%
- 921; 18%
- 380; 7%
- 247; 5%
- 2948; 57%

Source: EWEA, GWEC, EEW SPC
Foundation types share in Europe (wind park in operation/in progress)

- Monopile: 75.5%
- Jacket: 5.9%
- Tripod: 4.7%
- Tripile: 2.9%
- Gravity: 11.0%
- Floating: 0.1%

Source: WPO Intelligence Database, EEW SPC
Offshore wind parks (in operation and in progress) from 2010-2015

Foundation Manufacturers Market Share

Source: WPO Intelligence Database, EEW SPC
Offshore Wind Reference List

- Consortium/JV EEW/Bladt
- EEW SPC
- Bladt Industries

**USA:**
- Cape Wind: 59,000 to

**GB:**
- 408,540 to

**DK:**
- 79,000 to

**B:**
- 33,410 to

**D:**
- 524,060 to

**NL:**
- 75,000 to

**No. foundations:**
- MP’s: 1,486
- TP’s: 1,301
- Tripods: 80
- Tripiles: 40
- Jackets: 41
Monopiles remain the dominant foundation concept, but trend toward deeper water is shifting to jacket foundations.

<table>
<thead>
<tr>
<th>Foundation</th>
<th>Depth (m)</th>
<th>Cum 2012</th>
<th>Trend 2020</th>
<th>Comments</th>
</tr>
</thead>
</table>
| Gravity-based      | < 20      | 21%       |            | • Environmental restrictions  
| foundations        |           |           |            | • Complicated logistics  
|                    |           |           |            | • Suitable only for lower water depth                                    |
| Monopiles (incl. XL)| 10 - 40   | 75%       | 21%        | • XL pipes will pot. replace Jackets < 40 m depth  
|                    |           |           |            | • Known technology                                                       |
| Tripod/Tripile     | 25-50     | 2%        | 21%        | • High fabrication costs  
|                    |           |           |            | • Too heavy  
|                    |           |           |            | • Expensive logistics                                                    |
| Jacket             | 35-60     | 2%        | 21%        | • Stiffer structure/less steel  
|                    |           |           |            | • Higher installation efforts  
|                    |           |           |            | • Higher fabrication costs                                               |
| Floating           | > 50      | < 1%      |            | • Commercial realisation long term only  
|                    |           |           |            | • Logistical challenges                                                  |

Source: Roland Berger Presentation: Offshore Wind towards 2020
Development of MP’s

<table>
<thead>
<tr>
<th>Year</th>
<th>Project</th>
<th>Capacity (MW)</th>
<th>Water Depth</th>
</tr>
</thead>
<tbody>
<tr>
<td>2002</td>
<td>Horns Rev 1</td>
<td>2.0</td>
<td>up to 14 m</td>
</tr>
<tr>
<td>2008</td>
<td>Lynn</td>
<td>3.6</td>
<td>up to 18 m</td>
</tr>
<tr>
<td>2012</td>
<td>London Array</td>
<td>3.6</td>
<td>up to 25 m</td>
</tr>
<tr>
<td>2014</td>
<td>Baltic II</td>
<td>3.6</td>
<td>up to 27 m</td>
</tr>
<tr>
<td>2015</td>
<td>Gode Wind II</td>
<td>6.0</td>
<td>up to 35 m</td>
</tr>
<tr>
<td>2018</td>
<td>Future MP’s</td>
<td>8+</td>
<td>up to 40 m</td>
</tr>
</tbody>
</table>

**Dimensions**

- **L 34 m** Ø 4 m 160 t
- **L 45 m** Ø 4.7 m 350 t
- **L 68 m** Ø 5.7 m 650 t
- **L 73.5 m** Ø 6.5 m 930 t
- **L 80 m** Ø 8.5 m 1050 t
- **L >80 m** Ø >9 m >1050 t

Source: A2Sea News - Winter 2013 and EEW SPC
# Comparison MP‘s vs. Jackets

## Fabrication Costs

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<thead>
<tr>
<th></th>
<th>MP’s/TP’s</th>
<th>Jackets</th>
<th>XL MP’s/TP’s</th>
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</thead>
<tbody>
<tr>
<td>Turbine</td>
<td>Siemens 3.6 MW</td>
<td>Siemens 3.6 MW</td>
<td>Siemens 6 MW</td>
</tr>
<tr>
<td>Water depth</td>
<td>23 - 27 m</td>
<td>27 - 44 m</td>
<td>25 - 40 m</td>
</tr>
<tr>
<td>Diameter</td>
<td>5,5 - 6,5 m</td>
<td>3 m</td>
<td>7,5 - 9 m</td>
</tr>
<tr>
<td>Weight</td>
<td>630 - 930 to</td>
<td>1160 to</td>
<td>900 - 1500 to</td>
</tr>
<tr>
<td>Costs</td>
<td>100%</td>
<td>+34%</td>
<td>+8%</td>
</tr>
</tbody>
</table>
Thank you
<table>
<thead>
<tr>
<th>Author</th>
<th>Title</th>
<th>Session</th>
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<td>Khalid Abdel-Rahman</td>
<td>Numerical Approach for the Derivation of Interaction Diagrams for Piles under Cyclic Axial Loading</td>
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<tr>
<td>Martin Achmus</td>
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<td>Basics of a Design Concept for Bucket Foundations of Offshore Wind Turbines</td>
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<td>Martin Achmus</td>
<td>Influence of Soil Resistance Approach on Overall Structural Loading of Large Diameter Monopiles for Offshore Wind Turbines</td>
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<tr>
<td>F. Adam</td>
<td>The GICON-TLP for Wind Turbines - Wind and Wave Tank Test Results</td>
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<td>Masoud Asgarpour</td>
<td>Robust Installation Planning of Offshore Wind Farms</td>
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<td>José Azcona</td>
<td>Improved Tank Test Procedures for Scaled Floating Offshore Wind Turbines</td>
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<td>Anne Bechtel</td>
<td>Experimental Fatigue Tests on Axially Loaded Grouted Joints</td>
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<td>Freede Blaabjerg</td>
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<td>Tobias Bohne</td>
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<td>Stephan Brauser</td>
<td>Supply Chain Concept for Industrial Assembling of Offshore-Wind-Jackets</td>
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<td>C. Buchhagen</td>
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<td>M. Collmann</td>
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