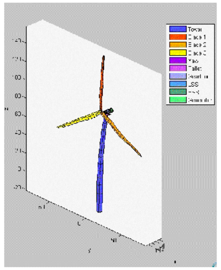
Aeroelastic Simulations of Wind Turbines



Tower design and Modal analysis

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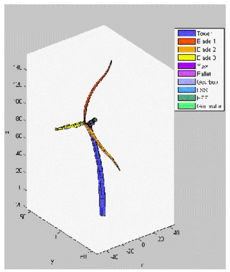
Prof. Dr. M. Kühn

ForWind – Wind Energy Systems

Topics

- Repetition – Dynamics of a wind turbine
- Tower design
- Modal analysis

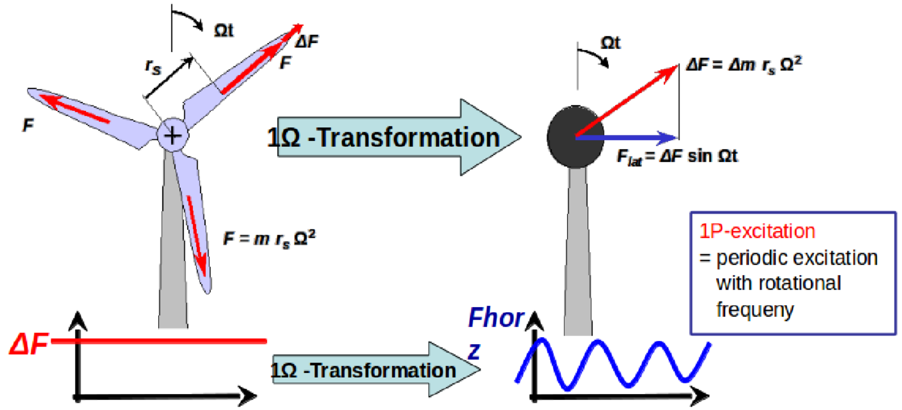


Section I: Repetition – Dynamics of a wind turbine

Load transformation from the rotating into the fixed coordination system: Unbalance

rotating (blade) system:
unbalance results in
stationary axial force in blade
ΔF = Δm rs Ω2

fixed (nacelle) system:
unbalance results in
rotating excitation, so called
1P-excitation
Fhorz = Δm rs Ω2 sin Ωt

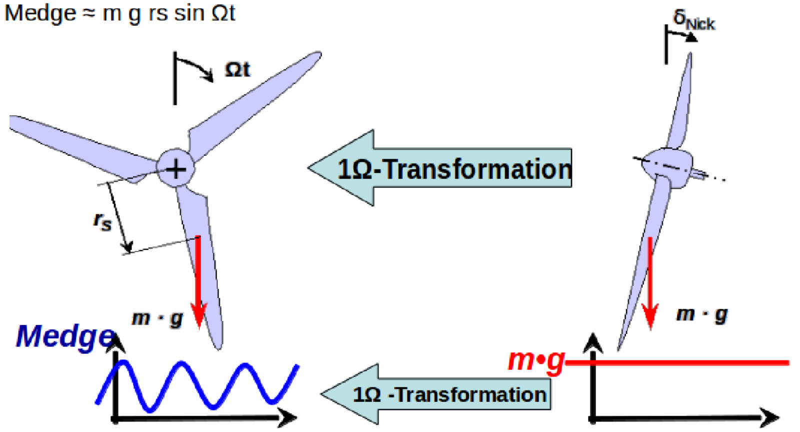


1P-excitation
= periodic excitation
with rotational
frequeny

Load transformation from the fixed into the rotating coordination system: Gravitational load

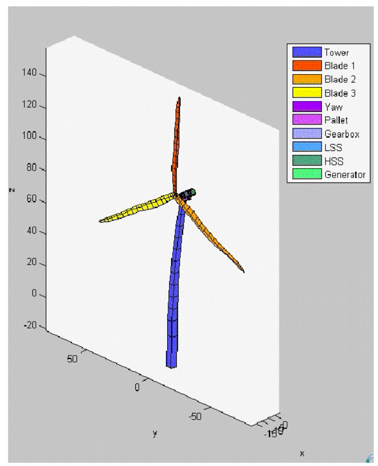
<u>rotating (blade) system</u>:
gravitational load causes *1P-excitation* with highly alternating edgewise bending moment
Medge ≈ m g rs sin Ωt

<u>fixed (nacelle) system</u>:
gravitational load is *stationary* and acting in a fixed direction
F = m g

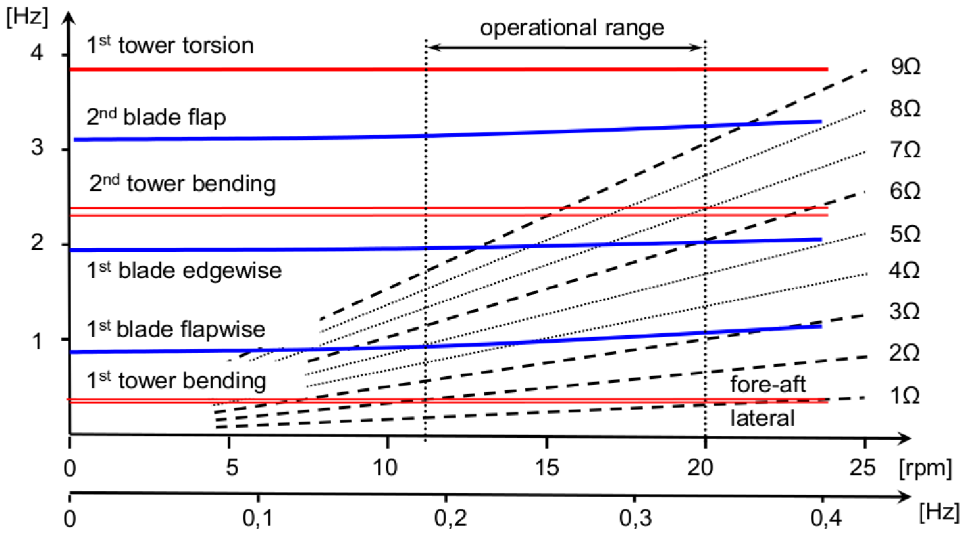


Eigenmodes coupled/uncoupled

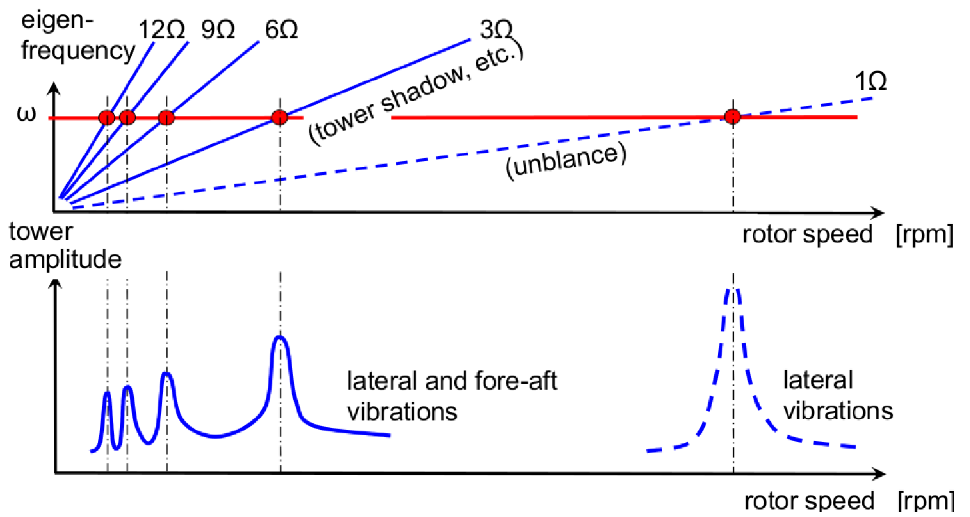
- Tower Side-to-side



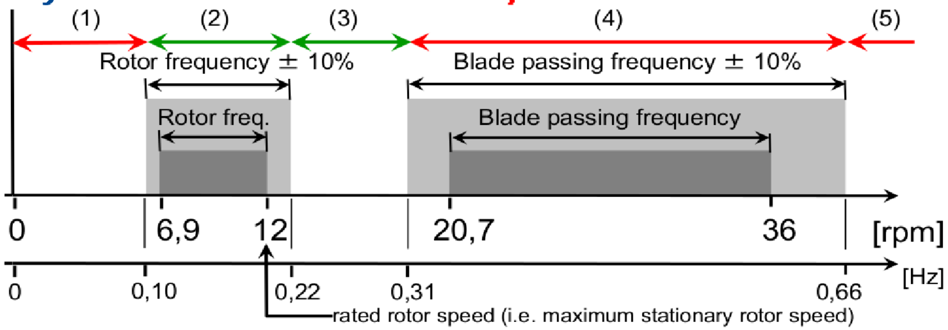
Campbell diagram with eigenfrequencies of the tower-nacelle-system and blade – drive train system



Tower resonances (3-bladed rotor) illustrated at Campbell diagram (upper fig.) and tower amplitude (lower fig.)



Variations of the dynamic design of the rotor nacelle system of a variable rotor speed turbine



Design ranges:

(1) Very soft, hardly possible due to strength requirements and excessive dynamic wave excitation (unless a compliant design with an eigenfrequency below the wave excitation is chosen)

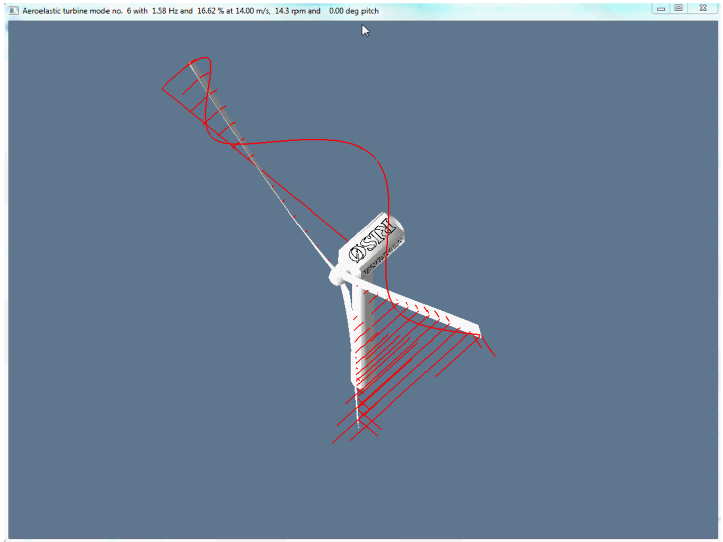
(2) Soft-soft design range in the resonance range of the rotor speed, requires an exclusion window for stationary operation of the rotor speed, soft-soft design with quite significant wave excitation

(3) Classical soft-stiff design range, already resulting in significant wave excitation

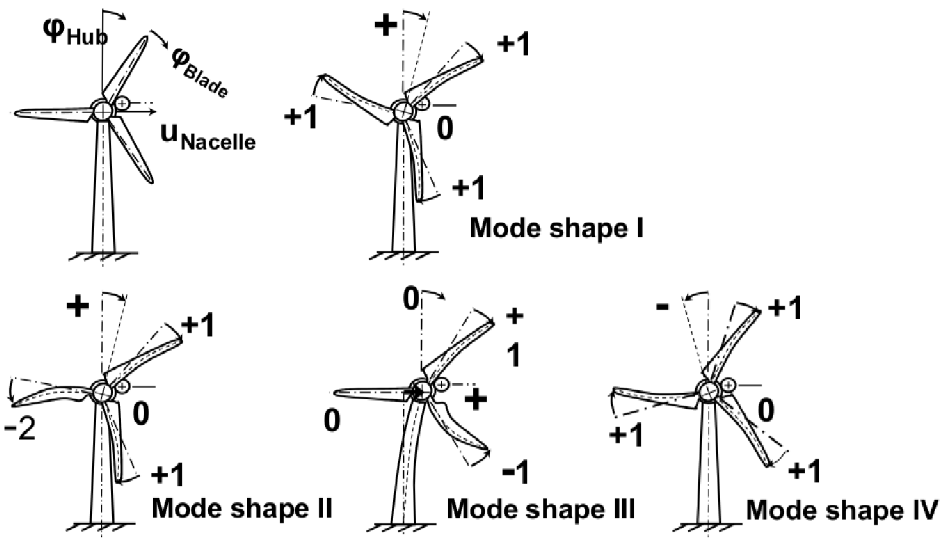
(4) Blade resonance range with excessive excitation from cyclic aerodynamic loading, design impossible without a large exclusion window of the rotor speed

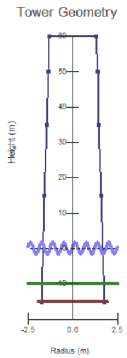
(5) Stiff-stiff, uneconomical design due to too high stiffness requirements

Hawc2



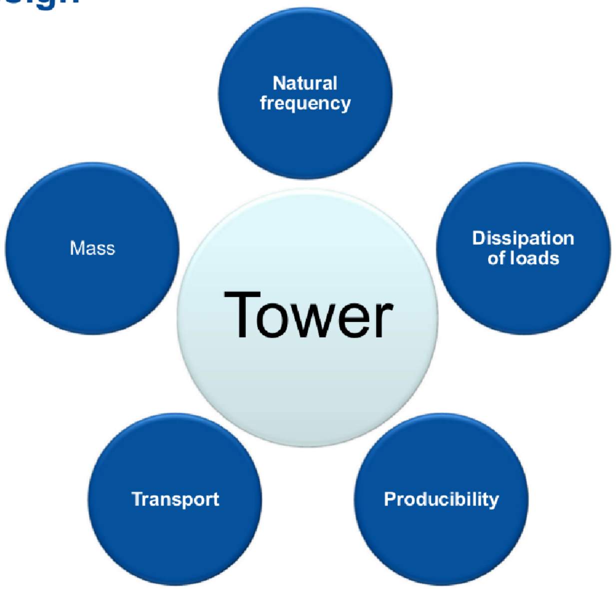
Mode shapes of the coupled blade – drive train – nacelle - tower system





Section II: Tower design

Tower design

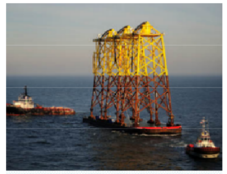


Tower design

- Transport:


[adamhenning@flickr]


[saturn@flickr]


[alpha ventus]

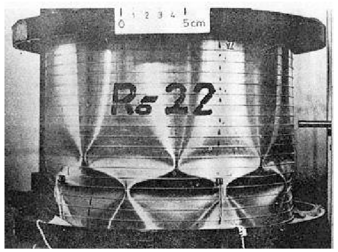

[windsordi@flickr]

- Producibility


[Bilfinger Berger]

Tower design

- Mass
  - Important in terms of economy
  - Transport
- Dissipation of loads
  - Buckling forces
  - Extreme Loads
  - Fatigue loads
- Natural frequency


[University of Western Australia]

Example for buckling

Remark

- Tower design very simplified
  - Cylindrical tower
  - Normally: Conical with more stations (hybrid)
  - No preliminary load calculation done
  - Buckling
  - Extreme loads
  - Fatigue loads
  - No material selection
  - No producibility check
  - No transportability check

Tower design

- Eigenfrequency
  - Influenced by modal mass and stiffness

$$\omega_0 = \sqrt{\frac{k}{m_{\text{modal}}}} = f_0 \cdot 2\pi$$

  - Modal mass is approximated by swinging part of the tower + tower head mass

$$m_{\text{modal}} = m_{\text{towertop}} + 0.25 \cdot m_{\text{towermass}}$$

  - Stiffness and mass are depending on the wall thickness

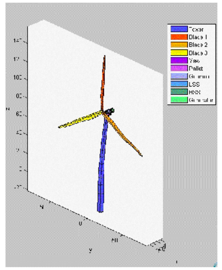
$k = \frac{3EI}{l^3}$
where
k = Stiffness
E = E – modulus
I = Moment of inertia

$I = \frac{\pi D^3 t}{8}$
where
I = Moment of inertia
D = Tower diameter
t = Wall thickness

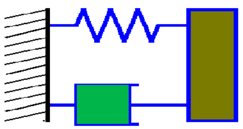
$m_{\text{tower}} = \rho \cdot \pi \cdot D \cdot t \cdot l$
where
ρ = Material density
D = Tower diameter
t = Wall thickness
l = Tower height

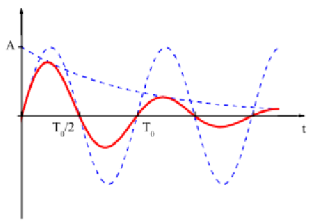
- Eigenfrequency = Maximum rotor speed + 10%

Section III: Modal analysis

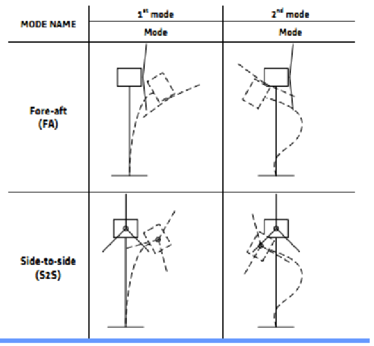


Excurse: Modal analysis (ii)





$$\omega = \omega_0 \cdot \sqrt{1 - \delta}$$



Campbell diagram of a typical 1.5 MW turbine