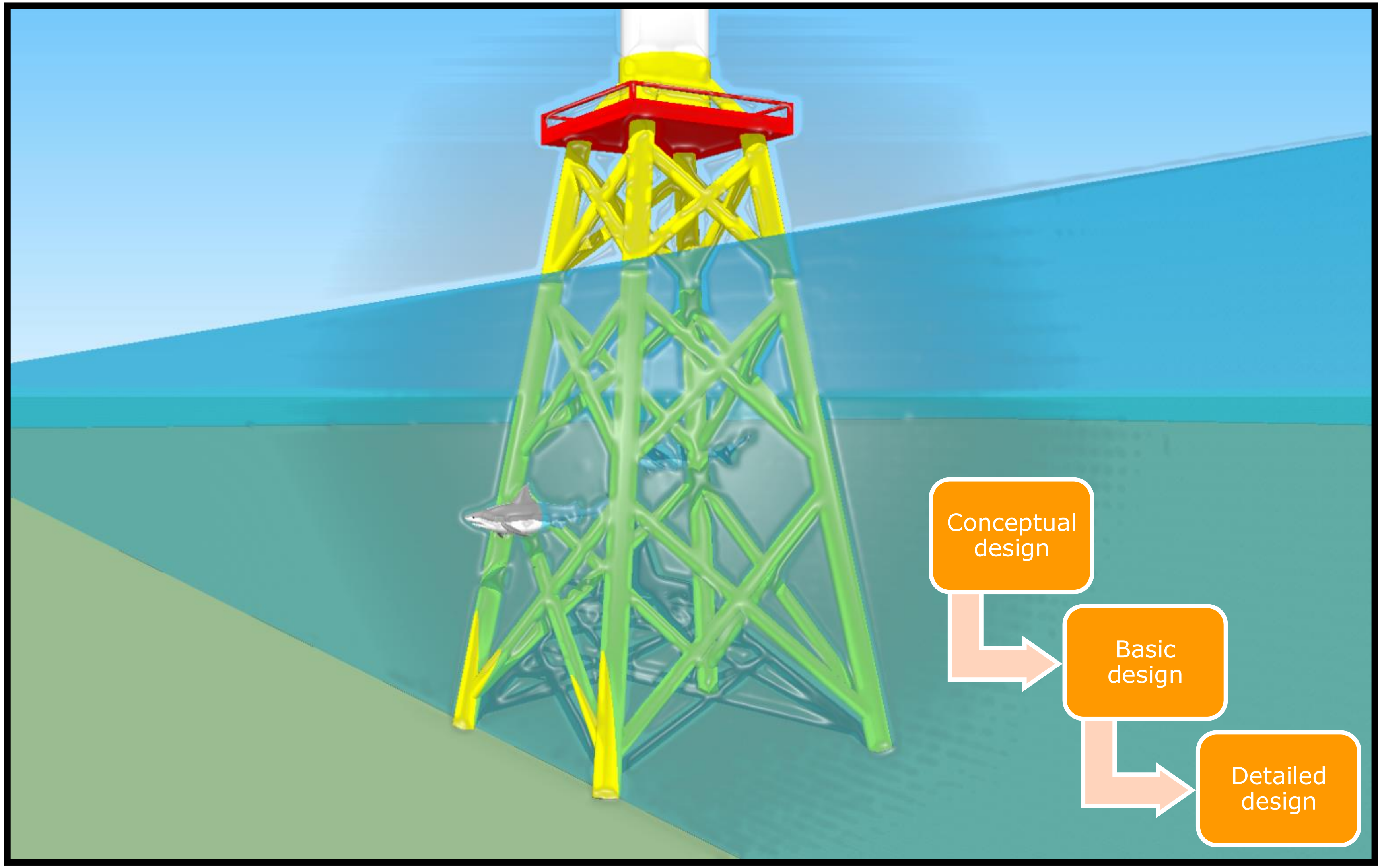
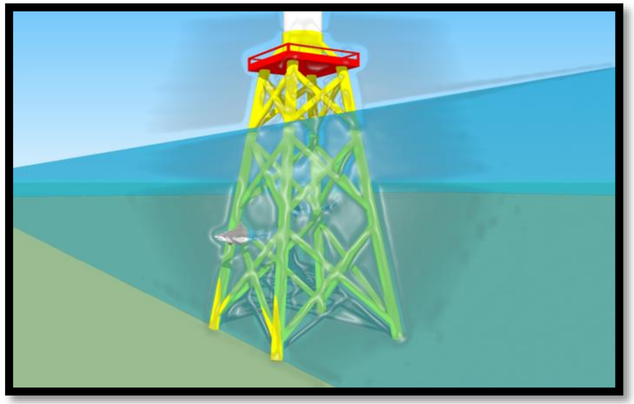
Investigating Optimal Leg Distance, using Conceptual Design Optimization

Kasper Sandal
Alexander Verbart
Mathias Stolpe
Technical University of Denmark,
Dept. of Wind Energy

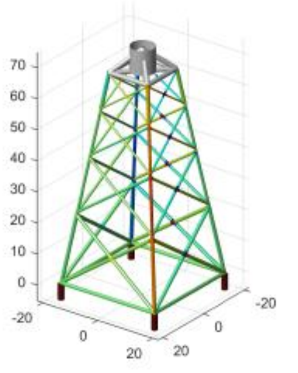


This talk presents conceptual design optimization of jacket structures for offshore wind turbines

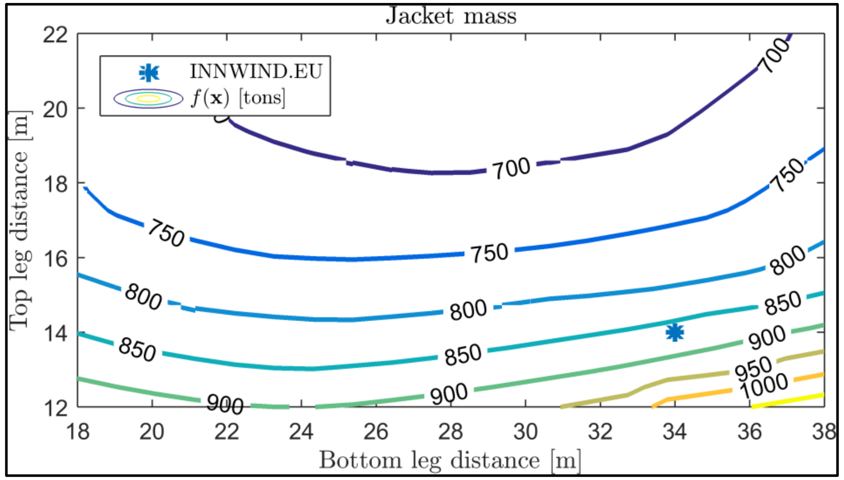


Design considerations

$$
\begin{aligned}
\text{minimize} \quad & f(\boldsymbol{x}) \\
\text{subject to} \quad & \mathbf{K}(\boldsymbol{x})\boldsymbol{u} - \Delta\boldsymbol{P} = \mathbf{0} \\
& \underline{\boldsymbol{\sigma}} \le \boldsymbol{\sigma}(\boldsymbol{x}) \le \overline{\boldsymbol{\sigma}} \\
& \underline{\boldsymbol{\omega}} \le \boldsymbol{\omega}(\boldsymbol{x}) \le \overline{\boldsymbol{\omega}}
\end{aligned}
$$



Optimal design problem



Design trends

A good jacket design has low mass to minimize material, transportation, and installation costs

To avoid resonance, the natural frequency must lie in the soft-stiff range between the 1p and 3p rotor frequencies

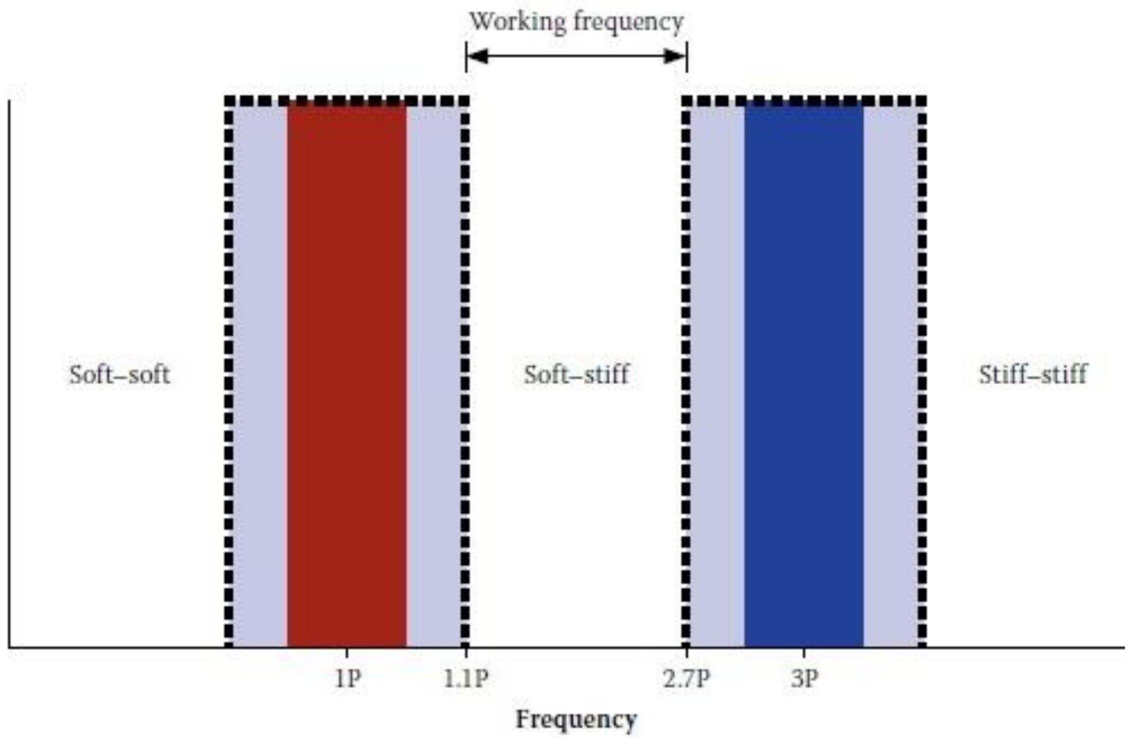
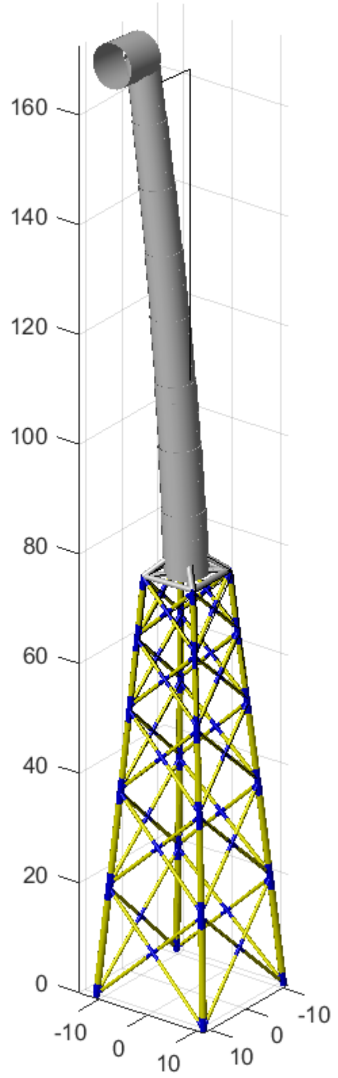


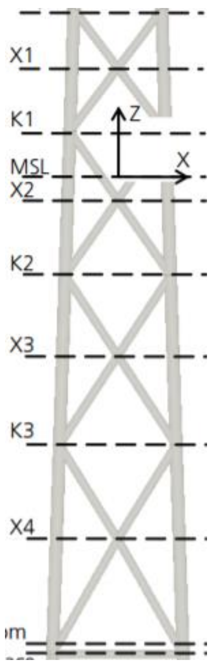
Photo: W S van Zyl; G P A G van Zijl



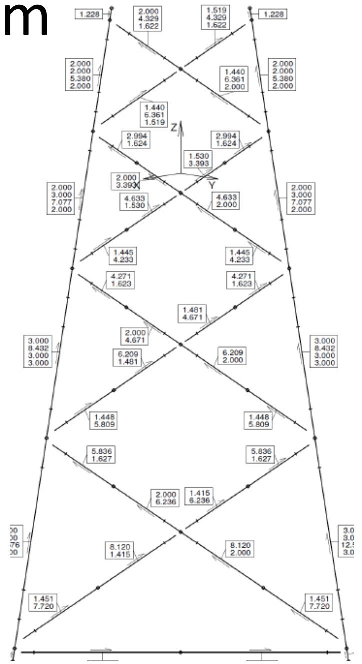
Reference jackets in the literature have very different leg distances

OC4 jacket

Designed for
NREL 5 MW



INNWIND.EU jacket
Designed for
DTU 10 MW

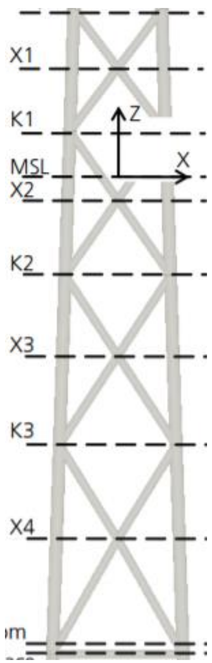


Placing the same tower and turbine on two jackets allows us to compare them

OC4 jacket

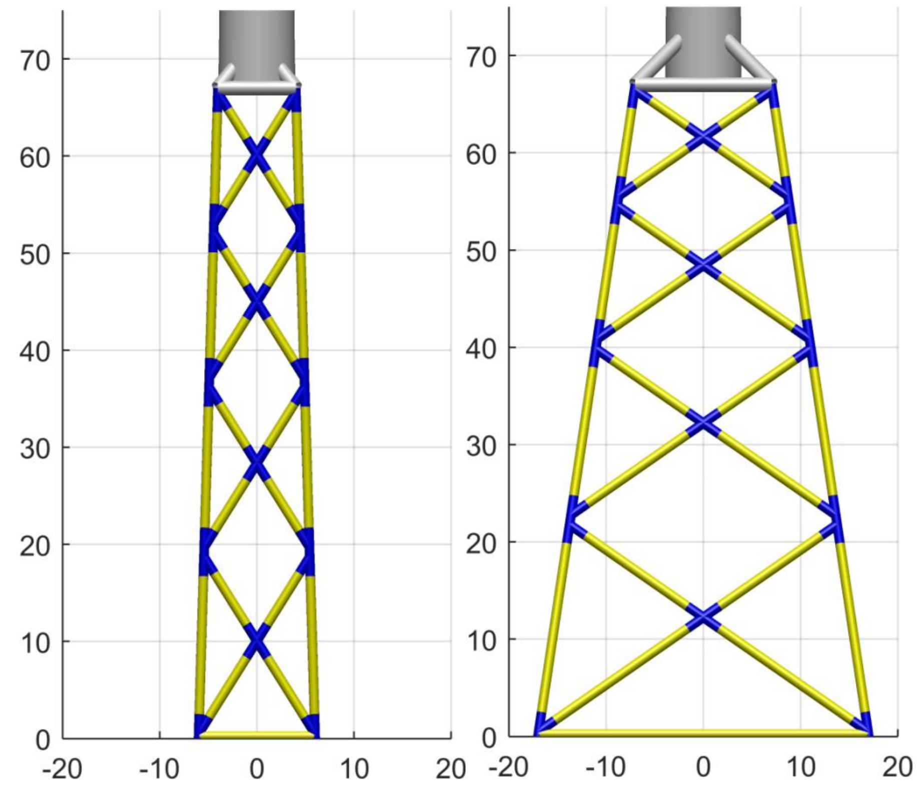
Designed for NREL 5 MW

8 m



12 m

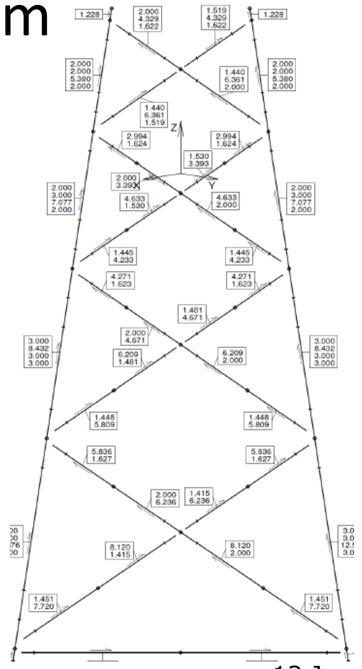
JADOP models with DTU 10 MW tower & turbine



INNWIND.EU jacket

Designed for DTU 10 MW

14 m

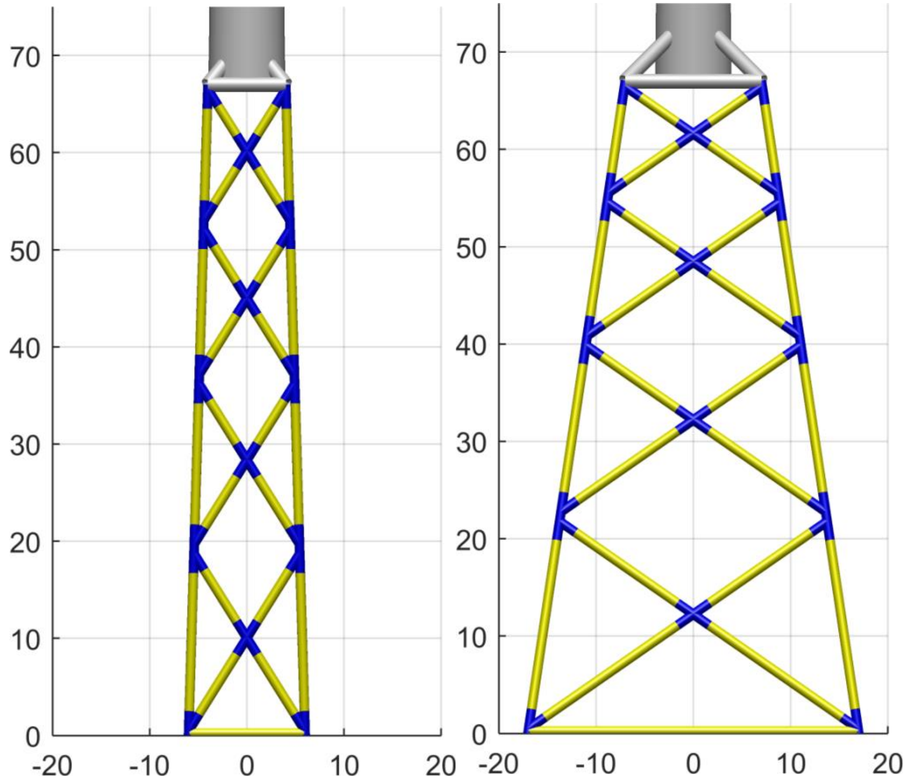


34 m

When cross sections are equal, a slender jacket will have a lower mass and a lower frequency than a bulky jacket

JADOP models with DTU
10 MW tower & turbine
Diameters = 1 m
Thicknesses = 50 mm

Slender
1060 tons
0.20 Hz

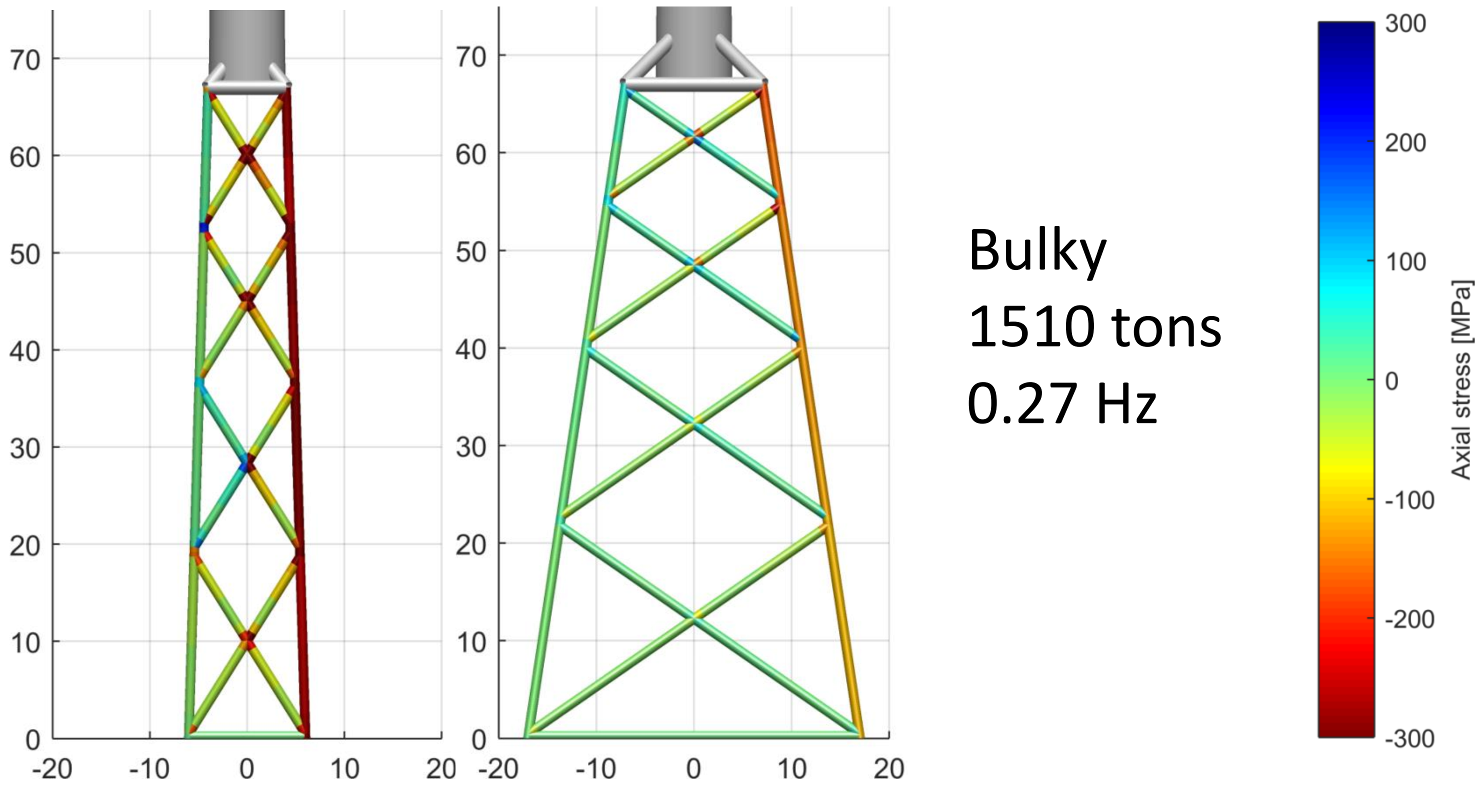


Bulky
1510 tons
0.27 Hz

To satisfy the fatigue and ultimate limit states, the cross sections have to change when the slenderness change

JADOP models with DTU
10 MW tower & turbine
Diameters = 1 m
Thicknesses = 50 mm

Slender
1060 tons
0.20 Hz



Bulky
1510 tons
0.27 Hz

The optimization problem for conceptual design is formulated with static loads, and constraints on stress, buckling, and frequency

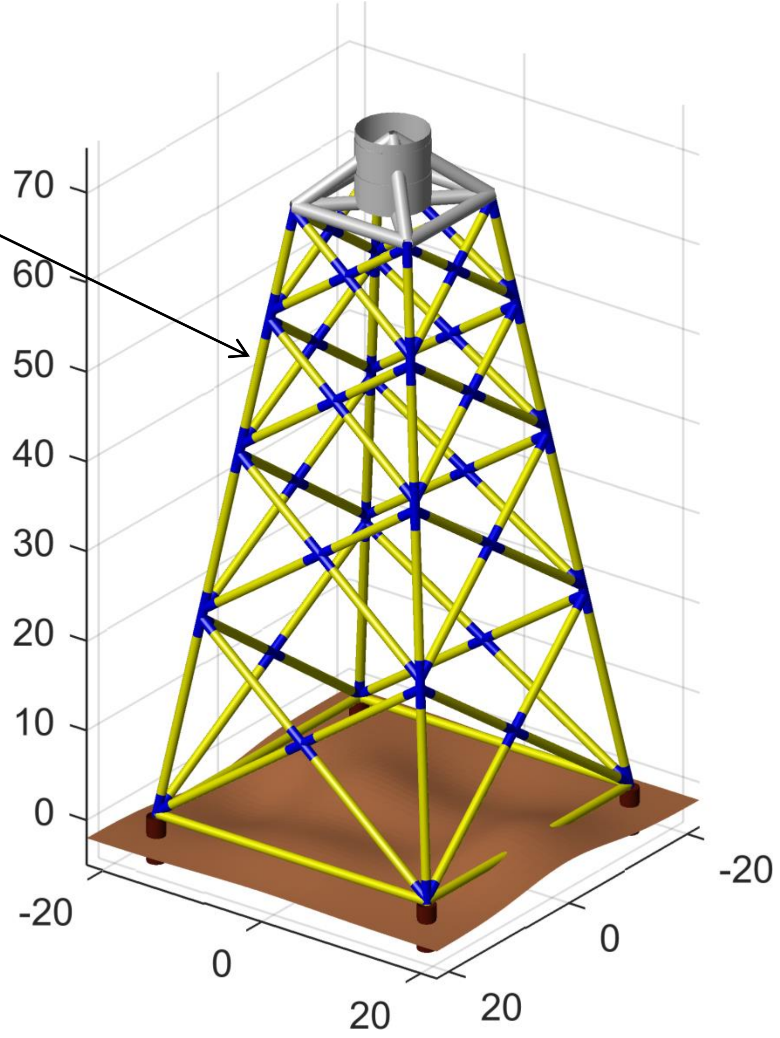
$\boldsymbol{x}$ = cross sections

$$
\begin{aligned}
\text{minimize} \quad & f(\boldsymbol{x}) \\
\text{subject to} \quad & \mathbf{K}(\boldsymbol{x})\boldsymbol{u} - \boldsymbol{f}(\boldsymbol{x}) = \mathbf{0} \\
& \underline{\boldsymbol{\sigma}} \leq \boldsymbol{\sigma}(\boldsymbol{x}) \leq \overline{\boldsymbol{\sigma}} \\
& \underline{\boldsymbol{\omega}} \leq \boldsymbol{\omega}(\boldsymbol{x}) \leq \overline{\boldsymbol{\omega}}
\end{aligned}
$$

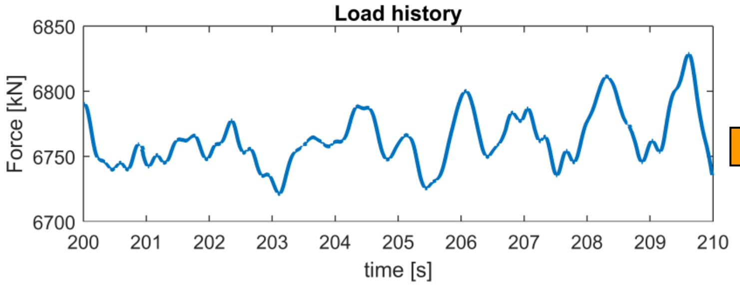
Jacket mass

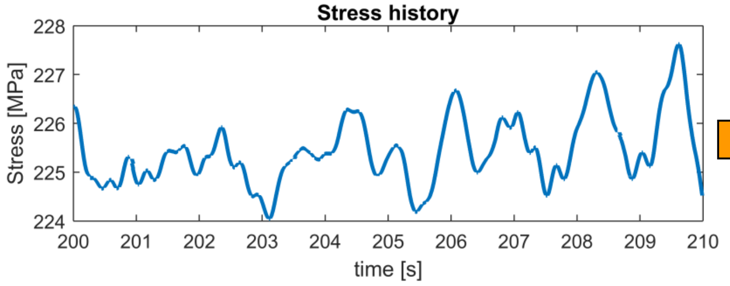
Stress

Frequency



Damage equivalent loads are used to make an approximate fatigue constraint using static stress constraints

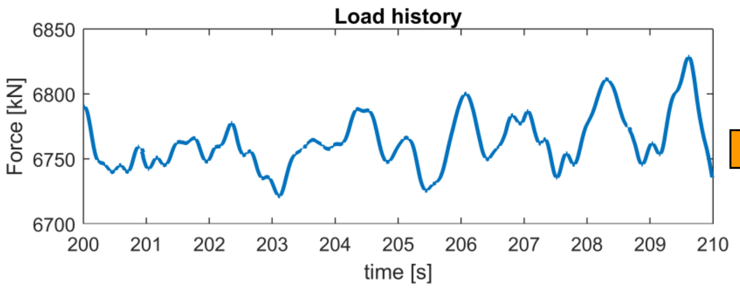


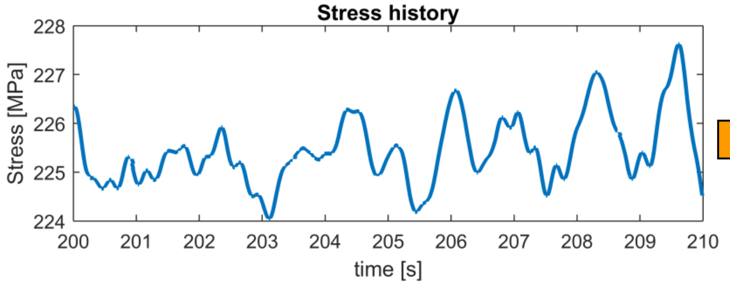


Rainflow counting: $\Delta\sigma_i, n_i$

$$D = \sum_i \frac{n_i(\Delta\sigma_i)^m}{\bar{a}} \leq D^{max}$$

Damage equivalent loads are used to make an approximate fatigue constraint using static stress constraints





Rainflow counting: $\Delta\sigma_i, n_i$

$$D = \sum_i \frac{n_i(\Delta\sigma_i)^m}{\bar{a}} \leq D^{max}$$

Rainflow counting: $\Delta P_i, n_i$

$$n_T(\Delta P^{1Hz})^m = \sum_i n_i(\Delta P_i)^m$$

$$\Rightarrow \Delta P^{1Hz} = \left(\frac{1}{n_T}\sum_i n_i(\Delta P_i)^m\right)^{\frac{1}{m}}$$

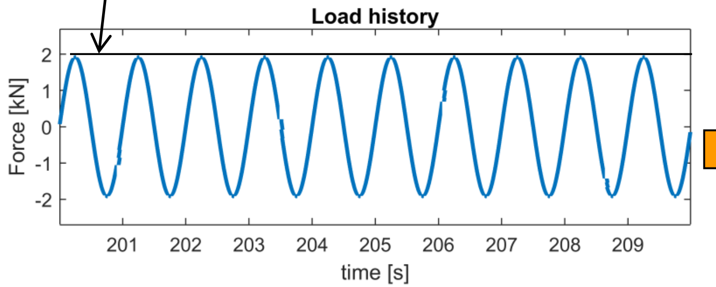
Equal fatigue damage

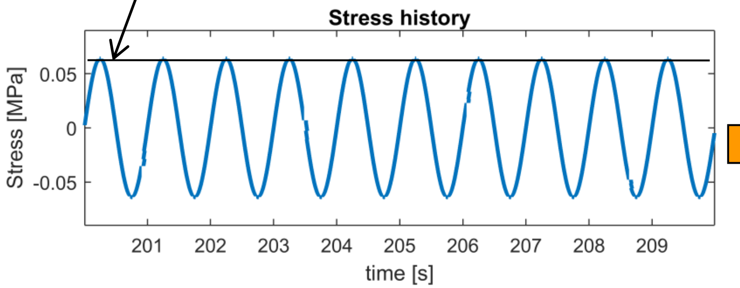
Quasi-static behaviour

1 degree of freedom load

High-cycle SN-curve

$\Delta\sigma^{1Hz}$





$$D = \frac{n_T\left(\Delta\sigma^{1Hz}\right)^m}{\bar{a}} \leq D_{max}$$

$$\Delta\sigma^{1Hz} \leq \overline{\Delta\sigma} = \left(\frac{D_{max}\bar{a}}{T^{life}}\right)^{\frac{1}{m}}$$

The problem is solved using the JAcket Design OPtimization tool JADOP and the open source optimization solver IPOPT

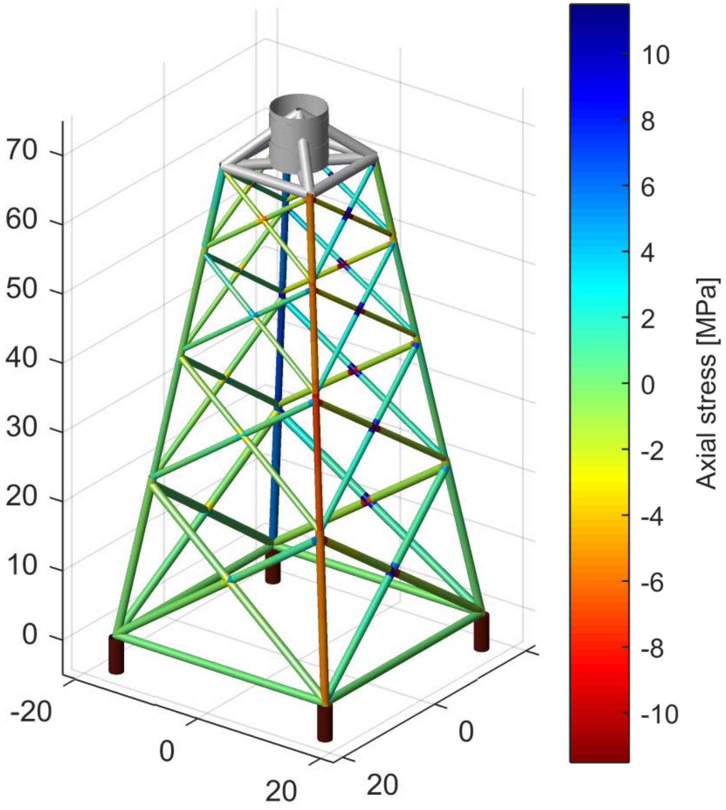
```
% Specify settings
S                                = settings;

% Specify DTU 10 MW turbine and INNWIND.EU jacket without piles
S.Geometry.Piles     = 0;     % 1 for piles, 0 for clamped.
S.Geometry.LegdistB = 24;    % Leg distance at seabed
S.Geometry.LegdistT = 14;    % Leg distance at transition piece
S.Geometry.Sections = 4;     % Number of sections
S.Geometry.Height    = 67;    % Jacket height (bottom of transition piece)
S.Geometry.MSL_h     = 50;    % Mean sea level
S.Geometry.Turbine   = 1;     % 1-DTU10MW, 2-NREL5MW

% Optimization settings
S.Optimization.flag              = 1;
S.Optimization.sand_flag         = 0;
S.Optimization.maxIter           = 500;
S.Optimization.constraints       = {'SCF-validity';'Stress ULS';'Buckling';
                                     'Equivalent fatigue';'Frequency'};
S.Optimization.variable_linking = '4nv';
S.Optimization.scale_obj         = 1;
S.Optimization.con_lim.disp      = 10;
S.Optimization.scale_stress      = 1e-6;
S.Optimization.con_lim.stress    = 350e6;
S.Optimization.con_lim.FLSstress= 11.5e6;
```

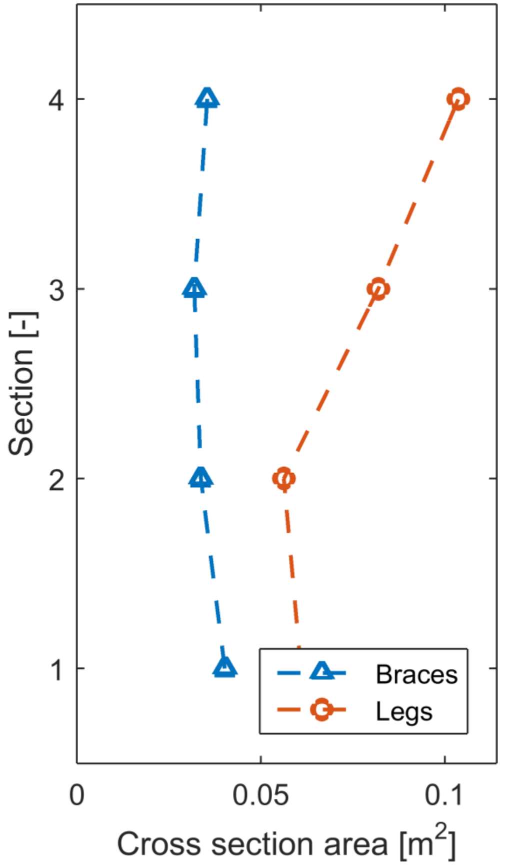
DTU 10 MW: Tower, turbine, and loads

- Parametric input
- Analytic sensitivities
- Many types of constraints

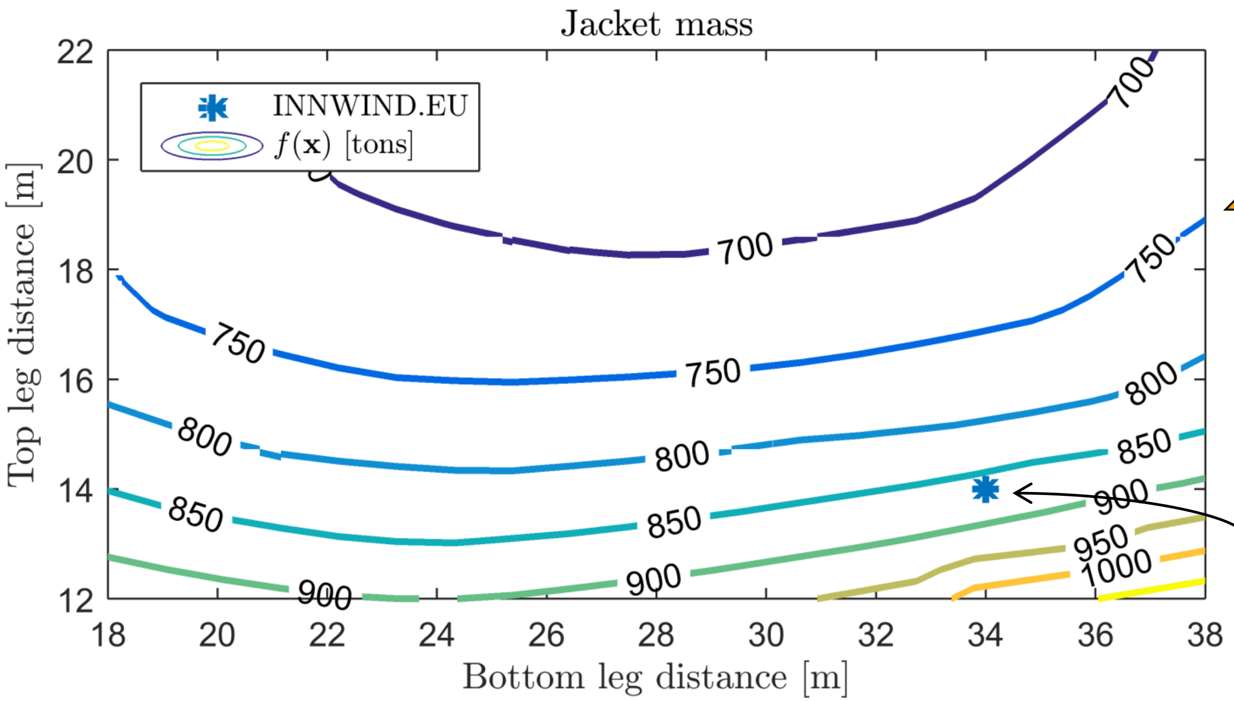
With leg distances from the INNWIND.EU jacket, the mass was minimized to 870 tons in 2 minutes on a laptop

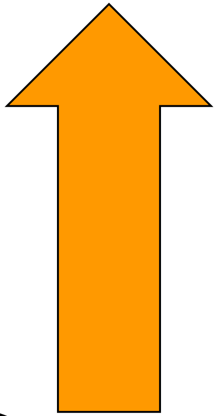
Top leg distance	= 14 m
Bottom leg distance	= 34 m
Jacket mass	= 870 tons
Natural frequency	= 0,264 Hz
Computation time	= 118 s





Optimization of 400 jackets indicate that an increased top leg distance reduces the jacket mass with about 20 percent



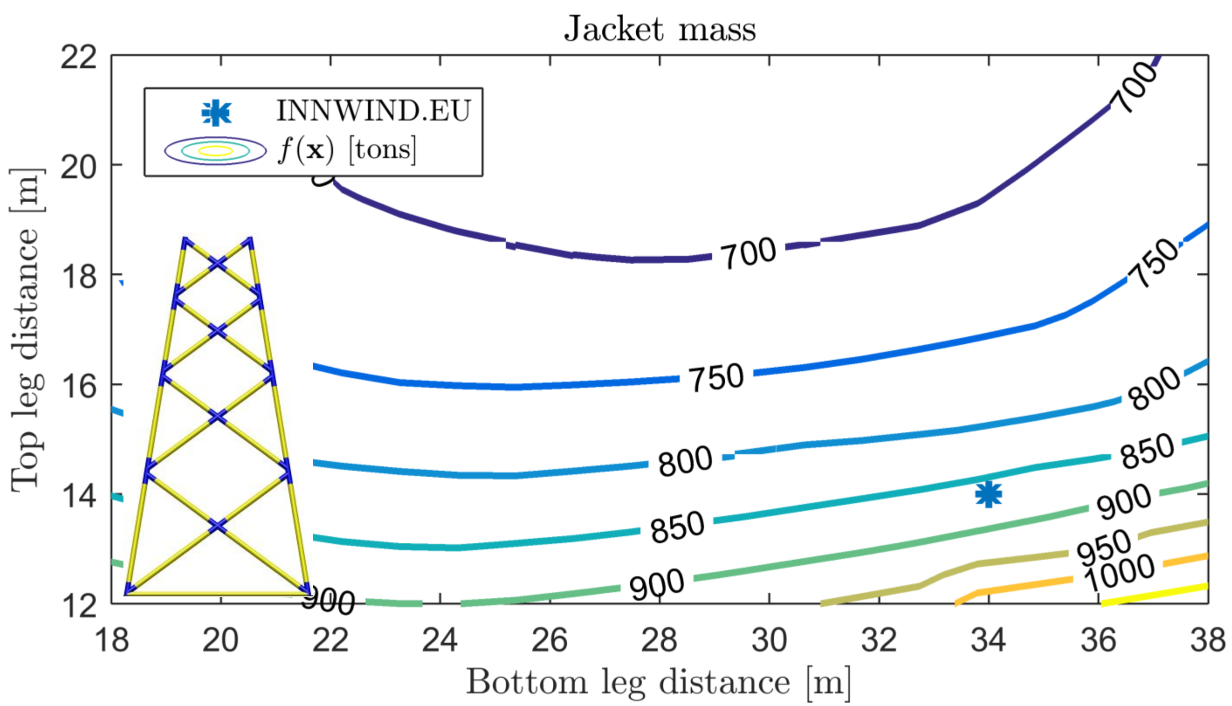


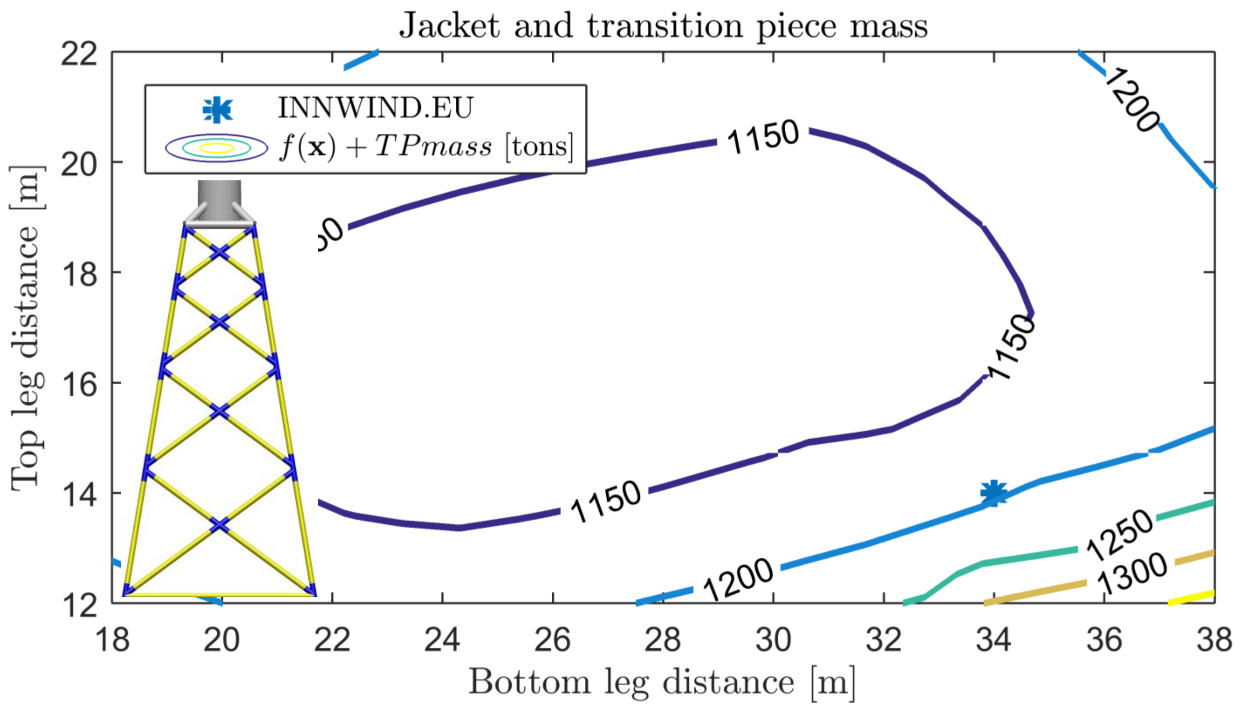
INNWIND.EU leg distances

Top leg distance = 14 m

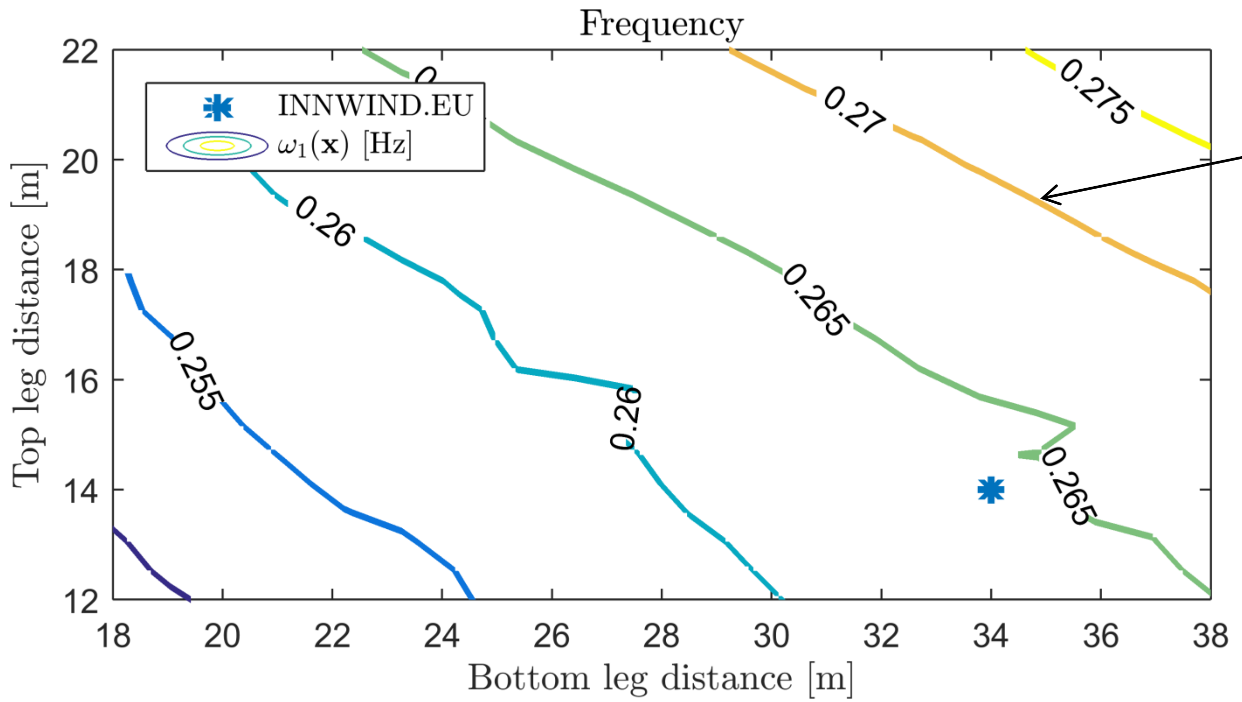
Bottom leg distance = 34 m

Since transition piece mass increases with larger top leg distance, the overall mass reduction is much less



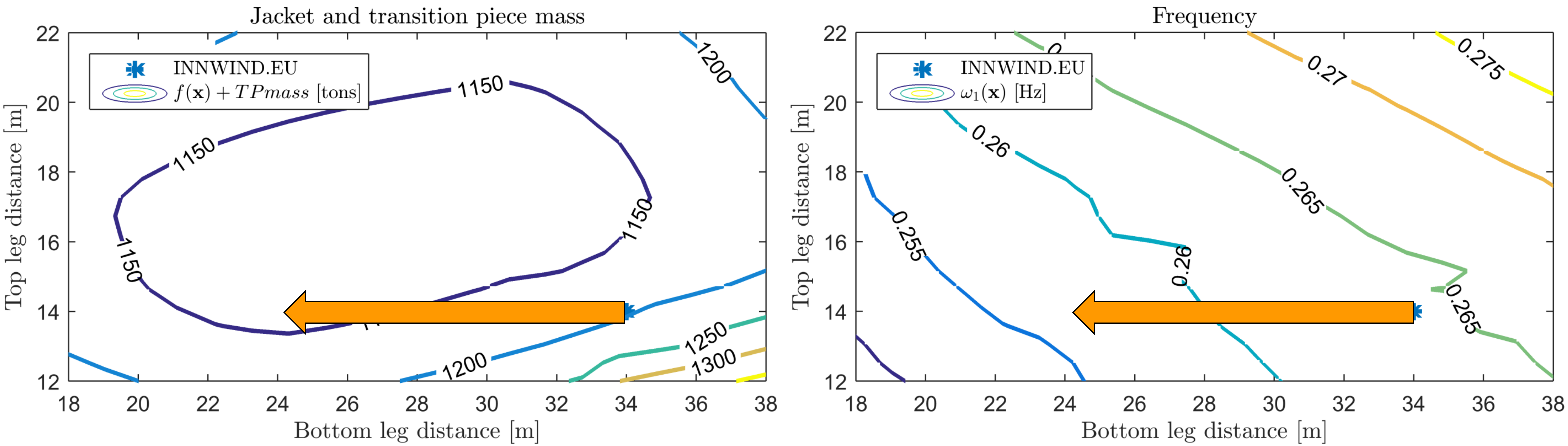


High leg distances at both bottom and top increase the natural frequency

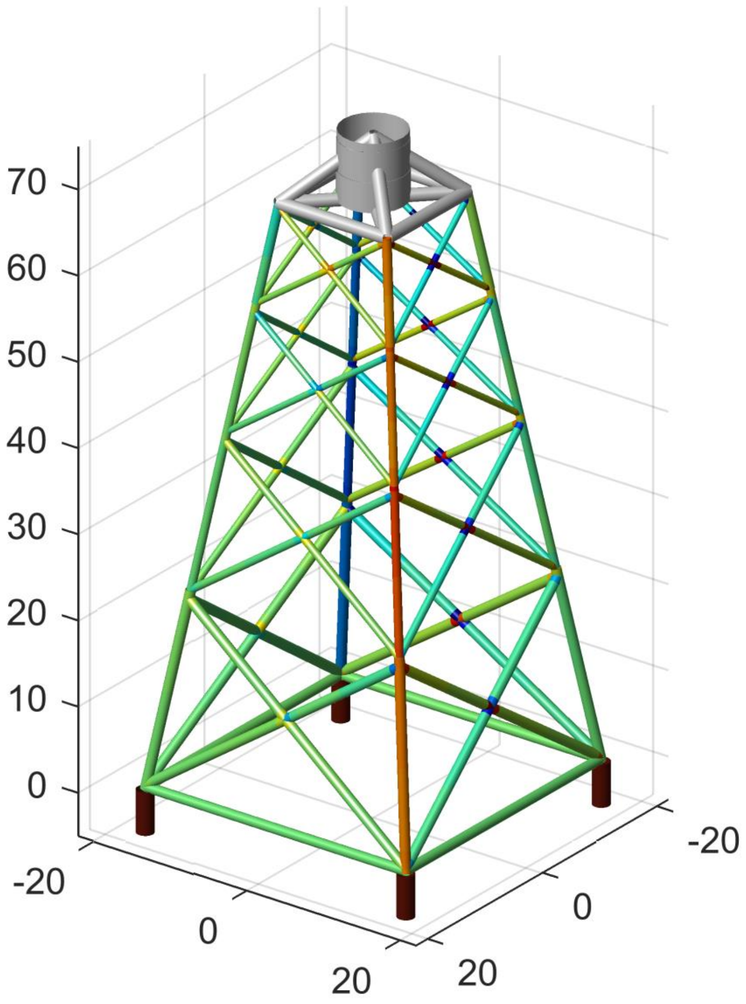


Upper bound on soft-stiff range

Reducing the bottom leg distance of the INNWIND.EU jacket from 34 to 24 meters, reduces both overall mass and frequency



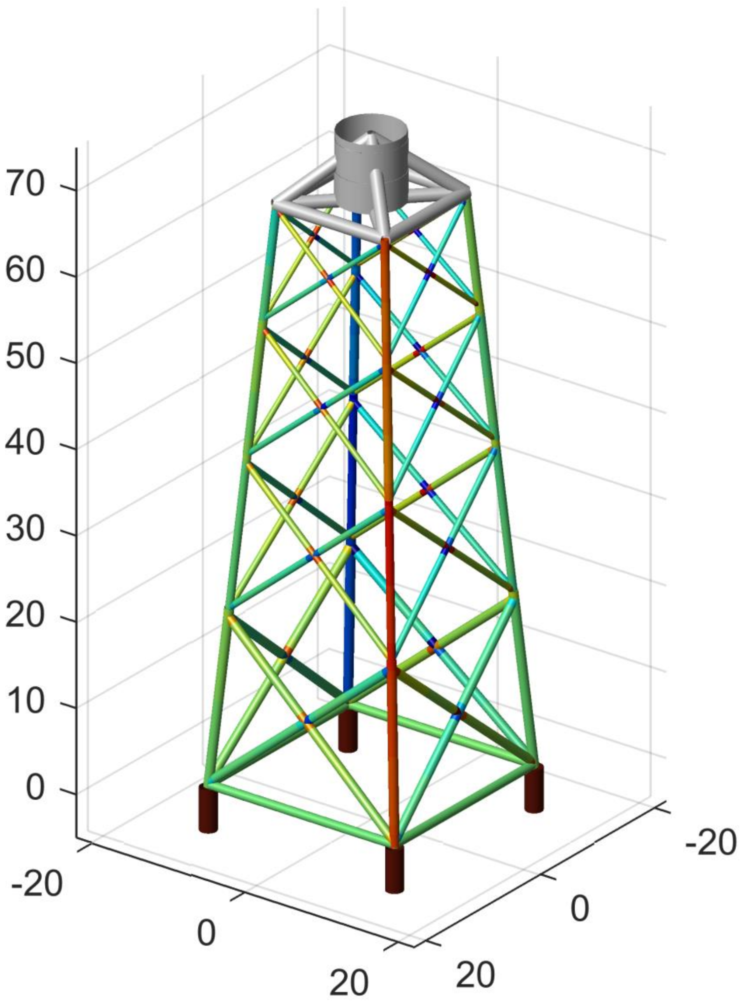
In conclusion, the conceptual design optimization is a fast and useful tool for investigating key parameters such as leg distance



Bottom leg distance: 34 m → 24 m

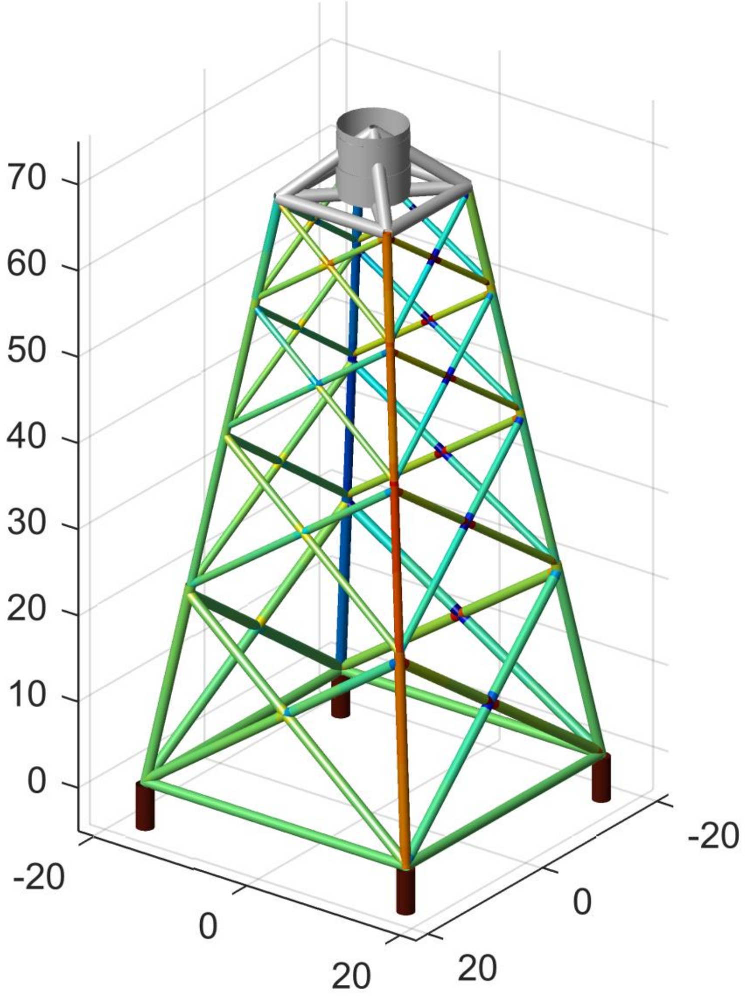
Mass: -6.7 percent

Frequency: -2.8 percent

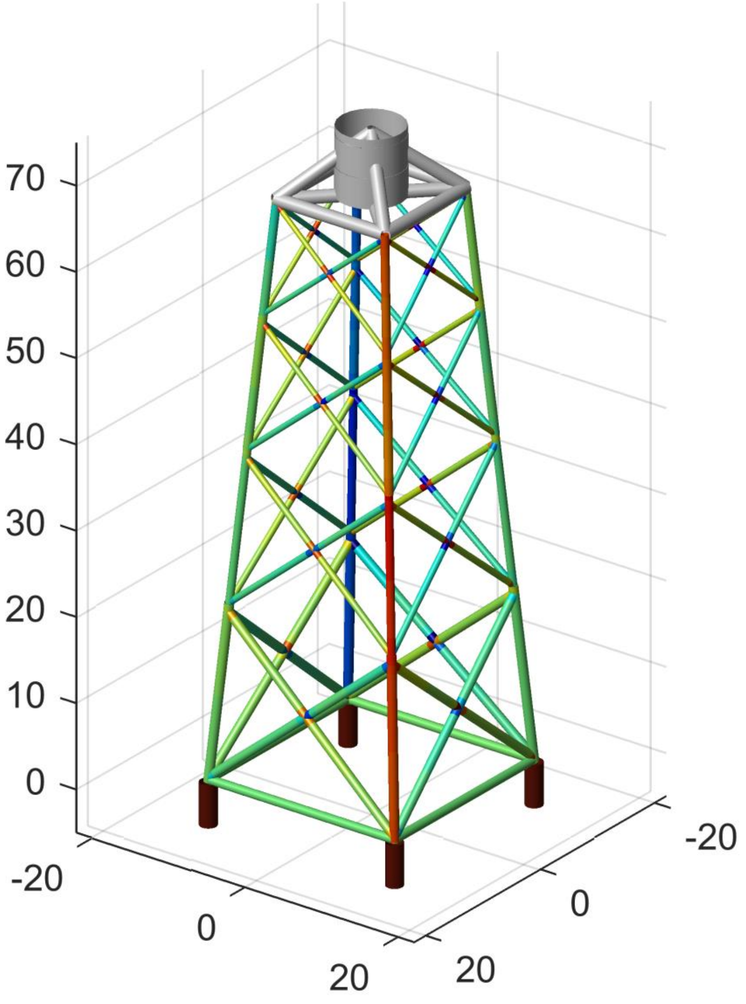


Can also be used for pile stick-up, number of sections, height, etc.

In conclusion, the conceptual design optimization is a fast and useful tool for investigating key parameters such as leg distance



Bottom leg distance:	34 m → 24 m
Mass:	-6.7 percent
Frequency:	-2.8 percent



Can also be used for pile stick-up, number of sections, height, etc.

Questions?

EXTRA SLIDES

Design according to DNVGL offshore standard and recommended practices

DNVGL-OS-C101 Design of offshore steel structures

DNVGL-RP-C203 Fatigue design of offshore steel structures

DNV-RP-C202 Buckling strength of shells

Optimal design problem

$$
\begin{aligned}
\underset{\mathbf{v}\in\mathbf{R}^{n_v},\mathbf{u}\in\mathbf{R}^{dn_l}}{\text{minimize}} \quad & f(\mathbf{v}) = \rho \sum_{e=1}^{n} A_e(d_e,t_e)l_e \\
\text{subject to} \quad & \mathbf{A}\mathbf{v} \ \leq \mathbf{b} \\
& \mathbf{K}(\mathbf{v})\mathbf{u}^l - \mathbf{f}^l(\mathbf{v}) = \mathbf{0}, && l = 1,...,n_l \\
& \underline{\sigma} \leq \sigma_{ehl}^{scf}(\mathbf{v},\mathbf{u}^l,\gamma_h) \leq \overline{\sigma}, && e = 1,...,n, h = 1,...,n_h, l = 1,...,n_{FLS} \\
& \sigma^b(\mathbf{v}) - \sigma_{ehl}(\mathbf{v},\mathbf{u}^l,\gamma_h) \leq 0, && e = 1,...,n, h = 1,...,n_h, l = n_{FLS}+1,...,n_l \\
& \underline{\omega_i} \leq \omega_i(\mathbf{v}) \leq \overline{\omega_i}, && i = 1,...,n_f \\
& g_e(\mathbf{v}) \leq 0, && e = 1,...,n \\
& \underline{\mathbf{v}} \leq \mathbf{v} \leq \overline{\mathbf{v}},
\end{aligned}
\tag{16}
$$

Load cases

Table 3: Description of static load cases

	Load type	Limit state	Rotation [deg]	Tower top load
1	Thrust	Fatigue	0	$F_x + M_y + \frac{1}{2}M_z$ from Δp^{1Hz}
2	Thrust	Fatigue	45	$F_x + M_y + \frac{1}{2}M_z$ from Δp^{1Hz}
3	Torsion	Fatigue	0	$\frac{1}{2}F_x + \frac{1}{2}M_y + M_z$ from Δp^{1Hz}
4	Torsion	Fatigue	45	$\frac{1}{2}F_x + \frac{1}{2}M_y + M_z$ from Δp^{1Hz}
5	Thrust	Ultimate	0	$F_x^{max} + M_y^{max}$ from [5]
6	Thrust	Ultimate	45	$F_x^{max} + M_y^{max}$ from [5]
7	Torsion	Ultimate	0	M_z^{max} from [5]

Shell buckling

$$\sigma^b(\mathbf{v}) - \sigma_{ehl}(\mathbf{v}, \mathbf{u}^l, \gamma_h) \leq 0, \tag{31}$$

where the shell buckling capacity in compression $\sigma^b(\mathbf{v})$, is defined as

$$\sigma^b(\mathbf{v}) = \frac{-\sigma^y}{\gamma_M \sqrt{1 + \left(\frac{\sigma^y}{f_{Em}}\right)^2}}, \qquad f_{Em} = C \frac{\pi^2 E}{12(1-\nu^2)} \left(\frac{t_e}{L_e}\right)^2, \qquad C = \sqrt{1 + (\rho\xi)^2} \tag{32}$$

$$\rho = \frac{1}{2\sqrt{1 + \frac{d_e}{600 t_e}}}, \qquad \xi = 1.404 \frac{L_e^2}{d_e t_e} \sqrt{1 - \nu^2}, \tag{33}$$

Column buckling

Column buckling need only be assessed for element e if

$$\frac{(kL_e)^2 A_e}{I_e} \geq \frac{2.5E}{\sigma^y}. \tag{34}$$

where $k = 0.7$ is the effective column length. To avoid assessing column buckling, the inverse of equation (34) can be formulated as a non-linear constraint $g_e(\mathbf{v}) \leq 0$, where

$$g_e(\mathbf{v}) = \sqrt{\frac{3.2\sigma^y}{E}} kL_e - d_e^2 + 2d_e t_e - 2t_e^2. \tag{35}$$

SCF validity constraints

The linear constraints $\mathbf{Ax} \leq \mathbf{b}$ enforce the SCF validity range [2], which states that for a joint where a brace is welded onto a leg, the dimensions should satisfy the following relations:

$$0.2d_{Leg} - d_{Brace} \leq 0 \tag{17}$$

$$d_{Brace} - d_{Leg} \leq 0 \tag{18}$$

$$0.2t_{Leg} - t_{Brace} \leq 0 \tag{19}$$

$$t_{Brace} - t_{Leg} \leq 0, \tag{20}$$

and that for all elements, the following should hold

$$16t - d \leq 0 \tag{21}$$

$$d - 64t \leq 0. \tag{22}$$

Stress & SCF

In the analysis of the offshore wind turbine structure, we assume that only normal stress $\sigma(\mathbf{v}, \mathbf{u}, \xi, \eta, \zeta) \in \mathbb{R}$ is significant. The normal stress in element e, position h, is computed as

$$\sigma_{eh}(\mathbf{v}, \mathbf{u}_e^g, \gamma_h) = E\mathbf{b}(\mathbf{v}, \gamma_h)\mathbf{T}_e\mathbf{u}_e^g, \tag{12}$$

where $\mathbf{b}(\mathbf{v}, \gamma_h) \in \mathbb{R}^{1\times 12}$ is the strain displacement vector for normal stress at postition h, and E is the materials Youngs modulus.

To account for stress concentrations in welded tubular joints, the recommended practice [2] provides a method using stress concentration factors (SCFs). This method assumes superposition of the normal stress components coming from axial forces (ax), moments in plane (mi) and moments out of plane (mo). We decompose the normal stress $\sigma_{eh}(\mathbf{v}, \mathbf{u}_e^g, \gamma_h)$ by decomposing the strain displacement vector:

$$\mathbf{b}(\mathbf{v}, \gamma_h) = \mathbf{b}^{ax}(\mathbf{v}, \gamma_h) + \mathbf{b}^{mi}(\mathbf{v}, \gamma_h) + \mathbf{b}^{mo}(\mathbf{v}, \gamma_h) \tag{13}$$

The recommended practice then provides coefficients that are to be multiplied onto each stress component. These coefficients are functions of diameter and thickness of all elements in the joint, as well as joint geometry, and the position h along the element circumference. The number of hot spots n_h in each element should be at least eight. The scf-stress $\sigma_{eh}^{scf}(\mathbf{v}, \mathbf{u}_e^g, \gamma_h)$ in element e, hot spot h is computed as

$$\sigma_{eh}^{scf}(\mathbf{v}, \mathbf{u}_e^g) = \mathbf{b}_{eh}^{scf}(\mathbf{v}, \gamma_h)\mathbf{T}_e\mathbf{u}_e^g \tag{14}$$

$$\begin{aligned}\mathbf{b}_{eh}^{scf}(\mathbf{v}, \gamma_h) &= SCF_{eh}^{ax}(\mathbf{v})\mathbf{b}_{eh}^{ax}(\mathbf{v}, \gamma_h) + SCF_{eh}^{mi}(\mathbf{v})\mathbf{b}_{eh}^{mi}(\mathbf{v}, \gamma_h) \\ &+ SCF_{eh}^{mo}(\mathbf{v})\mathbf{b}_{eh}^{mo}(\mathbf{v}, \gamma_h)\end{aligned} \tag{15}$$