Financial Insurance GmbH
Offshore Wind Farm Baltic Eagle

Preliminary Design

Collision Analysis Report for WTG-Foundation Structures

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Revision status

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1 Introduction

1.1 Project description

The OWF Baltic Eagle is located in the Baltic Sea within the German exclusive economic zone "Ausschließliche Wirtschaftszone" (AWZ), see Figure 1-1.

The distance to the island of Rügen is ~ 28 km.

The farm will comprise 83 wind turbines of the 5 to 6.15 MW class. Hence the total capacity will amount to 415 to 510 MW. A farm-internal substation (OSS) will be installed.

The offshore wind turbines will be installed on 3-leg jacket foundations includes Transition Pieces and pre-installed piles.

The jacket substructures are designed for loads (wind and waves) with a return period of 50 years, unless stated otherwise. The design service life of the foundation structures for the wind turbine generators (WTGs) shall be 25 years.

![Map of OWF Baltic Eagle in the German AWZ of the Baltic Sea](image.png)

Figure 1-1: Location of OWF Baltic Eagle in the German AWZ of the Baltic Sea

The water depth in the wind farm area varies from 40.3 m to 45.2 m.

The wind turbines are connected to a network of in-field cables and connected to the substation (inter array grid). The cables shall be placed about 1.0 m to 1.5 m below mudline. The installation depth of the cables is governed by the BSH-requirement to avoid a temperature increase of more than 2 K in the inhibited soil zone (BSH-2 K criterion), and shall amount to about 1.0 m to 1.5 m below mudline.

The layout of the OWP Baltic Eagle and the surrounding OWPs and ship routes are shown in Figure 1-2. The locations of the WTGs are marked with points.
1.2 Definitions and abbreviations

1.2.1 Definitions

A wind turbine generator (WTG) as understood in this document consists of a wind turbine and its support structure, see Figure 1-3. The support structure is divided into two parts: the tower and the substructure. The tower is directly connected to the wind turbine. The substructure of WTGs consists of the Transition Piece connecting the tower with the actual foundation. The actual foundation is realised by a 3-leg jacket structure.
Figure 1-3: Definition of offshore wind turbine sections, jacket foundation.
1.2.2 Abbreviations and symbols

Main abbreviations and symbols are listed below.

- BAM Bundesanstalt für Materialforschung und –prüfung (Federal Institute for Materials Research and Testing)
- BAW Bundesanstalt für Wasserbau (Federal Waterways Engineering and Research Institute)
- BSH Bundesamt für Seeschifffahrt und Hydrographie (Federal Maritime and Hydrographic Agency)
- COG centre of gravity
- DMS degrees, minutes, seconds
- E Young’s Modulus
- EEZ exclusive economic zone (German: AWZ)
- HAT highest astronomical tide [m]
- HH hub height
- IMS IMS Ingenieurgesellschaft mbH
- L embedment length of pile
- LAT lowest astronomical tide [m]
- MSL mean sea level [m]
- OWF offshore wind farm
- SCF stress concentration factor
- STD standard deviation
- TDW tons dead weight
- TP Transition Piece
- w.r.t. with reference to
- WEC wind energy converter
- WKU WindKraftUnion AktienGesellschaft
- WSD Wasser- und Schifffahrtsdirektion (Waterways and Shipping Directorate)
- WTG wind turbine generator (synonym for WTG)
- \(\sigma\) normal stress [N/mm²]
- \(P\) density of material
- \(f\) friction coefficient
- \(\Delta\) displacement of a ship

1.3 Motivation and background

The deployment of offshore wind turbines includes an inherent risk of ship collisions with its substructures. The pollution of the marine environment especially by collisions with tankers cannot be excluded. Consequently, the hull-retaining behaviour of substructures designed for site specific conditions has to be proved according to the standard "Design of Offshore Wind Turbines" by BSH (Bundesamt für Seeschifffahrt und Hydrographie, Federal Maritime and Hydrographic Agency) [7] (Part C, Section 2). Detailed requirements are defined in Appendix 1 of [7].

According to the BSH-standard, the hull-retaining behaviour of the foundation structure has to be assessed by a quantitative risk analysis. A risk matrix with risk priority numbers is given in Table 3 of the afore-mentioned Appendix 1, also see Table 1-1. The lines of this risk matrix refer to certain consequence classes (“catastrophic”, “serious” etc.). The classification of a foundation structure is based on the results of a collision analysis. The collision analysis may be carried out with the help of a dynamic calculation using an explicit solution algorithm, e. g. using the FE code LS-DYNA. Some requirements and
boundary conditions are given in [7], Appendix 1. For the assessment of the obtained results it is referred to [2].

Table 1-1: Risk matrix with risk priority numbers, cf. [2] and [7].

<table>
<thead>
<tr>
<th>Catastrophic</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Serious</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Considerable</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>Not significant</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>Frequency of occurrence of the consequence</td>
<td>Extremely rare</td>
<td>Rare</td>
<td>Occasional</td>
<td>Frequent</td>
</tr>
</tbody>
</table>

Table 1-2 relates the consequences given in the previous table to frequencies of occurrence. The table is taken out of the BSH-standard “Design of Offshore Wind Turbines” [7], Appendix 1, Table 4.

Table 1-2: Frequency of occurrence: number of cases per year.

<table>
<thead>
<tr>
<th>qualitative [1/a]</th>
<th>Offshore Wind Turbine / Ship</th>
<th>Environment</th>
<th>Safety</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequent</td>
<td>H &gt; 10^4</td>
<td>H &gt; 2 · 10^4</td>
<td>H &gt; 10^4</td>
</tr>
<tr>
<td>Occasional</td>
<td>10^3 ≤ H &gt; 10^2</td>
<td>2 · 10^3 ≤ H &gt; 10^2</td>
<td>10^4 ≤ H &gt; 10^2</td>
</tr>
<tr>
<td>Rare</td>
<td>10^2 ≤ H &gt; 10^0</td>
<td>2 · 10^2 ≤ H &gt; 10^3</td>
<td>10^4 ≤ H &gt; 10^4</td>
</tr>
<tr>
<td>Extremely rare</td>
<td>10^0 ≤ H</td>
<td>2 · 10^1 ≤ H &gt; 10^5</td>
<td>10^4 ≤ H</td>
</tr>
</tbody>
</table>

The Subject of this report is the planning, execution, and interpretation of the aforementioned collision analyses, which are part of the overall risk analysis.

1.4 Scope of work

The ordered services comprise the execution and documentation of two collision analyses for the 3-leg jacket foundation structure according to the existing Basic Design (cf. drawing 90267-PD-20-02). A collision scenario for an accident assessment ship drifting sideways with a velocity of 2 m/s has to be simulated according to [7]. The modelling procedure and the choice of model parameters and assumptions are based on research results documented in [2].

The following tasks have to be carried out:

- meshing of the foundation structure and the two ships
- definition of simulation model in the input format of LS-DYNA
- implementation of a soil profile in LS-DYNA
- execution of test calculations to verify the input parameters
- execution of final simulations, interpretation and further processing of the results

The definition and calculation of the numerical model is described in the following sections in more detail. Based on the obtained results, the examined foundation structure is
classified according to the risk matrix in Table 1-1, i.e. the line of the risk matrix is determined. The determination of the frequency of occurrence (column in risk matrix) has to be taken out of a corresponding risk analysis [16].

1.5 Procedure

The collision analyses in this report are based on results obtained by research activities at the Institute for Ship Structural Design and Analysis of the Technical University of Hamburg-Harburg from 2002 to 2007 [2], [3], [4] and [6]. The research was supported by the Federal Ministry for the Environment, Nature Conservation and Nuclear Safety. The investigations of ship / OWTGs collisions are based on previous research done at the afore-mentioned institute, e.g. side collisions of double-hulled tankers.

It is pointed out in [2], p. 94, that it was not possible to model all effects during the collision process in the scope of the research work. Prior objective was the comparability of the obtained results for different foundation structures according to [2]. Consequently, the parameters were not chosen due to the most realistic modelling of a single scenario but as consistent over the investigated scenarios as possible. Furthermore, it is pointed out that there are no tests to verify the numerically obtained results. Thus a probable collision scenario was developed and defined in [7], which enables the assessment of possible damages to ships (scenario-based method).

The concepts and assumptions described in [2] are taken for the simulations carried out for this report. This holds for boundary conditions, like the drift velocity and the mass of the colliding ship, as well as for further model parameters, like the definition of the failure criterion for steel and the modelling of the subsoil reaction.
2 Design basis

2.1 Standards and reference documents

All relevant standards and rules are listed in Design Basis Part A [1]. Reference documents explicitly used in this document are listed in the following:


[14] International Association of Classification Societies: Method of correction for the effect of free surface of liquids in tanks (Regulation 10(2), UR L3 and UI LL45), Rev. 1 July 2008.


2.2 Investigated scenarios

In accordance with the new BSH specification (BSH-letter dated at 30th May 2011 Ship hull-retaining configuration of substructure – design ship type for collision analysis) a relevant ship has especially been chosen with regard to the operation area to be examined in this analysis. After consultation of all parties (WSD Nord, BSH, DNV, WKU AG) an agreement has been made for the selection of the two assessment ships on basis of IMS experience.

The first ship is a double-hulled tanker belonging to the Aframax class (“Stena Artica”) and the second one is a ferry of the company Scandlines (“Sassnitz”); cf. Chapter 3.1. These two ships were selected with regard to the frequency of passages and environmental safety.

In the investigated scenarios the ship drifts with a velocity of 2 m/s sideways. It collides with the foundation structure at the bottom part of the ship in its middle, see Figure 2-1. In this way the largest possible collision forces are activated. A relative movement between ship and foundation structure is not induced. Additional velocity components beside the described horizontally directed drift velocity are not taken into account in the investigated scenario. The water level is set to MSL ± 0 m in the simulation.

Further loads like wind, currents, or waves are not accounted for in the investigated scenario additionally to their indirect influence on the drift velocity, cf. Section 3.1.5 on this topic.

![Initial configuration of the collision process](image)

2.3 Software

The collision analyses are carried out using the Finite Element Method (FEM) and by choosing an explicit solution algorithm according to [7]. The program package LS-DYNA was applied for the simulation.

LS-DYNA is a multi-purpose FE-programme with implicit and explicit solvers. The focus of the programme is on transient analyses of multi-nonlinear, three-dimensional, highly dynamic systems (e.g. crash simulations for the car industry). The collision of ships with offshore foundation structures is comparable to crash simulations. Consequently, LS-DYNA was also selected for the present simulations.
The element types described in the following list are used by LS-DYNA in the given sim-
ulations, also see [10]:

- **Mass point**: definition via the card "**ELEMENT_InERTIA"", mass point with inertia
  forces

- **Springs**: definition via the card "**ELEMENT_DISCRETE", translatory or rotational
  spring

- **Beam element**: Belytschko-Schwer resultant beam, standard beam element with
direct input of sectional properties

- **Shell element**: Belytschko-Tsay (standard shell element)

- **C^0 triangular shell**, similar to square Belytschko-Tsay-element

The chosen element types are compatible with each other.

### 2.4 Material behaviour

#### 2.4.1 General aspects

The correct modelling of the actual material behaviour is of some importance for the
given simulations. The components of the collision partners meshed with shell elements
are modelled with material 24 of LS-DYNA as used in [2] and recommended in [7] for
metals Thus, a linear-elastic, isotropic-plastic material behaviour is described while a
material solidification can be taken into account as well. For this purpose a user-defined
stress-strain-relationship is given for the plastic domain. Elastic deformation is described
by Hooke’s law via Young’s modulus and Poisson’s ratio.

#### 2.4.2 Stress-strain-relationship

The definition of the function $\sigma_r = f(\varepsilon_{pl}^r)$ for the material model Mat_24 is taken out
of [2] for the calculations done in the scope of this report. The corresponding file was
provided by the Institute for Ship Structural Design and Analysis of the Technical Unive-
sity of Hamburg-Harburg, cf. Section 1.5. The provided table of values describes the
behaviour of steel type S235. This type of material is used for all parts of the ship struc-
ture. Steel type S355 is assumed for the foundation structure. As a reference for the
Corresponding stress-strain relation, the values for the yield stresses of the material
S235 were scaled by multiplying them with the factor 355/235 = 1.51.

The both stress-strain-diagrams used for the materials are depicted in Figure 2-2. It
should be pointed out that the stress-strain-curves are only used in the simulations up to
predefined ultimate plastic strain limits, see Section 2.4.3.
2.4.3  Failure criterion

It is of decisive importance for the classification of a collision scenario into a consequence class, whether the outer hull breaks and the ship springs a leak, cf. Section 1.5. Therefore, the definition of the criterion of failure or breaking plastic strain for the ship steel directly influences the assessment of collision consequences. Furthermore, a later breaking of the foundation steel may reduce the risk of a collapsing wind turbine. On the other hand side, this also leads to a stronger substructure and may cause that the ship hull loses its integrity more easily. Especially concerning the last aspect, the correct choice for the criterion of failure is of some importance in collision analyses.

For the simulative calculation described here and the application of the material model 24 from JS-DYNA which took place in [2], the failure criterion is determined by defining an elongation at break of $\varepsilon_{\text{eff, break}}^p$. For each Mat-24 element, the calculated effective plastic elongation $\varepsilon_{\text{eff}}^p$ is compared to the elongation at break $\varepsilon_{\text{eff, break}}^p$ during each time step of the simulation. If the effective plastic strain exceeds the predefined value of failure plastic strain in all integration points, the element is deleted and removed from the model.

The value taken for the decisive simulation in [2] amounts to 12% for the ship steel and to 20% for the foundation steel. Especially for the steel of the foundation structures, failure strains down to 3% are discussed due to the large element dimensions in combination with high sheet thicknesses in this area. Especially for the foundation steel with thick plates modelled by large elements, failure strains down to 3% are discussed. For the considerations in this report the procedure defined in [2] is taken over in principle. However, the values for the failure strain are not varied and set to larger values considered in most cases to be on the safe side. The following values are taken in these collision analyses:
- $\varepsilon_{\text{eff, failure}}^{\text{P}} = 12\%$ for ship steel and
- $\varepsilon_{\text{eff, failure}}^{\text{P}} = 20\%$ for foundation steel.

### 2.4.4 Friction coefficients

Only a relatively small part of the total collision energy is dissipated by friction during the simulation. As a consequence, the choice of the friction coefficients is less important than the selection of parameters discussed in the previous sections.

A value of $f=0.3$ is taken for dynamic friction and of 0.4 for static friction in the collision simulations. Nevertheless, quite low values are determined for the energy due to friction. This confirms the above-mentioned assumption a posteriori that the influence is of less importance for the results.
3 Modelling

3.1 First assessment ship: Double-hulled tanker

3.1.1 Description of the structure

For the first collision scenario, a double-hulled tanker with 113,500 tdw is being implemented as assessment ship. The ship's design parameters are summarised in Table 3-1. Figure 3-1 shows the selected ship.

Table 3-1: Main parameters of double-hulled tanker.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Abbreviation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship type</td>
<td>-</td>
<td>Double-hulled tanker</td>
</tr>
<tr>
<td>Deadweight</td>
<td>TDW</td>
<td>113,500</td>
</tr>
<tr>
<td>Total mass / displacement</td>
<td>m&lt;sub&gt;ship&lt;/sub&gt; / ∆₁₅,₁₄₅m</td>
<td>137,000 t</td>
</tr>
<tr>
<td>Width</td>
<td>b</td>
<td>44.0 m</td>
</tr>
<tr>
<td>Draught</td>
<td>t</td>
<td>15.4 m</td>
</tr>
<tr>
<td>Height</td>
<td>h</td>
<td>21.3 m</td>
</tr>
<tr>
<td>Length</td>
<td>lₚ</td>
<td>250.0 m</td>
</tr>
</tbody>
</table>

Figure 3-1: Double-hulled tanker "Stena Artica".

3.1.2 Structural model

A 32.0 m long middle segment of the double-hulled tanker between two bulkheads is meshed in detail with all important structural components, see Figure 3-2. This part of the ship model functions as the deformable body which directly collides with the substructure. Contact and friction forces as well as material non-linearities, inertia forces, and large deformations are considered. The mass of the segment amounts to m<sub>s</sub>=1,127 t and is considered as part of the total mass.
All further masses (the rest of the ship structure, cargo, etc.) as well as the structural behaviour and the interaction between fluid and vessel are modelled with the help of rigid bodies, beam elements, and discrete spring elements, which are connected to the meshed ship segment via both bulkheads. Figure 3-3 provides an overview of the ship model.

Figure 3-3 shows the division of the ship body into four cuboidal, idealised parts. The two central cuboids model the mass of the cargo in this part, m_i, i.e. the mass which is not part of the ship structure. The outer cuboids represent the remaining ship mass including the cargo (m_a + m_b). Height and width are equal to the values given in Table 3-1 for the accident assessment ship. The length of the central cuboids amounts to 16.0 m each corresponding to the total length of the meshed segment of 32.0 m. The length of the outer cuboids amounts to 84.0 m each, summing up to 200.0 m for all cuboids. Thus, the block is a bit shorter than the real ship, but has the same displacement at full draught while the mass inertias of the prow and the stern of the ship are illustrated with sufficient precision.
In LS-DYNA, the mass of the beam elements connecting the outer blocks is being calculated according to their geometry. Since the beams are to depict the ship’s cross-section, their mechanical parameters have to correspond with those of the ship. The cross-section values of a regular double-hulled tanker are summarised in Table 3-2.

Table 3-2: Section properties of the beam model for the non-detailed part of the double-hulled tanker.

<table>
<thead>
<tr>
<th>Size</th>
<th>Abbreviation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Position of the gravity centre above the</td>
<td>$z_g$</td>
<td>10.65 m</td>
</tr>
<tr>
<td>keel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cross-sectional area</td>
<td>$A_b$</td>
<td>5.969 m²</td>
</tr>
<tr>
<td>Second moment of inertia around the $y$-</td>
<td>$I_{yy}$</td>
<td>549.63 m⁴</td>
</tr>
<tr>
<td>axis</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Second moment of inertia around the $z$-</td>
<td>$I_{zz}$</td>
<td>1,412.39 m⁴</td>
</tr>
<tr>
<td>axis</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Second torsional moment</td>
<td>$I_p$</td>
<td>834.82 m⁴</td>
</tr>
<tr>
<td>Young’s modulus</td>
<td>$E$</td>
<td>210,000 MN/m²</td>
</tr>
<tr>
<td>Density</td>
<td>$\rho_b$</td>
<td>7,850 kg/m³</td>
</tr>
</tbody>
</table>

The gravity centre of the ship’s hull is supposed to be at lateral mid-height. The mass of a beam is:

$$m_b = \rho_b \frac{l_a}{2} A_b = 1,968 \, t$$

whereas the length of the beam is defined as $\frac{l_a}{2} = 42m$. The beam elements are arranged horizontally, thus, their axis run over the whole length at the height of the assumed centre of the cross-section.

The mass $m_i$ constitutes the cargo of the tanks which is situated in the discretised part of the model. The volume of the tanks is approximately 22,930 m³. With an oil density of 0.9 t/m³, the cargo has a weight of 20,640 tons and $m_i = 10,320$ t.

The mass of the outer cuboids can be calculated as:

$$m_a = \frac{\Delta_{15.4m} - (m_s + 2m_b + 2m_i)}{2} = 55,695 \, t$$

Whereas $m_s$ is the mass of the discretised structure and $\Delta_{15.4m}$ corresponds to the total mass of the ship.

All four described cuboids are modelled in the simulation as mass points. It is assumed that the mass is distributed homogeneously over each cuboid. The moments of inertia are calculated according to this assumption.

For the two outer cuboids the moments of inertia are referred to their centres of gravity. These two mass points will be connected stiffly to the discretised model of the central segment by means of the beam segments mentioned above.
The moments of inertia for the outer cuboids are determined to the following values:

\[ l_{xx,A} = \frac{1}{12} \cdot m_a \cdot h^2 + b^2 = 1,109 \cdot 10^{10} \text{ kg.m}^2 \]

\[ l_{yy,A} = \frac{1}{12} \cdot m_a \cdot h^2 + l_a^2 = 3,485 \cdot 10^{10} \text{ kg.m}^2 \] (2.1)

\[ l_{zz,A} = \frac{1}{12} \cdot m_a \cdot b^2 + l_a^2 = 4,173 \cdot 10^{10} \text{ kg.m}^2 \]

It is assumed that the x-axis points in the direction of the ship, the y-axis to portside and the z-axis upwards.

The moments of inertia of the central cuboids, which are representing the cargo in the middle segment, are determined in the same way. However, the values are not referred to the centre of each cuboid but to the inner ends of the beam elements. Applying the Steiner ratio to be considered, the following values are determined:

\[ l_{xx,I} = \frac{1}{12} \cdot m_i \cdot h^2 + b^2 = 2,055 \cdot 10^9 \text{ kg.m}^2 \]

\[ l_{yy,I} = \frac{1}{12} \cdot m_i \cdot h^2 + l_i^2 + m_i \cdot \frac{l_i^2}{2} = 1,271 \cdot 10^9 \text{ kg.m}^2 \] (2.2)

\[ l_{zz,I} = \frac{1}{12} \cdot m_i \cdot b^2 + l_i^2 + m_i \cdot \frac{l_i^2}{2} = 2,546 \cdot 10^9 \text{ kg.m}^2 \]

Figure 3-4 shows the applied ship model once more.

\[ \text{Figure 3-4: Summarizing depiction of the first ship model.} \]

### 3.1.3 Material definition

A linear-elastic constitutive model is used for the parts of the ship which are modelled with beam elements. This does not constitute any restriction of the resulting accuracy since the yield strength in these parts is far from being reached during the simulation.

For the detailed part (cf. Figure 3-2), steel S235 is assumed analogous to [2] and material model 24 of LS-DYNA is used featuring a linear-elastic / nonlinear-plastic material behaviour, cf. Section 2.4.
3.1.4 Interaction between water and ship

In order to model the ship behaviour in a realistic way, the interaction between the movement of the ship and the hydromechanical forces acting on the ship has to be considered.

The consideration of the hydrostatical restoring forces acting on the ship is of primary importance for these simulations. This is done by implementing eight spring elements into the LS-DYNA model. These spring elements are part of the LS-DYNA’s general element library. Compatibility problems are therefore avoided.

The arrangement of the spring elements is depicted in Figure 3-5: four translational and four rotational springs. Each of these springs act on one end of the beam elements described in Section 3.1.2. The connection to the detailed model is realised by the rigid planes at the bulkheads. The distribution of the total spring stiffness for both types of springs is done proportionally to the length of the respective cuboid. The determination of each spring stiffness is described in the following paragraphs.

It should be remarked to Figure 3-5 that the translational springs as well as the rotational springs are defined in reference to the global fixed coordinate system. I.e. the restoring force caused by a displacement in z-direction is not influenced by e.g. the horizontal drift. The same applies for the rotational springs.

The increase of the restoring force with growing displacements is linear and can be modelled with the linear spring elements accurately. The total spring stiffness $C_{t,\text{tot}}$ of all translational springs is determined in the following way:

$$ C_{t,\text{tot}} = d \Delta \cdot g = 94,43 \text{ MN/m} $$

(2.3)

With $d \Delta = 98.3 \text{ t/cm}$, the gradient of water displacement taken out of the hydrostatic table for the draft $T=15.4 \text{m}$.

The distribution to the four translational springs is proportional to the lengths given in Figure 3-3:

$$ C_{T,1} = \frac{16}{200} \text{C}_{t,\text{tot}} = 7.71 \text{ MN/m} $$

$$ C_{T,2} = \frac{84}{200} \text{C}_{t,\text{tot}} = 40.5 \text{ MN/m} $$

(2.4)
The assumed position of the centre of buoyancy $B_0$, of the metacentre $M_0$ and of the centre of gravity $G$ is depicted in Figure 3-6. The half draught $T$ is taken as the distance between the centre of buoyancy $B_0$ and the keel. The half height $h$, according to Table 3-1, is taken as the distance between the centre of gravity $G$ and the keel.

![Figure 3-6: Cross-section with metacentre $M_0$, centre of gravity $G$, and centre of buoyancy $B_0$ for double-hulled tanker.](image)

The metacentric height $GM_0$ (distance between the centre of gravity $G$ and the metacentre $M_0$) describes the ship’s initial stability and corresponds with:

$$GM_0 = KB_0 + B_0M_0 - KG$$

where:

$$KB_0 = \frac{T}{2} = 7.71m$$

$$KG \approx \frac{d}{2} = 11.02m$$

$$B_0M_0 = \frac{I_{WLx}}{\Delta_{15.4m}} = 10.62m$$

The waterline’s moment of inertia around the x-axis $I_{WLx}$ of the ship is approximated as follows:

$$I_{WLx} = \frac{I_{ges} \cdot b^3}{12} = 1.42 \cdot 10^6 m^4$$

Then, the metacentric height is:

$$GM_0 = KB_0 + B_0M_0 - KG = 7.32m$$

The presence of a free surface in the tanks would theoretically reduce the ship’s stability. However, in [14] it is mentioned that this effect must not be considered in case of a charging level exceeding 98%. Since, in this scenario, the ship is fully loaded, the impact of the free surface is being neglected.

The presence of a free surface in the tanks would theoretically reduce the ship’s stability. However, in [14] it is mentioned that this effect must not be considered in case of a charging level exceeding 98%. Since, in this scenario, the ship is fully loaded, the impact of the free surface is being neglected.
The overall stiffness of the torsion springs amounts to:

\[ C_{R,tot} = GM_0 \cdot \Delta_{15.4m} : g = 9.838 \cdot 10^6 \frac{Nm}{Rad} \]

The stiffness coefficient of the outer and the inner springs is defined according to the length of the individual cuboids:

\[ C_{R,i} = \frac{16m}{200m} C_{R,tot} = 787 \, MNm/Rad \]
\[ C_{R,a} = \frac{84m}{200m} C_{R,tot} = 4131 \, MNm/Rad \]

The values are summarized for each spring in Table 3-3.

<table>
<thead>
<tr>
<th>Spring stiffness</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Translational spring, inside</td>
<td>( C_{T,i} )</td>
<td>7.71 MN/m</td>
</tr>
<tr>
<td>Translational spring, outside</td>
<td>( C_{T,a} )</td>
<td>40.5 MN/m</td>
</tr>
<tr>
<td>Rotational spring, inside</td>
<td>( C_{R,i} )</td>
<td>787 MNm/Rad</td>
</tr>
<tr>
<td>Rotational spring outside</td>
<td>( C_{R,a} )</td>
<td>4131 MNm/Rad</td>
</tr>
</tbody>
</table>

### 3.1.5 Loads on the ship

The collision scenario is described in [7]. Thus, a sideways drifting ship with a velocity of 2 m/s has to be considered. Working loads, like wind, waves or currents, which may lead to the assumed drift velocity in reality, are not considered directly. The same assumptions are made in [2]. Accordingly, no working loads on the ship’s body are taken into account in these simulations. The movement of the ship due to the sea state and the tidal range are also not regarded.
3.2 Second assessment ship: Ferry

3.2.1 Description of the structure

For the second collision scenario, a ferry with 4,700 tdw is being implemented as assessment ship. The ship’s design parameters are summarised in Table 3-4. Figure 3-7 shows the selected ship.

Table 3-4: Main parameters of the ferry.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Abbreviation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship type</td>
<td>-</td>
<td>Ferry</td>
</tr>
<tr>
<td>Deadweight</td>
<td>TDW</td>
<td>4 700 t</td>
</tr>
<tr>
<td>Total mass / displacement</td>
<td>$m_{\text{ship}} / \Delta_{15,4\text{m}}$</td>
<td>14 200 t</td>
</tr>
<tr>
<td>Width</td>
<td>b</td>
<td>23.7 m</td>
</tr>
<tr>
<td>Draught</td>
<td>t</td>
<td>5.7 m</td>
</tr>
<tr>
<td>Height</td>
<td>h</td>
<td>18.8 m</td>
</tr>
<tr>
<td>Length</td>
<td>$l_{\text{ship}}$</td>
<td>165.3 m</td>
</tr>
</tbody>
</table>

Figure 3-7: The ferry "MS Sassnitz".

3.2.2 Structural model

The 20.0 m long middle segment of the ferry between two bulkheads is meshed in detail with all important structural components, see Figure 3-8. The mass of the segment amounts to $m_s=618$ t.

All further masses (the rest of the ship structure, cargo, etc.) as well as the structural behaviour and the interaction between fluid and vessel are modelled with the help of rigid bodies, beam elements, and discrete spring elements, which are connected to the meshed ship segment via both bulkheads. Figure 3-9 provides an overview of the ship model.
Figure 3-8: Meshed segment of the ferry (parts of the outer shell are not depicted)

Figure 3-9 shows the division of the ship body into four cuboidal, idealised parts. The division of the total mass of the ship into the individual masses $m_a$, $m_b$, $m_i$ and $m_b$ is done on the same way than explained for the first assessment ship in section 3.1.2. The length of the central cuboids amounts here to 10.0 m each corresponding to the total length of the meshed segment of 20.0 m. The length of the outer cuboids amounts to 60.0 m each, summing up to 140.0 m for all cuboids.

Figure 3-9: Model for the non-detailed part of the ferry.

The cross-section values of a regular double-hulled tanker are summarised in Table 3-5. For the calculation of the masse of the beam $m_b$, these values are taken as reference.
Table 3-5: Section properties of the beam model for the non-detailed part of the ferry

<table>
<thead>
<tr>
<th>Size</th>
<th>Abbreviation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Position of the gravity centre above the keel</td>
<td>$z_s$</td>
<td>9.4 m</td>
</tr>
<tr>
<td>Cross-sectional area</td>
<td>$A_b$</td>
<td>3.33 m²</td>
</tr>
<tr>
<td>Second moment of inertia around the y-axis</td>
<td>$I_{yy}$</td>
<td>183.2 m⁴</td>
</tr>
<tr>
<td>Second moment of inertia around the z-axis</td>
<td>$I_{zz}$</td>
<td>201.7 m⁴</td>
</tr>
<tr>
<td>Second torsional moment</td>
<td>$I_p$</td>
<td>157.5 m⁴</td>
</tr>
<tr>
<td>Young’s modulus</td>
<td>$E$</td>
<td>210,000 MN/m²</td>
</tr>
<tr>
<td>Density</td>
<td>$\rho_b$</td>
<td>7,850 kg/m³</td>
</tr>
</tbody>
</table>

The gravity centre of the ship’s hull is supposed to be at lateral mid-height. The mass of a beam is:

$$m_b = \rho_b \frac{l_a}{2} A_b = 784 \text{ t}$$

whereas the length of the beam is defined as $\frac{l_a}{2} = 30m$.

The mass $m_i$ constitutes the cargo of the tanks which is situated in the discretised part of the model. The cargo has a weight of 628 tons thus $m_i = 314 \text{ t}$.

The mass of the outer cuboids can be calculated as:

$$m_a = \frac{\Delta_{5,7m} - (m_s + 2m_b + 2m_i)}{2} = 5,693 \text{ t}$$

Whereas $m_s$ is the mass of the discretised structure and $\Delta_{5,7m}$ corresponds to the total mass of the ship.

The moments of inertia for the outer cuboids are determined to the following values:

$$I_{xx,a} = \frac{1}{12} \cdot m_a \cdot h^2 + b^2 = 4,341 \cdot 10^8 \text{ kg.m}^2$$

$$I_{yy,a} = \frac{1}{12} \cdot m_a \cdot h^2 + l_a^2 = 1,876 \cdot 10^9 \text{ kg.m}^2$$

$$I_{zz,a} = \frac{1}{12} \cdot m_a \cdot b^2 + l_a^2 = 1,974 \cdot 10^9 \text{ kg.m}^2$$

The coordinate system is defined as in section 3.1.2.

Applying the Steiner ratio to be considered, the moments of inertia of the central cuboids are calculated as following:
\[
I_{xx,1} = \frac{1}{12} \cdot m_i \cdot h^2 + b^2 = 2,394 \cdot 10^7 \text{ kg.m}^2 \\
I_{yy,1} = \frac{1}{12} \cdot m_i \cdot h^2 + \frac{l_i^2}{4} + m_i \cdot \frac{l_i^2}{2} = 1,972 \cdot 10^7 \text{ kg.m}^2 \\
I_{zz,1} = \frac{1}{12} \cdot m_i \cdot b^2 + l_i^2 + m_i \cdot \frac{l_i^2}{2} = 2,516 \cdot 10^7 \text{ kg.m}^2
\]

(2.6)

Figure 3-10 shows the applied ship model once more.

3.2.3 Material definition

For the model considered in this simulation, the same material properties as for the first scenario apply; see the details in section 3.1.3.

3.2.4 Interaction between water and ship

The hydromechanical forces acting on the ship are implemented in the model similarly to section 3.1.4.

The arrangement of the spring elements is depicted once again in Figure 3-11.

Figure 3-11: Modelling of hydrostatical restoring forces via spring elements.
The total spring stiffness $c_{T,\text{tot}}$ of all translational springs is determined in the following way:

$$c_{T,\text{tot}} = d\Delta \cdot g = 31.7 \text{ MN/m} \quad (2.7)$$

With $d\Delta = 32.3 \text{ t/cm}$, the gradient of water displacement taken out of the hydrostatic table for the draft $T=5.7\text{m}$.

The distribution to the four translational springs is proportional to the lengths given in Figure 3-9:

$$c_{T,1} = \frac{10}{140} c_{T,\text{tot}} = 2.26 \text{ MN/m} \quad (2.8)$$

$$c_{T,a} = \frac{60}{140} c_{T,\text{tot}} = 13.59 \text{ MN/m}$$

The metacentric height $GM_0$ (distance between the centre of gravity $G$ and the metacentre $M_0$, cf. Figure 3-6) describes the ship’s initial stability and corresponds with:

$$GM_0 = KB_0 + B_0M_0 - KG$$

where:

$$KB_0 = \frac{T}{2} = 2.85 \text{ m}$$

$$KG \approx \frac{h}{2} = 9.4 \text{ m}$$

$$B_0M_0 = \frac{i_{W,L,x} \rho}{\Delta_{S,7m}} = 11.18 \text{ m}.$$ 

The waterline’s moment of inertia around the $x$-axis $i_{W,L,x}$ of the ship is approximated as follows:

$$i_{W,L,x} = \frac{I_{\text{gas}} \cdot b^3}{12} = 1.55 \cdot 10^5 \text{m}^4$$

Then, the metacentric height is:

$$GM_0 = KB_0 + B_0M_0 - KG = 4.63\text{m}$$

The overall stiffness of the torsion springs amounts to:

$$C_{R,\text{tot}} = GM_0 \cdot \Delta_{S,7m} \cdot g = 6.450 \cdot 10^8 \text{N/m} \quad \text{Rad}$$

The stiffness coefficient of the outer and the inner springs is defined according to the length of the individual cuboids:

$$C_{R,l} = \frac{10\text{m}}{140\text{m}} C_{R,\text{tot}} = 4.607 \cdot 10^7 \text{N/m} \quad \text{Rad}$$

$$C_{R,a} = \frac{60\text{m}}{140\text{m}} C_{R,\text{tot}} = 2.764 \cdot 10^8 \text{N/m} \quad \text{Rad}$$

The values are summarized for each spring in Table 3-6.
Table 3-6: Spring stiffness for the beam model for the non-detailed part of the double-hulled tanker.

<table>
<thead>
<tr>
<th>Spring stiffness</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Translational spring, inside</td>
<td>$c_{T,I}$</td>
<td>2.26 MN/m</td>
</tr>
<tr>
<td>Translational spring, outside</td>
<td>$c_{T,A}$</td>
<td>13.59 MN/m</td>
</tr>
<tr>
<td>Rotational spring, inside</td>
<td>$c_{R,I}$</td>
<td>46.1 MNm/Rad</td>
</tr>
<tr>
<td>Rotational spring outside</td>
<td>$c_{R,A}$</td>
<td>276.4 MNm/Rad</td>
</tr>
</tbody>
</table>

3.3 Wind turbine

3.3.1 Description of WTG-structure

The actual wind turbine generator (WTG) mainly consists of tower, nacelle and rotor blades and is installed on the foundation structure. The rotor blades consist of three blades and the rotor hub. The connection between WTG and substructure is realised with bolted flanges.

As the influence of the WTG on the investigated collision friendliness is small, a simplified WTG model of the type AREVA M5000-135 with a rated capacity of 5.0 MW is applied. The WTG model is derived from data provided by the manufacturer, see [15].

Table 3-7: Data of WTG of type AREVA WIND M5000-135.

<table>
<thead>
<tr>
<th>WTG data of type AREVA M5000-135</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated capacity</td>
</tr>
<tr>
<td>Hub height</td>
</tr>
<tr>
<td>Interface level (bottom of tower)</td>
</tr>
<tr>
<td>Length of tower</td>
</tr>
<tr>
<td>Diameter of tower, top</td>
</tr>
<tr>
<td>Diameter of tower, bottom</td>
</tr>
<tr>
<td>Tower mass</td>
</tr>
<tr>
<td>Rotor mass (hub and blades)</td>
</tr>
<tr>
<td>Rotor COG (w.r.t. tower top centre)</td>
</tr>
<tr>
<td>Rotor mass moment of inertia (w.r.t. rotor COG)</td>
</tr>
<tr>
<td>Nacelle mass</td>
</tr>
<tr>
<td>Nacelle COG (w.r.t. tower top centre)</td>
</tr>
<tr>
<td>Nacelle mass moment of inertia (w.r.t. nacelle COG)</td>
</tr>
</tbody>
</table>
3.3.2 Model of WTG-structure

The WTG of type AREVA M5000-135 is transferred into a model for the collision analysis. This is done in such a way that all parts of the WTG are modelled sufficiently accurately regarding their influence on the results.

Accordingly, nacelle and rotor blades are discretised in a simplified way, see Figure 3-13, and modelled as mass point with inertia properties, also see [2]. The discretised shape is defined as rigid body and fixed to the mass point. Therefore, neither rotor blades nor nacelle can be deformed. The mass of the point amounts to 365 t. The mass point is located at the combined centre of gravity for the nacelle and the rotor blades according to their respective masses. The coordinates of the centre of gravity and the resulting moments of inertia for the mass point are shown in the Table 3-8.

Table 3-8: Data of the mass point for the rotor and the nacelle.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Unity</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Global coordinates (in the local coordinate system from Figure 3-12)</td>
<td>[m]</td>
<td>X=-2.58</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Y=0.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Z=4.15</td>
</tr>
<tr>
<td>Moment of inertia</td>
<td>[kg.m²]</td>
<td>Iₓₓ=3.16E10⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Iᵧᵧ=3.26E10³</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Izz=3.25E10⁷</td>
</tr>
</tbody>
</table>

Figure 3-12: Local coordinate system with the origin on the top edge of the tower.
The tower of the WTG is meshed with shell elements and defined as deformable body. The mass of 270 t is distributed uniformly over the two parts of the tower shaft (upper part 87 t, lower part 183 t). For each part a mean plate thickness is determined. The first 13.5m of the tower are cylindrical with a diameter of 6.0 m and taper then to 4.0 m at the top. Nacelle and the top of the shaft are coupled rigidly, just as the bottom of the tower and the TP. No working loads are applied on the WTG, because their influence on the results is negligible.

### 3.3.3 Material definition

The material of the tower is assumed to be steel type S355. The applied stress-strain-relationship is described in Section 2.4.2. A failure criterion of 20 % effective plastic strain is used following the discussion in Section 2.4.3. Further information on the material behaviour can be found in Section 2.4.

### 3.4 Substructure

#### 3.4.1 Description of 3-leg Jacketstructure

The jacket structure modelled in the collision analysis is a three-legged structure. At its top, the structure is connected to the tower of the wind turbine described in section 3.3 by means of a bolted flange connection. The foundation consists of three main legs, inclined to the vertical centre axis and connected by braces. The main legs have a diameter of 1.27m, which increases up to 3.40m in the lower part. The diameter of the braces is 0.609m. The distance between the legs at the mudline is 33.4m. Where the braces are connected to the main legs, the wall thickness of the tubular members is increased. These parts of the main legs are so-called Cans.
As a foundation steel piles are hammered into the bottom, to which the jacket is connected by means of a grouted connection. Therefore, main legs are inserted into the foundation piles (2.89m in diameter) for up to 4.35m. Over this length grouting material is filled in the space between the legs and the piles.

For more details on the jacket’s geometry please refer to the drawing 90267-PD-20-02 Rev.0.

3.4.2 Model of structure

The model of the substructure discretised with shell elements is shown in Figure 3-14. The nodes are located on the outer surface of the actual steel plates. Thus, the contact surfaces of all structural parts are modelled close to reality.

In LS-DYNA a rigid connection is created between the neighbouring nodes of the pile and the leg. In this way the grouted connection is modelled. As the axial forces applied on the main legs during the collision are far below the strength limit of the grout connection used, this way of modelling is appropriate. In the estimate of the allowable interface shear strength of the grout connection, both, the strength due to friction and the strength due to shear keys were considered.

Figure 3-14: Model of the substructure, bedding springs are depicted with small rings.

All structural components are mainly discretised by means of four-sided shell elements. In some areas, triangular elements are used.

In circumferential direction of the piles and the legs as well as the horizontal and diagonal struts and braces, 12 elements are positioned respectively. The different material thicknesses of the pile sections, cans and struts are considered in the model.

Figure 3-15 and Figure 3-16 show respectively an example of a connecting node and the transition piece.
The four-node shell elements are realised in LS-DYNA according to Belytschko / Tsay and the compatible C⁰-triangle element are selected for the three-node shell element. The whole model for the substructure comprises 15277 nodes and 16921 elements.

The interaction between foundation piles and subsoil is modelled with non-linear bedding springs according to API rules [13], also see Section 5.1.3 in [2]. The horizontal or lateral load bearing capacity is thus realised with non-linear p-y-curves. The vertical or axial load bearing capacity is modelled with t-z-curves for the shaft friction and a Q-z-curve for the pile end bearing capacity. The soil profile given in drawing 90267-PD-20-02 Rev.0 is taken. For the skin friction in chalk 60 kPa is chosen. The bedding springs are depicted in Figure 3-14 as small rings around the pile.

Figure 3-17 shows the complete model including WTG and the middle segment of the accident assessment ship in case of scenario 1.
3.4.3 Material definition

Steel of type S355 is assumed for the material of the jacket. The applied stress-strain-relationship is described in Section 2.4.2. A failure criterion of 20% effective plastic strain is used following the discussion in Section 2.4.3. Further information on the material behaviour can be found in Section 2.4.
4 Results

In the following sections the results of the executed simulations are described and depicted in detail. Beforehand, it has to be pointed out once again that the calculated results are based on the described models and assumptions. Therefore, the following statements are only valid in the scope of the applied models and parameters, cf. for example [12].

4.1 Scenario 1: Collision double-hulled tanker / jacket structure

Figure 4-1 shows the chronological development of the simulated collision between double-hulled tanker and jacket in compliance with the requirements defined in [2] and [7]. The calculated ship velocity is displayed for each time step, also see Figure 4-2. The calculated space of time amounts to 21 s while the intervals are indicated by steps of 2s. At the point of time t = 1.0 s, the ship and the frontal leg of the jacket foundation are already in contact.

The corresponding segment of the jacket structure cannot absorb the ship impact and fails due to the formation of a plastic link at the height of the ship’s bottom edge (bilge of the ship’s body). Shortly after that, the foundation structure fails step by step since the diagonal and horizontal braces buckle; see Figure 4-3. The legs of the jacket buckle as well which leads to the fall of the wind turbine.

After several oscillations, the WTG is accelerated in the opposite direction of the ship’s drift and hits the water surface behind the ship with a sufficient distance.

In the course of the simulation, the ship segment penetrates the foundation structure more and more and is gradually slowed down, cf. Figure 4-2. After 21s the ship drifts on with a speed of approximately 0.9 m/s.
Figure 4-1: Chronological development of simulated collision tanker / jacket
The back-calculated collision force is shown in Figure 4-4. The collision force is the force which acts between ship and foundation structure during contact. The force is here determined via the deceleration of the ship multiplied by its mass. The maximum value amounts to about 13 MN and is reached after 2s.
The development of various energies during the simulation is depicted in Figure 4-5. The top diagram in Figure 4-5 shows the chronological curves and the bottom diagram shows the curves related to the displacement of the ship. At the start of the simulation the kinetic energy of the ship amounts to:

\[ E_{\text{kin}} = \frac{1}{2} m v^2 = \frac{1}{2} 137,000 \, \text{t} \cdot 2.0^2 \, \text{m}^2/\text{s}^2 = 274,000 \, \text{kJ} \]

The kinetic energy decreases continuously in analogy to the drift velocity of the ship until reaching about 65 MJ. The internal and kinetic energies of the WTG increase at the same time. The internal energy of the wind turbine contains that amount of energy which is absorbed by the foundation structure by means of the deformation during the collision. In comparison, the internal energy of the ship, i.e. basically the energy of deformation and work of the ship remains low.

From this can be directly deduced that the kinetic energy of the ship is predominantly absorbed by the foundation structure and that the deformations and, thus, the damages to the ship are rather moderate. In the course of the collision and while the structure tips over, the potential energy of the nacelle and the rotor blades is transformed into kinetic/deformation energy. In the following diagrams, the potential energy of the WTG is not included.
Figure 4-5: Development of energies during the collision tanker / jacket; top: related to time, bottom: related to displacement of ship.
Figure 4-6 depicts the stress resultants at the top of the tower. The moments around the Y-axis show a maximum of approximately 35 MNm due to the impact of the ship.

![Graph showing stress resultants](image)

The stress resulting from the maximum moment of 35 MNm shows a value far below the yield stress of the material. Consequently, no collapse of the tower at its top leading to a separation of the tower and the nacelle is expected. This statement is only for information purposes and should not be taken as conclusion of this report.

\[
\sigma_{\text{max}} = \frac{M_{x,\text{max}} \cdot R}{I_x} = \frac{35 \text{ MNm} \cdot 4.0 \text{ m}}{0.495 \text{ m}^4 \cdot 2} = 141 \text{ MPa}
\]

\[
\sigma_{\text{max}} \cdot \gamma_f = 141 \text{ MPa} \cdot 1.1 = 155.6 \text{ MPa} < \frac{\sigma_{\text{yield}}}{\gamma_m} = \frac{355 \text{ MPa}}{1.1} = 323 \text{ MPa}
\]

### 4.2 Scenario 2: Collision ferry / jacket structure

Figure 4-7 shows the chronological development of the simulated collision between ferry and jacket in compliance with the requirements defined in [2] and [7]. The calculated ship velocity is displayed for each time step, also see Figure 4-8. The calculated space of time amounts to 6 s while the intervals are indicated by steps of 1s. At the point of time \( t = 1.0 \) s, the ship and the frontal leg of the jacket foundation are already in contact.

The drifting velocity of 2m/s is not sufficient to topple the wind turbine. During the simulation, some diagonal braces of the jacket buckle slightly, without consequence on the global stability of the structure. As result of the collision, the ship’s body is deformed, but there is no formation of cracks, see Figure 4-9.

In the course of the simulation, the ship segment is gradually slowed down, cf. Figure 4-8. After about \( t=2.2 \) s, the ship is completely stopped and is pushed back in reason of the elastic deformation of the jacket and foundation piles.
Figure 4-7: Chronological development of simulated collision ferry / jacket
Figure 4-8: Calculated development of the velocity of the ship, collision between ferry and jacket

Figure 4-9: Deformed jacket structure during the collision ferry / jacket at t=6s
The back-calculated collision force for the scenario 2 is shown in Figure 4-10. An average value of the velocity of 0.3 s time range is used to determine the deceleration of the ship. The maximum value amounts to about 16 MN and is reached after 1 s of simulation time. The contact between ship and jacket is lost at about t = 4.5 s. After this point in time the back-calculated collision force is equal to zero.

![Collision force vs. time graph]

**Figure 4-10:** Collision force, determined indirectly via the deceleration of the ship.

The development of various energies during the simulation is depicted in Figure 4-11. At the start of the simulation the kinetic energy of the ship amounts to:

\[ E_{\text{kin}} = \frac{1}{2} m v^2 = \frac{1}{2} \times 14,200 \times 2.0^2 \text{ m}^2/\text{s}^2 = 28,400 \text{ kJ} \]

The kinetic energy decreases continuously in analogy to the drift velocity of the ship. The internal energy of the WTG and the bedding springs increases at the same time. In comparison, the internal energy of the ship, i.e. basically the energy of deformation and work of the ship, remains very low.

From this can be directly deduced that the kinetic energy of the ship is predominantly absorbed by the foundation structure and the soil. Thus, the deformations and consequently, the damages to the ship are rather insignificant.

The sum of energies slightly decreases during the simulation, as the sum does not include the energy absorbed by the potential energy of the WTG.
Figure 4-11: Development of energies during the collision ferry / jacket

Figure 4-12 depicts the stress resultants at the top of the tower. The moments around the X-axis show a maximum of approximately 48 MNm due to the impact of the ship.

Figure 4-12: Calculated stress resultants at the top of the tower, collision ferry / jacket.
The stress resulting from the maximum moment of 48 MNm shows a value below the yield stress of the material. Consequently, no collapse of the tower at its top leading to a separation of the tower and the nacelle is expected. This statement is only for information purposes and should not be taken as conclusion of this report.

\[
\sigma_{\text{max}} = \frac{M_{x,\text{max}} \cdot R}{l_z} = \frac{48 \text{ MNm} \cdot 4.0 \text{ m}}{0.495 \text{ m}^4 \cdot 2} = 194 \text{ MPa}
\]

\[
\sigma_{\text{max}} \cdot \gamma_f = 194 \text{ MPa} \cdot 1.1 = 213 \text{ MPa} < \frac{\sigma_{\text{yield}}}{\gamma_m} = \frac{355 \text{ MPa}}{1.1} = 323 \text{ MPa}
\]
### 5 Assessment of Results

#### 5.1 Applied assessment criteria

The assessment of the executed simulations for the jacket structure follows the procedure described in Section 1.5 and according to [7], Appendix 1. The given task is the classification of the investigated foundation structure into a consequence class. This is based on the results of the executed simulations following the assumptions described in Section 2 and 3. The classification is done by applying Table 5 of [7], Appendix 1. This table is here given as Table 5-1. The assessment of damages should be made in compliance with [2] according to [7].

Table 5-1: Consequences according to [7], Appendix 1, and [2].

<table>
<thead>
<tr>
<th>Qualitative</th>
<th>Offshore Wind Turbine / Ship</th>
<th>Environment</th>
<th>Safety</th>
</tr>
</thead>
<tbody>
<tr>
<td>Not significant</td>
<td>Offshore wind turbine can continue to be operated (if necessary following extensive repair)</td>
<td>No or minor environmental pollution</td>
<td>No injuries</td>
</tr>
<tr>
<td>Considerable</td>
<td>Offshore wind turbine defective</td>
<td>Considerable environmental pollution: Service products from side tanks/ double floors flow into the water (double hull and double floors not fully penetrated)</td>
<td>Few injuries</td>
</tr>
<tr>
<td>Serious</td>
<td>Offshore wind turbine destroyed</td>
<td>Major environmental pollution: Loading tanks leaking, leakage of content (double hull/ floor fully penetrated)</td>
<td>Serious injuries, small number of fatalities</td>
</tr>
<tr>
<td>Catastrophic</td>
<td>Nacelle or large parts of the nacelle crash down on the vessel</td>
<td>Ship breaks apart/sinks</td>
<td>Large number of fatalities</td>
</tr>
</tbody>
</table>

The ship hull retaining behaviour of the substructure has to be shown in the scope of the overall risk analysis according to [7]. It has to be demonstrated that no environmental pollution of larger extent occurs. The actual criterion for the classification therefore is the column "Environment" of Table 5-1.

In this table, precise information can be found in particular with regard to the evaluation of determined damages to double-hulled tankers. It is being differentiated whether only the outer or also the inner hull of the ship are penetrated. This is done against the background that the side tanks of double hulled tankers are usually filled with water ballast and, thus, do not contain any oils or fuels or any other substances which are harmful to the environment. Hence, as a consequence of a collision that will only lead to a penetration of the outer hull, no polluting substances will be emitted.

Summarising [2], [3], [4] and [11], the classification of the structure is made without considering the danger of the nacelle crashing down on the vessel.
5.2 

Assessment concerning environment

5.2.1 Evaluation of the scenario 1

The assessment follows the criteria defined in the previous section and is based on the numerical results. During the simulation, the contact of the jacket leg with the ship doesn’t lead directly to an opening of the outer hull. However, as a result of the collapse of the jacket, the transition piece and the tower of the WTG fall down on the ship. Thereby, the outer hull of the ship is strongly damaged in the superior part of the ship’s side, as shown in Figure 5-1. However, at the end of the simulation, the inner hull remains intact and no leaking of oil is expected. Thus, according to [2], the determined consequences for the environment are "not significant" in the investigated scenario double-hulled tanker / jacket.

![Figure 5-1: Effective plastic strain for the collision tanker / jacket. Bottom figure: range to 5%, elements for which the effective plastic strains reach 5% are marked red.](image)
The danger of the impact of the nacelle or other parts of the wind turbine into the ship is not regarded in this classification, cf. Section 5.1.

5.2.2 Evaluation of the scenario 2

The assessment of the scenario 2 is carried out on the same basis as for scenario 1. The damages on the hull of the ferry are not considerable. As shown on Figure 5-2, the deformed areas on the outer hull of the ship do not exceed an effective plastic strain of 5%. Only the sliding rail, which is welded on the outer hull of the ship, is strongly deformed. Furthermore, the empty space between the fuel tanks and the hull leads to the conclusion that a leakage is very unlikely for this scenario.

The small damages to the ship hull can be explained by the relatively low mass of the ship. Thus, according to [2], the determined consequences for the environment are “not significant” in the investigated scenario ferry / jacket.

Figure 5-2: Effective plastic strain for the collision ferry / jacket. Bottom figure: range to 5%, elements for which the effective plastic strains reach 5% are marked red.
The danger of the impact of the nacelle or other parts of the wind turbine into the ship is not regarded in this classification, cf. Section 5.1.

5.3 Assessment concerning ship and WTG

5.3.1 Evaluation of the scenario 1: Double-hulled tanker

The assessment of the investigated foundation structure is only based on the consequences concerning environmental impact. The classification due to the consequences for the ship and the WTG is carried out for information only.

During the simulation of the first scenario, the complete structure is destroyed. The nacelle falls down behind the ship in water. Through this, the tower falls on the deck of the ship. The damages to the ship and the WTG are respectively "not significant" and "serious". The ship can continue to be operated and requires some repair, whereas the collapsed WTG becomes out of order and probably irreparable.

5.3.2 Evaluation of the scenario 2: Ferry

As for the scenario 1, the classification due to the consequences for the ship and the WTG is carried out for information only.

During the simulation of collision of the ferry with the jacket structure, some braces of the jacket are damaged. However, the global stability of the structure remains entire. The strong vibration of the wind turbine may lead to cracks or internal damages (for example, electronic or mechanical components). For this simulation any collapse of the tower is expected. Some inspections and repairs are required for the WTG.

The hull of the ferry is deformed during the collision, but any formation of crack has been observed. The ship can continue to be operated and requires some repair.

The damages to the ship and the WTG are respectively "not significant" and "considerable".
6 Summary of Collision Analysis

The 3-leg jacket foundation is investigated for the OWF Baltic Eagle, see drawing 90267-PD-20-02. Objective of the described collision analysis is the classification of the foundation structure regarding consequences according to BSH standard "Design of Offshore Wind Turbines" [7] as well as BSH-letter dated at 30th May 2011 Ship hull-retaining configuration of substructure – design ship type for collision analysis.

The approach for the collision analyses follows the assumptions and requirements defined in [7], Appendix 1. The various model parameters, like material models, failure criteria or hydrodynamic influences, are applied according to [2]. The comparability of results to those described in [2] is therefore given. According to the requirements in [7], the assessment of damages and finally the classification of the investigated structures comply with [2].

The explicit FEM software LS-DYNA is applied for the executed collision simulations. The suitability of LS-DYNA for this kind of simulations was proven in various studies of ship-ship collisions at the TU Hamburg-Harburg, cf. [5], [9]. Furthermore, LS-DYNA is explicitly listed in [7]. However, the obtained results enable the deduction of qualitative conclusions but not of proven quantitative conclusions, because no large-scale tests were carried out to verify the numerically obtained results. The qualitative conclusions based on the executed simulations are obtained by comparison with the results described in [2].

The 3-leg jacket foundation was positioned in such a way that the assessment ship drifts laterally only against one leg of the structure. The ship hits the foundation structure with the middle of its side wall, at a water level of MSL ±0 m and a water depth of 44.3m.

The first investigated collision scenario includes the collision of a double-hulled tanker with a total mass of 137,000 t drifting sideways with a velocity of 2 m/s.

From the executed simulation the following results are found:

1. The foundation structure collapses. The tower topples on the ship. The nacelle falls into the water behind the ship.

2. The ship is slowed down by the simulated collision to 0.9 m/s after 21sec. At this point of time, the distance travelled since the first contact of the collision partners is approximately 35 m.

3. The transition piece of the jacket is pushed into the ship’s body due to the fall of the WTG. As consequence, the outer ship hull is damaged on the upper part.

4. The second hull remains intact only water ballast is emitted by the hole in the affected side tank.

Since the inner hull of the assessment ship is not damaged, any leakage of oil is expected in this scenario. Analogous to the approach in [2] the jacket is then classified as “not significant” regarding environmental impact. A possible collapse of the wind turbine or parts of the wind turbine onto the ship is not considered in this classification.
Collision double-hulled tanker / Jacket

<table>
<thead>
<tr>
<th>Category</th>
<th>Consequence</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship</td>
<td>Not significant:</td>
</tr>
<tr>
<td></td>
<td>The ship can continue to be</td>
</tr>
<tr>
<td></td>
<td>operated.</td>
</tr>
<tr>
<td>WTG</td>
<td>Serious:</td>
</tr>
<tr>
<td></td>
<td>The offshore wind turbine is</td>
</tr>
<tr>
<td></td>
<td>destroyed.</td>
</tr>
<tr>
<td>Environment</td>
<td>Not significant:</td>
</tr>
<tr>
<td></td>
<td>The outer hull is not broken.</td>
</tr>
<tr>
<td></td>
<td>No leakage occurs.</td>
</tr>
</tbody>
</table>

The risk analysis [16] determines a frequency of occurrence of $1.8 \cdot 10^{-3}$ [collisions/year] for drifting tanker with cumulative calculation for eight surrounding wind farms. This value is below $2 \cdot 10^{-3}$ [1/year], i.e. category “extremely rare” for the category environment, see Table 1-2. This leads to the risk priority number of “1” corresponding to an allowed risk according to [7].

The second investigated collision scenario includes the collision of a ferry with a total mass of 14,200 t drifting sideways with a velocity of 2 m/s.

From the executed simulation the following results are found:

1. The foundation is damaged. Few braces are buckled. The tower oscillates but doesn’t collapse.

2. The ship is completely stopped after the 2 first second of the simulation. After this time, the ship is pushed back.

3. No cracks in the ship hull are present at the end of the simulation.

Because of the low mass of the ferry, the kinetic energy of the ship is completely converted into the internal energy of the jacket structure and the soil. Furthermore, the position of the tank in the middle of the ship allows concluding that no oil leaks during the scenario 2. Analogous to the approach in [2] the jacket is classified as “not significant” regarding environmental impact.
### Collision double-hulled ferry / Jacket

<table>
<thead>
<tr>
<th>Category</th>
<th>Consequence</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship</td>
<td><strong>Not significant:</strong></td>
</tr>
<tr>
<td></td>
<td>The ship can continue to be operated.</td>
</tr>
<tr>
<td>WTG</td>
<td><strong>Considerable:</strong></td>
</tr>
<tr>
<td></td>
<td>Some braces of the jacket must be repaired or replaced. The offshore wind turbine should be inspected.</td>
</tr>
<tr>
<td>Environment</td>
<td><strong>Not significant:</strong></td>
</tr>
<tr>
<td></td>
<td>The hull is not broken. No leakage occurs.</td>
</tr>
</tbody>
</table>

The risk analysis [16] determines a frequency of occurrence of $4.4 \cdot 10^{-4}$ [collisions/year] for drifting tanker with cumulative calculation for eight surrounding wind farms. This value is below $2 \cdot 10^{-3}$ [1/year], i.e. category "extremely rare" for the category environment, see Table 1-2. This leads to the risk priority number of “1” corresponding to an allowed risk according to [7].

With these two collision analyses, the investigated 3-leg jacket structure is to be classified as collision-friendly according to the BSH-Standard construction.

Hamburg, 02 October 2012

**IMS** Ingenieurgesellschaft mbH

M. Eng. C. Rota  
Dipl.-Ing. Jörn Uecker