Dynamic Analysis of Ship Collisions with Offshore Platforms

by

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DECLARATION FOR THESIS CONTAINING PUBLISHED WORK AND/OR WORK PREPARED FOR PUBLICATION

This thesis contains published work and/or work prepared for publication, which has been co-authored. The bibliographical details of the work and where it appears in the thesis are outlined below.

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Abstract

The work performed in this thesis is concentrated on the response of offshore facilities upon ship impacts, evaluated by means of non-linear dynamic FE code LS-DYNA. The numerical studies are carried out because of high-energy ship impacts on offshore structures that, despite their low chance of occurrence, are not totally covered/predicted by hand calculations according to the simplified plastic mechanisms as suggested and commonly adopted in the current design practice. Explicit FEA are known for being widely used in collision problems and offer a good alternative to the lack of experiments (especially full-scaled), which would still be extremely costly in comparison to the computational resources demanded by such numerical analyses. However, in order to be able to carry out a significant number of numerical simulations necessary to perform parametrical studies, some simplifications still need to be assumed while trying to keep the accuracy of the FEM. Therefore, an extensive literature review is presented in the thesis to review and discuss every main aspect involved in both internal and external mechanics of the collisions and its respective preponderance on the problem’s outcome. Likewise, special care is also devoted to the FEM techniques employed in both ship and installation structures, in particular modelling the members in the contact zone and its surroundings, subjected to significant plastic deformations. It is intended that the numerical study may constitute a solid base for the solutions eventually proposed for easy application in practical engineering problems.

The numerical assessment of the collisions considered in this study concentrates primarily on jacket based steel platforms as they are the most representative platform structures among the offshore installations. Based on the principles of strain energy dissipation, the results here obtained can be extrapolated, to some extent, to other platform types.
By assuming that head-on impacts between supply vessels and offshore facilities are responsible for high amounts of impact energy, owing to higher impact speeds, the ship models are developed and calibrated based on their bow deformation, bow scantlings, mechanical properties and adopted mesh size/type. The developed bow models are compared against theoretical and experimental data from the literature to verify their accuracy. Some effort is also put in developing the most representative ship generic models with varying layouts and dimensions to reduce the number of case studies, but at the same time the majority of the plausible collision scenarios can be covered with acceptable accuracy by these models. The platform response, in turn, is evaluated locally on a first stage by taking into account the deformation behaviour of tubular members individually, and later assuming the whole installation model. While the localized response parameters with respect to the tube’s geometry, boundary conditions or axial loads are investigated in terms of the energy dissipation due to local denting or beam bending, the complete platform models, as well as the equivalent SDOF and MDOF systems are developed to predict the overall dynamic responses of the platform structures. The accuracy of the equivalent SDOF and MDOF analysis is discussed with respect to the FEM simulations.

In regards to the intensive numerical results for impact between a deformable ship bow and steel tubular members, discussions are made with respect to the accuracy of the current code of practice in offshore platform design to resist possible vessel impact.

Concerning the deck response of the installation to high energy ship impacts, the use of the proposed equivalent systems with a reduced number of DOF’s is shown to provide accurate results at significantly less computational efforts as compared to the FE simulations. The derivation of some parameters of the equivalent dynamic elastic-plastic SDOF systems however needs to account for the complexity of the analysed steel frames and perform preliminary non-linear static analyses.

Intensive numerical FE simulations are also performed with the aim of providing a clearer understanding on the strain energy dissipation phenomenon, particularly upon the ship-structure interaction that is very superficially addressed by the design codes of offshore structure design. Since plastic deformations in the installation may occur in other members than the struck legs, braces or joints, the contribution and combination
between the global and local deformation must be taken into account in the analysis for accurate predictions. The full platform models need therefore be considered. Based on numerical results, ships of different dimensions and layouts and different platform models of different sizes and configurations are categorized for different impact types. The collision cases are run for different kinetic energy amounts of the vessels and different impact orientations. Simplifications from the FEA based on the relative stiffness of the two structures are derived. A broader range on the collision assessment and design possibilities of fixed offshore steel platforms towards high-energy ship impacts is recommended.
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Joao Paulo Travanca de Oliveira

July 2014
Statement of Candidate Contribution

This thesis is composed of my original work, and contains no material previously published or written by another person except where due reference has been made in the text.

I have obtained the permission of all other author to include the published works in this thesis. I have clearly stated the contribution by others to jointly-authored works that I have included in my thesis, including experimental assistance, professional editorial advice, and any other original research work used or reported in my thesis.

Signature:  
Joao Paulo Travanca Oliveira

July 2014
Statement of Candidate Contribution
Publications Arising from This Thesis

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\( \alpha \) coefficient for deformed tube cross section geometry
\( \alpha_1 \) coefficient for work
\( \alpha_2 \) coefficient for strain energy
\( \alpha_3 \) coefficient for kinetic energy
\( \beta \) angle of deformed tube cross section geometry
\( \beta_{1,2,3} \) coefficients for normalized deformation shape
\( \gamma' \) average buoyant unit weight
\( \delta \) stiffener spacing, angle of deformed tube cross section geometry
\( \Delta \) total membrane displacement of deformed pipe section
\( \varepsilon \) strain
\( \dot{\varepsilon} \) strain rate
\( \varepsilon_{50} \) strain of 50% of the ultimate stress in a laboratory stress-strain curve for clay
\( \varepsilon_u \) ultimate strain
\( \theta \) rigid deck rotation
\( \theta_{\text{max}} \) rigid deck maximum rotation
\( \kappa \) bow equivalent linear stiffness
\( \kappa_A \) forecastle deck equivalent linear stiffness
\( \kappa_B \) bulb equivalent linear stiffness
\( \kappa_i \) elastic stiffness of platform impact point
\( \kappa_s \) equivalent linear stiffness of deformed bow portion
\( \lambda \) dimensionless energy
\( \Lambda \) dimensionless energy
\( \mu \) parameter for normalized deformation shape function
\( \nu \) Poisson’s ratio
\( \xi \) equivalent stiffness coefficient
\( \rho \) material density
\( \sigma_c \) average crushing strength of ship bow
\( \sigma_d \) static flow stress
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma_0^d$</td>
<td>dynamic flow stress</td>
</tr>
<tr>
<td>$\sigma_u$</td>
<td>ultimate strength of steel</td>
</tr>
<tr>
<td>$\sigma_{us}$</td>
<td>static ultimate stress of steel</td>
</tr>
<tr>
<td>$\sigma_y$</td>
<td>yield stress</td>
</tr>
<tr>
<td>$\phi$</td>
<td>normalized deformation shape function</td>
</tr>
<tr>
<td>$\phi_0$</td>
<td>parameter for deformed tube cross section geometry</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>angle of internal friction for clay</td>
</tr>
<tr>
<td>$\Phi_{K,L,M}$</td>
<td>stiffness, load and mass factors</td>
</tr>
<tr>
<td>$\chi$</td>
<td>scale factor</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>shaft friction parameter</td>
</tr>
<tr>
<td>$D$</td>
<td>deck horizontal displacement</td>
</tr>
<tr>
<td>$D_{\text{max}}$</td>
<td>maximum deck horizontal displacement</td>
</tr>
<tr>
<td>$a$</td>
<td>acceleration</td>
</tr>
<tr>
<td>$A$</td>
<td>cross sectional area</td>
</tr>
<tr>
<td>$A_1$</td>
<td>coefficient for force and energy curves of bow response deformation</td>
</tr>
<tr>
<td>$A_s$</td>
<td>stiffener cross sectional area</td>
</tr>
<tr>
<td>$b$</td>
<td>length of tube dented area</td>
</tr>
<tr>
<td>$B$</td>
<td>coefficient for force and energy curves of bow response deformation</td>
</tr>
<tr>
<td>$C$</td>
<td>constant for strain hardening law</td>
</tr>
<tr>
<td>$c_1, c_2$</td>
<td>coefficients for boundary conditions of cantilever beam subjected to transverse load</td>
</tr>
<tr>
<td>$C_u$</td>
<td>undrained shear strength</td>
</tr>
<tr>
<td>$D$</td>
<td>diameter</td>
</tr>
<tr>
<td>$D_m$</td>
<td>maximum width of deformed cross section</td>
</tr>
<tr>
<td>$E$</td>
<td>energy, Young modulus</td>
</tr>
<tr>
<td>$E_k$</td>
<td>kinetic energy of ship/striker</td>
</tr>
<tr>
<td>$E_{k0}$</td>
<td>initial kinetic energy of ship/striker</td>
</tr>
<tr>
<td>$E_s$</td>
<td>strain energy</td>
</tr>
<tr>
<td>$E_t$</td>
<td>tangent moduli</td>
</tr>
<tr>
<td>$f$</td>
<td>shaft friction</td>
</tr>
<tr>
<td>$F$</td>
<td>force</td>
</tr>
<tr>
<td>$F_a$</td>
<td>allowable axial compressive stress</td>
</tr>
<tr>
<td>$H$</td>
<td>folding length</td>
</tr>
<tr>
<td>$I$</td>
<td>second moment of area of the beam cross section</td>
</tr>
<tr>
<td>$I_0$</td>
<td>rotational inertia of SDOF system</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>J</td>
<td>parameter for soft clay capacity</td>
</tr>
<tr>
<td>k</td>
<td>coefficient for smeared out thickness</td>
</tr>
<tr>
<td>K</td>
<td>coefficient for tube denting</td>
</tr>
<tr>
<td>K_θ</td>
<td>rotational stiffness</td>
</tr>
<tr>
<td>K_e</td>
<td>equivalent stiffness</td>
</tr>
<tr>
<td>L</td>
<td>tube length/element length</td>
</tr>
<tr>
<td>L_1</td>
<td>length of upper span for bulb impact on frame joint</td>
</tr>
<tr>
<td>L_2</td>
<td>total length of adjacent spans for frame joint impact</td>
</tr>
<tr>
<td>l_d</td>
<td>dented region for diamond dent shape</td>
</tr>
<tr>
<td>L_{pp}</td>
<td>ship length</td>
</tr>
<tr>
<td>m</td>
<td>mass</td>
</tr>
<tr>
<td>M</td>
<td>bending moment</td>
</tr>
<tr>
<td>m</td>
<td>mass distribution function</td>
</tr>
<tr>
<td>M_0</td>
<td>plastic bending moment</td>
</tr>
<tr>
<td>M_1</td>
<td>moment at cantilever root</td>
</tr>
<tr>
<td>M_2</td>
<td>moment at cantilever free end</td>
</tr>
<tr>
<td>m_e</td>
<td>equivalent mass</td>
</tr>
<tr>
<td>M_e</td>
<td>moment of equivalent SDOF system</td>
</tr>
<tr>
<td>m_{frame}</td>
<td>mass of platform steel frame</td>
</tr>
<tr>
<td>m_p</td>
<td>plastic moment of tube wall</td>
</tr>
<tr>
<td>m_{ship}</td>
<td>ship mass</td>
</tr>
<tr>
<td>m_{top}</td>
<td>mass of platform top deck</td>
</tr>
<tr>
<td>M_u</td>
<td>plastic collapse moment</td>
</tr>
<tr>
<td>n_c</td>
<td>number of cruciforms in the bow cross section</td>
</tr>
<tr>
<td>N_p</td>
<td>lateral bearing capacity factor</td>
</tr>
<tr>
<td>N_q</td>
<td>bearing factor for clay</td>
</tr>
<tr>
<td>n_T</td>
<td>number of T-sections in the bow cross section</td>
</tr>
<tr>
<td>n_{AT}</td>
<td>number of angle and T-sections in the bow cross section</td>
</tr>
<tr>
<td>p</td>
<td>distributed impact load</td>
</tr>
<tr>
<td>P</td>
<td>collision load, coefficient for strain hardening law</td>
</tr>
<tr>
<td>P_{bow}</td>
<td>bow collision load</td>
</tr>
<tr>
<td>P_e</td>
<td>equivalent impact load</td>
</tr>
<tr>
<td>P_u</td>
<td>static collapse load for tube bending</td>
</tr>
<tr>
<td>p_{ult}</td>
<td>ultimate resistance for clay</td>
</tr>
</tbody>
</table>
**List of Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_y )</td>
<td>maximum resistance for elastic perfectly plastic material behaviour of SDOF system</td>
</tr>
<tr>
<td>( q )</td>
<td>bearing capacity factor</td>
</tr>
<tr>
<td>( r )</td>
<td>parameter for deformed tube cross section geometry</td>
</tr>
<tr>
<td>( R )</td>
<td>radius</td>
</tr>
<tr>
<td>( R_c )</td>
<td>compactness factor</td>
</tr>
<tr>
<td>( r_0 )</td>
<td>radius of deformed cross section</td>
</tr>
<tr>
<td>( s )</td>
<td>ship crushing distance</td>
</tr>
<tr>
<td>( S )</td>
<td>perimeter of cross section</td>
</tr>
<tr>
<td>( S_f )</td>
<td>ship bow frame spacing</td>
</tr>
<tr>
<td>( s_{\text{max}} )</td>
<td>bow maximum indentation</td>
</tr>
<tr>
<td>( t )</td>
<td>thickness</td>
</tr>
<tr>
<td>( t_d )</td>
<td>impact load period</td>
</tr>
<tr>
<td>( t_{\text{eq}} )</td>
<td>equivalent smeared out thickness</td>
</tr>
<tr>
<td>( t_m )</td>
<td>transition time between first and second stage of SDOF two time-step evaluation</td>
</tr>
<tr>
<td>( T_n )</td>
<td>fundamental natural period of platform</td>
</tr>
<tr>
<td>( t_p )</td>
<td>load time of triangular dynamic load peak</td>
</tr>
<tr>
<td>( T_r )</td>
<td>local permanent thickness of deformed cross section</td>
</tr>
<tr>
<td>( u )</td>
<td>lateral displacement of platform/beam struck area</td>
</tr>
<tr>
<td>( U )</td>
<td>tube global displacement in beam bending</td>
</tr>
<tr>
<td>( \dot{u} )</td>
<td>velocity of beam cross section in transverse direction</td>
</tr>
<tr>
<td>( u_0 )</td>
<td>displacement of cantilever beam free end</td>
</tr>
<tr>
<td>( \dot{u}_0 )</td>
<td>initial velocity of beam cross section in transverse direction</td>
</tr>
<tr>
<td>( u_b )</td>
<td>cross section transverse for beam buckling</td>
</tr>
<tr>
<td>( u_y )</td>
<td>plastic limit displacement of SDOF system</td>
</tr>
<tr>
<td>( v )</td>
<td>ship velocity</td>
</tr>
<tr>
<td>( v_0 )</td>
<td>ship initial velocity</td>
</tr>
<tr>
<td>( X )</td>
<td>indentation depth</td>
</tr>
<tr>
<td>( Y )</td>
<td>modal displacement</td>
</tr>
<tr>
<td>( \dot{Y} )</td>
<td>modal velocity</td>
</tr>
<tr>
<td>( y_{50} )</td>
<td>parameter for lateral soil deflection</td>
</tr>
<tr>
<td>( z )</td>
<td>beam height</td>
</tr>
<tr>
<td>( z' )</td>
<td>soil depth</td>
</tr>
<tr>
<td>( z_{\text{deck}} )</td>
<td>vertical position of platform top mass</td>
</tr>
<tr>
<td>( Z_{\text{bow}} )</td>
<td>vertical position of equivalent ship single load</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Background

In Australia, nearly 90 per cent of the petroleum wealth is found offshore. Up to 100 offshore wells per year are drilled. About a quarter of these are development wells to produce oil or gas found by previous drilling. On August 21, 2009, the accident of West Atlas rig on the Montara oil field, located around 690 km near the city of Darwin, caused millions of litres of oil to leak that created a slick as wide as 15,000 sq km. According to DMP report, Western Australia now accounts for 71 per cent of gas and 66 per cent of oil and condensate production in Australia’s petroleum industry. The Department of Mines and Petroleum, WA has committed to developing the State’s oil and gas sectors to create a sustainable and prosperous economy. The potential for ongoing exploration success in Western Australia is enormous; the State is ranked among the best areas in the world for petroleum exploration. Not only is Western Australia a low risk business environment, but there are also many highly prospective areas that are underexplored. The recently occurred mass oil spill of Deepwater Horizon was reported to be the most severe disaster to environment in the American history. The Deepwater Horizon was an ultra-deepwater, dynamically positioned, semi-submersible offshore oil drilling rig owned by Transocean. On 20 April 2010, during the final phases of drilling the exploratory well, a geyser of seawater erupted from the marine riser onto the rig, shooting 73 m into the air. The accident caused eleven workers being killed in the initial explosion, numerous workers being injured and after burning for approximately 36 hours, the Deepwater Horizon sank on 22 April 2010. The direct
economic loss is not available but "the US president says BP has agreed to establish a $100 million fund to compensate unemployed oil rig workers affected by a six-month moratorium on deepwater drilling imposed in the wake of the Gulf oil spill."", and "People and businesses seeking a lump-sum settlement from BP’s $20 billion oil spill compensation fund will most likely have to waive their right to sue not only BP, but also all the other major defendants involved with the spill, according to internal documents from the lawyers handling the fund." This is an example of enormous impact of damage of an oil platform could have on lives, environment, and economy.

Common accidents associated with offshore platforms are fire, explosion and ship impact. Despite significant consequences such accidents might impose on structures and society, as will be reviewed in this study, the design and analysis of platform structure response to such accidental loads, in particular the ship impact load, is based on oversimplified and highly idealized approaches. These simplifications may lead to inaccurate predictions of offshore platform response and thus inaccurate designs.

1.2 Research goals

This study is undertaken with the aims of:

Building and calibrating detailed real scale FE models of different ships and offshore facilities involved in different collision scenarios, in order to more accurately assess the nonlinear structural response, energy dissipation, impact force, and deformation of both the platform and ship structures under high energy impact.

Verifying the current design practice and limitation of the design codes against the numerical results obtained from this study.

Developing more accurate simplified methods or hand calculations that can be efficiently adopted in the engineering practice.
1.3 Thesis organisation

This thesis comprises seven chapters. The six chapters following the introductory chapter are arranged as follows:

Chapter 2 gives some introductory notes on the different topics related to the subject of ship-platform impact. The different structure types and respective offshore design practice are presented with emphasis on the design guidelines for ship impacts. The modelling practice is reviewed, with emphasis on the relevant FEM principles of which the later numerical analyses lie on. A brief literature review on the collision problems is also provided.

In Chapter 3 extensive parametric studies are carried out for assessing the response of steel tubular members against ship bow impacts. The numerical models are calibrated based on recent literature from other researchers. The parameters studied in the numerical simulations include the deformable bow model stiffness, the tube geometrical and mechanical properties, axial preloading, strain rate effects and boundary conditions. The results are compared with the current design practice, and additional solutions are proposed.

Chapter 4 deals with the prediction of the dynamic response of steel offshore platforms to high energy impacts from typical supply vessels. The contribution of the high modes of a cantilever beam type structure with a concentrated top mass subjected to transverse impact from rigid and deformable strikers is analysed. A procedure to develop simplified equivalent systems for efficient structural response analysis is presented and its reliability verified by comparing the results from the explicit non-linear FE simulations. Effects such as the overall rotation of the installation, plastic deformations in the contact area, different impact locations and different hinge mechanisms are taken into account.

In Chapter 5 a series of FE numerical simulations are performed with the aim of providing a clearer understanding on the strain energy dissipation phenomenon, particularly upon the ship-structure interaction. Ships of different dimensions and layouts are modelled for impact simulations. Likewise, three platform models of different sizes and configurations are considered. The collision cases involve joints,
legs, and braces are simulated for several kinetic energy amounts of the vessels and different impact orientations. An overview of the plastic deformation mechanisms that can occur in both ship and platform is also given.

In Chapter 6 a brief note regarding the influence of jacket foundations on the platform response to high-energy ship impacts is given.

Finally, concluding remarks are made in Chapter 7, along with suggestions for future work.
Chapter 2

Background

2.1 Overview

In this chapter an introduction is given to the problem of collisions between ships and offshore structures. The main features regarding the structure types, structural analysis and current design practice are covered. A literature review on the state-of-art with respect to several aspects of the collision problem, such as plastic deformation models, FEA or hydrodynamics, is also covered.

Some aspects pointed out in this chapter are laterly further reviewed.

2.2 Offshore platform types

For the purpose of analysing possible accidents involving ships and offshore facilities it is important to distinguish between the different types of structures that constitute the backbone of the offshore industry. Furthermore, the frequency distribution of ships and offshore installations according to their nature is important in estimating the likelihood of certain collision scenarios in which the design practice also lies on. One possible classification of the different offshore structures can be a loose division into two main categories based on their main functions: offshore construction and production rigs/vessels, and offshore support vessels. While the larger vessels serve as main arms in offshore construction, production, maintenance and repair of subsea pipelines and fixed platforms, the smaller vessels consist primarily of standby, supply, survey and
dive support (DSV). These will be encountered in almost daily attendance to the production installation and construction fleets. For the first category (offshore construction and production rigs/vessels), the overwhelming majority of platforms are jacket based with deck structures, followed by gravity concrete structures (North Sea and Norwegian and British sectors). The floating production units represent the third type. Besides the oil and gas industry, wind farms might also be built offshore, with an increasing number of such constructions over the last decade. Offshore installations are generally more expensive than onshore, depending on the location. For the case of wind farms, for instance, offshore towers are slightly taller than the towers on the ground as the submerged part is included, making the foundation more expensive to build. Structures located in saline atmospheres, such as offshore, also increase maintenance costs due to corrosion attacks (but not in sweet waters). Repairing and maintenance are usually more expensive in offshore environments than in land, thus encouraging operators to reduce the number of turbines for a given total power, thereby installing the biggest units available.

The water depth also introduces an extra challenge for the designer and the construction of offshore platforms. While, for instance, jack-up drilling rigs are used for water depths up to 100-120 m (for deeper water floating rigs would be used), fixed platforms are known to have been installed in water depths of 410 m in the Gulf of Mexico, with the respective jacket weighing nearly 50,000 tons. In the North Sea the Troll field is situated in approximately 305 m deep water. For deeper locations subsea wells with flowlines to a nearby (not more than 10 km distance away) fixed platform at a smaller water depth can be used. In alternative, subsea wells may be used with flexible risers to a floating production unit. These can be feasible for 300-900 m deep water. Tension leg platforms (TLP) are another type of deep water production units. These consist of a semi-submersible pontoon, tied to the vertical seabed by vertical prestressed tethers, and are used in water depths up to about 1500 m. TLP can be found, for instance, in the Gulf of Mexico or the Norwegian Snorre and Heidrun fields. From the remaining distinct types of fixed or floating platforms (Figure 2-1) and rigs there are:

- Compliant towers: narrow flexible towers on a piled foundation supporting a conventional deck for drilling and production operations, designed to sustain significant lateral deflections and forces, typically used in water depths ranging from 400 to 900 m;
Chapter 2

- Semi-submersible: having legs of sufficient buoyancy to cause the structure to float, but of weight sufficient to keep the platform upright. Generally anchored by cable anchors during drilling operations, they can also be kept in place by the use of steerable thrusters. Used in depths from 200 to 1800 m;

- Ship board rigs: active steering ships, allow for drilling operations to be conducted from a ship holding its position relative to the drilling point;

- Floating production systems: large ships equipped with processing facilities and moored to a location for a long period. These can be of the type FPSO (floating production, storage and offloading system), FSO (floating storage and offloading system) or FSU (floating storage unit);

- Seastars platforms: mini low cost TLP’s for water depths from 200 m to 1000 m, also used as utility, satellite or early production platforms for larger deep water discoveries;

- Spar platforms, moored to the seabed like TLP, but with conventional mooring lines rather than vertical tension tethers.

Such type of platforms have in common a topside structure, responsible well control, power generation, support for well-over equipment or accommodation for operating and maintenance staff, among other functions. The deck types depend upon the lifting capacity of crane vessels and the load-out capacity at the yards, and can be single integrated decks, split decks in two four-leg units, integrated decks with living quarter modules or modularized topsides consisting of module support frame (MSF) carrying a series of modules. Smaller decks can bear up to approximately 10 000 tons of weight, with the support structure consisting of trusses or portal frames with deletion of diagonals, while modularized top decks usually weigh 20 000 to 40 000 tons. For gravity structures, the topsides to be supported are in a weight range of 20 000 up to 50 000 ton.
Figure 2-1 Offshore installation types
Chapter 2

2.3 Structural analysis of offshore installations

Due to the nature of the majority of offshore platforms, the analytical models used in offshore engineering are to some extent similar to those adopted for other types of steel structures. Models comprising beam elements assembled in frames are used extensively for tubular structures (jackets) and lattice trusses (modules or decks). The members are usually rigidly connected at their ends to other elements in the model. When more accuracy is required, particularly for the assessment of natural vibration modes, local flexibility of the connections is represented by a joint stiffness matrix. Regarding their geometrical and material properties, members are additionally characterized by their hydrodynamic coefficients. For decks and hulls of floating facilities involving large bulkheads plate elements are used. Membrane stresses or plate stresses are considered when the elements undergo axial load and shear or bending and lateral pressure respectively. The joint design must also be accounted. Tubular joints are extensively used offshore, particularly for jacket structures. Connection of I-shape sections or boxed beams whether rolled or built up are basically similar to those used onshore. For a joint to be able to be fabricated and to be effective, the geometrical ratios between the brace and chord dimensions have limitations. Table 2-1 shows these limits and their typical ranges:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Typical range</th>
<th>Limitations</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \frac{D_{brace}}{D_{chord}} )</td>
<td>0.4 – 0.8</td>
<td>0.2 – 1.0</td>
</tr>
<tr>
<td>( \frac{D}{t} ) (chord)</td>
<td>12 – 20</td>
<td>10 – 30</td>
</tr>
<tr>
<td>( \frac{t_{brace}}{t_{chord}} )</td>
<td>0.3 – 0.7</td>
<td>0.2 – 1.0</td>
</tr>
<tr>
<td>Angle between brace and chord</td>
<td>40° – 90°</td>
<td>30° – 90°</td>
</tr>
</tbody>
</table>

Elements are verified for their strength (relationship between their characteristic resistance and their yield strength) and stability (relationship between their characteristic resistance in compression and their buckling limit). Tubular joints are also designed against punching under various load patterns, which may need the reinforcement of the chord by increasing thickness or using ring-stiffeners (ring plates welded in the chord prior to welding the braces to it). Fatigue, corrosion, temperature or durability might also be considered in the design when relevant.
Permanent (dead) loads, operating (live) loads, environmental loads (including earthquakes), construction (installation) loads or accidental loads, where ship impacts might be categorized, together with fire or explosion, dropped objects or unintended flooding of buoyancy tanks, represent the actions considered in the design of offshore structures that have to comply with the respective norms. The most commonly used design guides include the API-RP2A [1], the Lloyds [2] or the DnV rules [3]. Accidental loads are specified as a separate category in the NPD regulations [4], but not in API-RP2A [1] or BS6235 [5] or even the DOE-OG rules [6]. From the NPD regulations [4], accidental loads shall be considered if their probability of occurrence is at least $10^{-4}$. Regarding some statistics of collisions between vessels and offshore facilities, the variation of incident frequency with time and seasons, causation factors, incident with respect to geographical distribution, and sea conditions are discussed in [7] for a total of 394 incidents within the period 1980 to 1997. The relationship between vessel types, sizes and operations is analysed, operating circumstances, reported primary failure case, and impart orientation for different vessel types are also described in the same study. The evaluation is carried out by classifying the incidents according to their ‘Installation Damage Class’, in accordance with the following criteria:

- Severe: damage affecting the integrity of an installation sufficient to require repair in the immediate or short term (up to 1 month);
- Moderate: Damage requiring repair in the medium (up to 6 months) or longer term (over 6 months);
- Minor: Damage not affecting the integrity of the installation;
- None: no damage occurred;
- Unspecified: damage believed to have occurred but was not specified in reports;
- Not applicable: report of incident which was not applicable to installation’s structure.

The summary of the impact vessel types indicates that the vast majority of the incidents (97.97%) have occurred with the oil field offshore support vessels or attendant vessels. Of the offshore vessels, 63.71% incidents were with supply vessels followed by 16.50% with standby vessels.

The summary of mean incident frequencies of all reported incidents and ‘moderate’ or ‘severe’ damage category incidents is shown in Table 2-2:
Table 2-2 The summary of mean frequencies of reported incidents

<table>
<thead>
<tr>
<th>Installation/rig type</th>
<th>All reported incidents/year</th>
<th>Incidents resulting in ‘moderate’ or ‘severe’ damage/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fixed installations</td>
<td>0.0945</td>
<td>0.0121</td>
</tr>
<tr>
<td>Floating installations</td>
<td>0.2290</td>
<td>0.0504</td>
</tr>
<tr>
<td>Jack-ups</td>
<td>0.2220</td>
<td>0.0075</td>
</tr>
<tr>
<td>All installations</td>
<td>0.1319</td>
<td>0.0191</td>
</tr>
</tbody>
</table>

In summary, offshore structure design codes are currently characterized by:
- Design criteria formulated in terms of limit states;
- Semi-probabilistic methods for ultimate strength design which have been calibrated by reliability or risk analysis methodology;
- Fatigue design checks depending upon consequences of failure and access for inspection;
- Explicit accidental collapse design criteria to achieve damage-tolerance for the system;
- Considerations of loads that include payload, wave, current, wind and ice, earthquake loads, as well as accidental loads such as fires, explosions and ship impacts;
- Global and structural analysis by finite element methods for ultimate strength and fatigue design;
- Nonlinear analysis to demonstrate damage tolerance in view of inspection planning and progressive failure due to accidental damage;

2.4 Platform foundations

The majority of offshore platforms are of the fixed type. It is also of concern to account for the characterization of the respective oil and gas fields’ seabed. The properties of the soil depend mainly on factors such as the density, the water content or the over consolidation ratio. In terms of design purposes, the influence of these factors on the soil behaviour is expressed by two parameters that are the friction angle and the undrained shear strength $C_u$. Soils are therefore classified either as granular (sands, silts) or cohesive (clay). The nature and characteristics of the soil surrounding a pile generally vary with the depth. The lateral soil resistance against the deflection is represented by P-y curves that vary with the depth and type of soil at the considered elevation. The
general shape of the curves for increasing displacement features the linear behaviour for small deflections, the linear/plastic behaviour for medium deflections and the constant resistance for large deflections or loss of the resistance when the initial soil configuration deteriorates (in particular for clays under cyclic loadings).

For analysis purposes, the soil is divided into several layers [1], each having constant properties throughout. The number of layers will always depend on the precision required for the analysis. Steel offshore platforms are normally founded on piles that are driven into the soil. These transfer the loads acting on the jacket into the sea bed.

There are basically three types of pile jacket arrangement (Figure 2-2):

- Pile-through-leg, where the pile is installed in the corner legs of the jacket;
- Skirt piles through pile sleeves at the jacket base, where the pile is installed in guides attached to the jacket leg. The skirt piles can be grouped in clusters around each of the jacket legs;
- Vertical skirt piles, that are directly installed in the pile sleeve at the jacket base. This has the advantage of having reduced structural weight and easier pile driving, whereas the inclined piles enlarge the foundation at the bottom providing a stiffer structure.
2.5 Dynamic behaviour of platforms

The majority of structural analyses are based on the linear theory of elasticity for total system behaviour. The dynamic amplification factors for time-dependent loads are generally accounted for rigid structures having a fundamental vibration period typically less than 3 s. For wave-attack, for instance, if the natural period exceeds 3 seconds many elements can exhibit local dynamic behaviour, such as compressor foundations, flare-stacks, crane pedestals, slender jacket members or conductors.

The dynamic models of the platform are normally derived from the main static model, although some simplifications may be assumed, e.g. lumped masses at the member ends; foundation models derived from the cyclic soil behaviour.

The mass of the structures include, besides the structure itself, the appurtenances, entrapped water in legs, the mass of marine growth and the added mass (mass of water displaced by the member and determined from potential flow theory). Lumped mass
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points are generally assumed for the dynamic models. Therefore the mass matrix becomes diagonal but local modes of vibration of single members are ignored. While the stiffness matrix is in all aspects similar to the one used in static analyses, the damping is in turn the most difficult to estimate among all parameters governing the dynamic response of the structure, which may consist of structural damping (loss of energy by friction in the material) or hydrodynamic damping. The damping is commonly represented as viscous damping, taken as either modal (0.5% for structural and 1.5% for hydrodynamic) or as a linear combination of stiffness and mass matrices.

2.6 Platform modelling against impact loads

Impact loading can be responsible for elastic or non-linear responses of the platform, which can be also mainly overall or governed by local effects. Simplified analytical methods or the model discretization into finite elements can be used for the prediction of the frame structure responses to impact loads.

Material can be modelled either as elastic-perfectly-plastic behaviour or gradual plastification strain hardening characteristics. Collision analysis methods are described in [8] for the elastic-plastic behaviour of offshore steel structures under impact loads, where a non-linear force-displacement relationship is used for the simulation of the local indentation on a hit tubular member and 3D beam-column elements are used for the modelling of the global behaviour of the struck structure. The elastic large displacement analysis theory and the plastic node method are combined in order to describe the effects of large deformation, plasticity and strain-hardening of the beam-column members. Strain-hardening effects are shown to play an important role for the impact response, resulting in smaller deformations and more energy absorbed by the striking structure, which means a bigger impact force. The advantage of the method is that the number of elements in the analysis model does not need to be larger than the number of elements used in normal linear analysis procedures.

Computer programs such as USFOS [9] that make use of two-node beam as the basic structural unit can be used to model an entire structural member and consequently large structural systems by means of a relatively small number of elements. Modelling of damaged members generally includes:

- Lateral distortion of tube axis
The dented section is idealized as shown in Figure 2-3. The cross section consists of a dented part and an undamaged part. The load shared by the dented part is assumed to be limited by the force causing yielding at the middle of the dent. Further loading is carried by the undamaged part alone. The total capacity of the cross section is dependent on the plastic interaction between axial and bending moment, according to the dent depth and orientation. The effects of local flexibility of tubular joints are also included.

Algorithms for ship impact analysis shall normally account for the local deformation of the tube wall at the point of impact, the beam deformation of the hit member and the global deformation of the platform. The effects of external hydrostatic pressure on the plastic capacities of tubular sections may also be accounted for. Parts of a structural system that behave completely elastically may be efficiently modelled by use of reduced super element stiffness matrices with a user-specified number of nodes. Non-linear time-domain dynamic analysis may be performed with special reference to the dynamic ship collision problem.

With the advancement in computer and computational technologies, finite element codes have gained popularity owing to the possibility of achieving a greater detail in structural modelling and consequent greater accuracy. The tubular members that represent the overwhelming majority of the structural elements of steel offshore facilities are generally treated as thin-walled since for their diameter $D$ and their thickness $t$, where $D \gg t$. For structural elements in which two dimensions are much greater than the third one and when the change of the analysed feature across this third
direction can be neglected shell elements are used. The results from the simple analysis in [10] suggest that the shell element model yields reliable predictions when the third dimension is at least 20 times smaller than the other ones. However, if the change of the analysed feature is on a comparable level in all directions of the analysed element, solid elements should be used. If there is no feature in the model through the thickness then modelling the structure by shell elements is preferred (CPU, problem size). Mindlin-Reisner [11] theory is needed for thicker shells and Kirchoff [12] theory will suffice for a much thinner shell (this is related to the rotational degree of freedom).

The advantages of the use of shell elements result mainly from time-saving due to reduced number of degrees of freedom (and consequently the equations to solve). Some other practical differences between shell and solid elements concerning FEM can be listed as follows:

1. Geometrical model
   - solid element model is often available, while shell element has to be created;
   - problems in precise connection of the surfaces for shell element model;
   - problems in proper connection (or contact) of multi-layered surfaces;

2. Mesh
   - shell mesh much easier to create, if good quality elements is needed (box instead of tetrahedron elements in solid);
   - more realistic boundary conditions for solid element model (faces instead of edges);

3. Analysis
   - less problems with stability using shell elements;
   - shell elements need much less disk space for storage, which is important for nonlinear/nonstationary analysis and big models;
   - post-processing of shell element models is faster (again for big model).

Codes such as ABAQUS [13] also offer the possibility of using continuous shell elements, which geometry (topology) of the element is like solid, but equations are formed up with shell elements.
The choice between shell and solid element will also depend on the loading conditions. If the loading is the shell plane (tangent to the surface) the shell will respond under a membrane effect (only two DOF are excited) and the 2D plane-stress element could be used or a 3D solid with only one layer through the thickness. If the loading is normal to the shell surface, here the shell element is preferred, and depending on the thickness and curvature, Kirchoff [12] or Mindlin [11] element could be used.

The element formulation is more complex. One might use the 3D solid element to model the bending, but here at least 3 layers of 3D solids are needed. As for the mesh refinement, even if element size is smaller than its thickness results can be correct. The shell element applicability does not depend on ratio \( L/t \), but on ratio \( R/t \) (where \( L \) – element length, \( t \) – element thickness and \( R \) – curvature radius of surface). For constant \( R \) and \( t \) with constant ratio \( R/t \), refining the mesh is possible even for quite small \( L \). When \( R/t \) is high enough, refinement can be done. A practical case is illustrated in Figures 2-4 and 2-5 for two clamped tubes under lateral impact of a rigid solid block of 1 m width, analysed using the LS-DYNA code [14, 15]. For a given diameter \( D \) of 1.5 m and thickness \( t \) of 30 mm the tubes are discretised using both shell and solid (one layer) elements. The assumed material properties of the steel tubes are provided in Table 2-3.

Table 2-3 Mechanical properties of tubes under lateral impact

<table>
<thead>
<tr>
<th>E</th>
<th>( \sigma_y )</th>
<th>( \nu )</th>
</tr>
</thead>
<tbody>
<tr>
<td>[GPa]</td>
<td>[Mpa]</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>250</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Figure 2-4 Tube deformation under lateral transverse load obtained with different element types
From the deformed shapes upon lateral loading, as well as from the load-deformation curves, there are clear differences between the using of the two element types, confirming the need of a much higher number of solid elements in order to obtain more accurate results.

2.7 Collision mechanics

The analysis of ship impacts on offshore structures follows the same principles of the ship-ship collisions, apart from the impact scenarios that differ in nature.

The impact analysis is normally determined by splitting the problem into two uncoupled analyses: the outer collision dynamics, which deals with the motion of the colliding ships and their interaction of the surrounding water, and the inner collision dynamics, which involves the crushing of the material in the bow of the striking ship and the side of the struck ship. The theory for absolutely inelastic collisions between two bodies resultant from central impacts are usually assumed. For the assessment of ship-platform collisions it is also important that historical data be collected, as well as risk analysis models that be necessary to predict other types of impacts, such as incidents caused by trading vessels. The way how the whole process is addressed can be schematically described in Figure 2-6.
2.7.1 Outer dynamics

The ship collision action is, as mentioned above, characterized by a kinetic energy governed by the mass of the ship, including hydrodynamic added mass, and the speed of the ship at the instant of impact. The amount of water dragged will depend upon the ship motions at the impact. The motion of a ship at sea has six degrees of freedom if
neglecting any deformation of the ship structure. They are three translational (heave, sway and surge) and three rotational (pitch, roll and yaw), as illustrated in Figure 2-7:

![Ship motions](image.png)

**Figure 2-7 Ship motions**

Modelling the interaction between the vessel and the surrounding water can be time consuming and computationally intensive. Alternative methods [16] for the inclusion of the buoyancy and vertical motions suggest the use of springs which stiffness is given by the water density times the tributary area of the ship node to which the spring element is linked. The force in each spring would correspond to the buoyance acting on the area tributary to the node. However, during a ship collision, the most significant motion of the ship will be in the water plane. Therefore, the equations will be derived under the assumption that there is no coupling between the motion in the water plane and the other motions of the ship. Structural vibrations of the ship can thus be neglected.

The designer normally takes the added mass as a constant value added to the ship’s mass ([17], [18] and [19] among others). It is assumed that all the kinetic energy is absorbed, such that during the slowing down process no other modes of motion develop than the one present. This means that all the kinetic energy is supposed to be absorbed when the plastic deformations have reached their maximum. However, for cases in which yawing starts to develop during the slowing down process, the kinetic energy is never to reach zero at the point of maximum deflection. Not much information is known with respect to the added mass being split up into, for instance, a coefficient pertaining to sway and another to yaw (sideway impact) in a hydrodynamically sound way. The added mass coefficients evaluated from experimental and analytical results, considering
Chapter 2

eccentric impacts [20] are shown to be higher than the usually assumed values ranging between 1.3 (30%) and 2.0 (100%).

The coefficients and damping are evaluated in [21] and [22] according to the vessel motion, velocity and wave oscillation. In [22] coupling coefficients are also estimated and a case of a 200,000 DWT tanker in shallow waters is studied.

In fact, the designer most of the times ends up taking a coefficient which has little if nothing to do with the subject case. Any dependency of the added mass coefficient on vessel size and shape, underkeel clearance, fender stiffness and characteristic and location of the point of contact on the ship’s hull is either not rightly understood or considered only marginal importance.

For the centric impact case the added mass coefficients can be obtained through the equation of momentum or the energy absorption. However, as already mentioned, the constant added mass coefficients derived from the both techniques are not consistent with each other [20]. The added mass coefficient derived from the energy equation is somewhat smaller than the one from the momentum equation.

Special emphasis is laid on a consistent treatment of the hydrodynamic pressures on the ship hull and the overall flexibility of the platform structure involved, and the hydrodynamic forces acting on the ship hull during the collision are calculated by means of unit response functions (determined by cosine transformations of sectional dampings) in [23]. It is shown that if the traditional concept of a constant added mass is to be used, then this constant will depend on the duration of impact and the form of the ship. It is also shown that in the case of a relatively long duration of the collision, the traditionally used added mass in the expression for the kinetic energy of the ship striking the platform should be chosen as a larger value than the normally prescribed. Of course, the same variation is found for the added mass coefficient of the platform. However, most floating platforms have relatively small water plane areas and, therefore, small variation of the hydrodynamic coefficients with frequency. Also, for the most interesting cases where a supply vessel strikes a floating platform, the mass of the platform will normally be much greater than the ship mass. In such situations, even great variations of the added-mass of the platforms have a small effect on the amount of energy released for deformation. Thus, for evaluation of the collision damage, the
hydrodynamic pressure of the hull of the ship will in most cases play the most important role. For steel jackets, the variation of added mass with frequency can actually be neglected.

The values most frequently used for the ship added mass are:

- 10% for forward motion, and
- 40% for sideway motion.

The value for sideway motion was first introduced by Minorsky [24] for collisions between two ships, where the linear dependence between the deformation energy and a collision resistance factor approximately equal to the crushed volume was demonstrated. In [25] it was shown that the added mass coefficients depend both on the duration of the collision and the relation between collision force and deformation, implying a simultaneous interaction between the outer and the inner collision dynamics. The hydrodynamic forces were obtained in [25] through a strip method under the condition that the ship is slender (the hydrodynamic forces related to the surge motion cannot be found by strip methods, but they are neglected as the surge is very small when compared to the sway force). In [26], owing to the fact that the lowest natural period for longitudinal vibrations of the ship is much smaller than the duration of a normal collision impulse, the amount of energy which goes into longitudinal vibration can be assumed to be negligible. For sway or yaw motions of the ship into an object, horizontal bending and torsional vibrations are excited. The added mass value of 40% of the ship’s total mass is therefore a reasonable approximation only when the duration of the impact is very short, i.e. less than 0.5–1s. For longer collision duration, the added mass can reach 100% [25], [27].

For small collision forces, all the kinetic energy of the ship is released for local plastic deformation. The added-mass coefficient that is to be used to determine this energy is approximately the added-mass for zero frequency. For head-on collisions, the lowest natural period for longitudinal vibrations of the ship is much smaller than the duration of the collision impulse and so the amount of energy going into the longitudinal vibration can be assumed to be negligible.

Since it is likely that colliding vessels will sway when bow impacts are eccentric, it is then important that during the modelling process the ships are given the correct physical
features, in particular the gyradii that is affected by, for instance, the ship’s weight distribution.

The longitudinal weight distribution is vital to the calculation of the longitudinal strength of the ship, but also affects the speed loss in a seaway [28]. Weight distributions of the three principal axes can be also used in calculating the gyradii of the ship [29] which has significant effects on the seakeeping ship’s performance. Traditionally, a stock approximation appropriate for the ship type is used and improved by distributing the large weight items separately. With the advancement of the computational and modelling techniques, the determination of a ship’s weight distribution has become less laborious giving rise to grouping methods. Weight data is generally stored in databases as large numbers of discrete details that are essentially lumped masses and can represent items which extend for large portions of the length of the vessel. Approximation methods are usually based on combinations of a mid-ship rectangle with forward and after trapezoids. More sophisticated methods are based on modelling a portion of the weight curve from the ship’s buoyancy curve. Some of the approximation methods (Figure 2-8) can be found in [30], [31], [32] and [33]. The longitudinal weight distributions by type for various military and support vessels are referred in [34].

![Approximation per Comstock](image)

As for the grouping methods, these consist of placing the weight details in buckets based on the location of their longitudinal centre of gravity [35] (Figure 2-9). Direct distribution method is another approach through which the individual weight records are distributed individually [36]. The fundamental representative shape of direct
distribution methods is the trapezoid. Representing a weight record such as trapezoid requires that the weight, longitudinal extents and longitudinal centre of the weight being represented are known.

![Figure 2-9 An illustration of the bucket method](image)

### 2.7.2 Inner dynamics

The different collision scenarios are established by considering bow, stern and side impacts on the structure as appropriate. The most probable collision orientation for supply vessels is stern-on for both fixed and mobile installations, accounting for about 50% of all known cases, and about 20% of the incidents where the orientation was known have occurred with sideways-on collision [37]. However, since ships higher speeds are associated with cruising, bow impacts shall be responsible for the highest energy collisions.

The approaches in analysing ship collisions can be usually categorized into experiments, numerical simulations or simplified analysis methods. All methods can be applied from the local element level to the level of small-scale ship substructures. At the global structure level, experimental data are scarce and limited to medium-scale structures. Most collision scenarios are based on rigid bows striking the side structure of a deformable ship [38], [39]. Tests with deformable bows colliding into a rigid structure were also reported in [40], [41], and ship collisions with two deformable bodies, the bow and the side ship, were tested in [42]. For one-to-one scale model of ship collisions, numerical simulations were conducted in [43], [44] and [45]. The interaction between the deformation on the striking and the struck ships is also analysed by means of explicit non-linear FE in [46], and the FE simulation results were used to validate the simplified analytical methods as experimental data is not available.
For analysing ship structures, the structure is divided into several basic elements. The resistance of each of the basic element is evaluated, and all the resistances are added to obtain the total response of the entire structure. Ship structures are mainly characterized by plate girder intersections and shell plating. For instance, when determining the energy absorption of the side structure of the ship in a collision between a ship and the leg of an offshore platform it is necessary to take into account both the energy absorption of the side-plate and of the webs, decks, bulkheads, etc. Some methods which appear to be suitable for this are developed in [47] and [48].

The plate girder intersections can be categorized into L-sections [49], T-sections [40], [50] or cruciform [51], [52] (Figure 2-10). The fourth type is the web girder that has no intersections and is fixed at the both ends. Various researchers have assumed different folding patterns to derive the internal energy, resulting in different mean crushing forces. The resistance of the elements is obtained by equating the external and internal rates of energy dissipation.

A series of bow collision tests containing six specimens can be found in [40]. The first five specimens consists of two model types with transverse frames, one box-shaped and the other wedge-shaped. Three of these include deck and have a varying frame system. The last model is a simplified model of the upper part of a supply vessel bow. The elements are assembled by continuous double fillet welds except in the front section where space availability permits a single weld only (Figure 2-11).
As for the bow type, according to statistical data [53], 40-50% of merchant vessels, for instance, have bulbous bows and the others have conventional bows. It is clear that bulbous bows produce higher crushing loads than the conventional ones due to their greater stiffness increased by the bulb. It should be noted that the above studies assumed rigid platform structures in predicting the impact force and bow structure deformations. The bulbous bow can be divided into two parts, one is the upper part which is similar to the conventional bow, and the other is the lower part-bulb, as schematically described in Figure 2-12.
For offshore facilities exposed to accidental loads, the response will be dominated by plastic straining, apart from frame deformation of the platform where elastic straining can be significant. The deformations that contribute to the dissipation of energy include local denting of the hit cross-section, beam deformation of the hit member, frame deformation of platform and deformation of the ship. The relative contribution from each of the modes varies with the impact scenario considered. This will be discussed in more detail in the following chapters.

### 2.8 Collision modelling

Besides ship-ship collision or identical problems dealing with ship impacts with floating offshore-installations, the study of ship impacts with bridge structures has also often been addressed in the literature, having all of these issues of similar nature. For instance, in alternative to the equivalent static forces used for determination of the structural demand of bridge structures [54], [55], [56], or to non-linear finite-element methods to conduct the analyses of vessel-bridge collisions [57], [58], simplified models have been suggested for practical engineering purposes. The ship deformable bow is usually simplified into a SDOF equivalent system, and the bridge reduced to a deformable mass beam [59] (Figure 2-13). The last assumption is shown to be unreasonable in [60] and [61]. In [62] the simplified model proposes a MDOF for the bridge structures and in [63] the same model is refined by considering the influence of the strain rate effects, which depend on the impact speed.

![Figure 2-13 Simplified interaction model for barge impact with bridge piers](image)
As for the more advanced modelling involving FE codes, two types of methodologies are normally assumed: implicit methodologies achieve the solution by simultaneously solving systems of equations needing frequent updating of the stiffness matrix for non-linear FE analysis, thus demanding computer capacity especially in terms of memory resources and CPU cycles; the explicit methodologies, in turn, can obtain system solutions based on mass matrix that remain constant not needing therefore frequent updating. However, explicit codes need smaller time steps to comply with stability requirement for equation solving. The calculation efforts might then be less for explicit method and convergence of calculations is much easier to realize.

Explicit FEA have been extensively used in collision problems [46], [64], [65] among others. For ship-ship collisions, uncertainties of input parameters in the FE simulations and their impact on the shape and size of the damage opening area, and time to capsize of the struck ship are addressed in [64]. The material modelling aspects and the effects of using rigid or deformable striking bow sections are also considered, as well as the friction coefficient, the collision angle and the impact speed.

The most significant uncertainties involving the overall circumstances around the collision, i.e., from a probabilistic point of view, are studied in [66], [67] and [68]. The external dynamics evaluating the ship motions, giving as a result the energy to be absorbed by structural deformations, has been studied, for instance, in [69] and [70]. Examples of the analysis of failure and fracture of large stiffened shell structures by means of FEA can be found in [71], [72], [73], [74], [75] and [76].

A de-coupled numerical procedure combining non-linear explicit FE analysis of the collision event followed by dynamic damage stability and survivability analysis of the struck structure (vessel) is presented in [77].

The effects of the friction during the impact are also addressed for collisions between vessels in [64]. The dynamic friction coefficient is commonly used as 0.3 in collision ship analyses [75], although values of 0.6 have also been considered [76]. The effects of friction for tanker collision have also been studied for values ranging from 0-0.7 in [77]. Engineering handbooks [78] suggest coefficients of 0.57 for non-lubricated mild steel against mild steel and 0.09 – 0.19 for lubricated surfaces. If in real collisions the
surfaces are wet and below the waterline of the ships and in ballast tanks, the steel plates used on ships, on the other hand, have generally a rather rough surface, making the real scatter in friction coefficients for ship collision events likely to lie between 0.1–0.6. In [64] it is concluded that the friction coefficient in contact conditions has minor influence on the outcome for the analyses. Other recommendations from the same study concerning the respective ship collision analyses cases include the modelling of the striking bow section and the collision angle and striking speed. While the use of rigid bow sections is very crude and should only be recommended in comparative studies, the collision angle has no statistical significance for the geometries considered in the study.

The DnV rules are also examined by means of FE in [79] for side impact of a supply vessel against fixed jacket steel platforms. Independent analyses are carried out for the vessel modelled either as rigid or ductile. The DnV rules [3] are shown to underestimate the resistance for a certain indentation due to the inaccurate description of the column deformation mode. The importance of the elastic response in the dynamic analysis is also demonstrated through its contribution in reducing the impact loads and local energy dissipation.

The numerical simulations primarily performed in the current study are carried out also using the explicit FE code LS-DYNA [14, 15]. A certain level of detail is, on a first stage, aimed in order to provide reliable/accurate results. The adopted modelling procedure is schematized in Figure 2-14:
2.9 Conclusion

The topics discussed in this chapter provide the basic concepts and necessary background for a better comprehension of the work developed in this thesis. The structure of the next chapters as well as the majority of assumptions was taken following the concepts and techniques here presented.
Chapter 3

Numerical Analysis of Steel Tubular Member Response to Ship Bow Impacts

3.1 Introduction

Current risk analysis of jacket installations is very limited when it comes to jacket legs being head-on impacted by vessels. As a result bow forces are usually estimated by assuming rigid offshore structures although it is commonly agreed that this simplification may not lead to accurate predictions of the contact force between ship and platform as the structural deformation of the platform also absorbs a significant amount of impact energy. On the other hand, deformations in the platform members are commonly predicted under the assumption that the ship hulls are strong enough to be treated as rigid. Offshore platforms are in most of the cases constituted by tubular steel members. In this chapter, the response of tubular members to ship bow impacts is analysed by means of FEM calculations. The study focuses on the analyses and predictions of damage pattern and damage intensity that can be provoked on jacket legs by ship impact as the collapse of such members would affect the integrity of the platform. A detailed finite element model is developed for impact analysis of merchant vessel bows against tubular members, representative of offshore jacket legs. The model comprises a general supply vessel in the range of 2000 ton to 5000 ton displacement and a vertical steel pipe representative of a jacket leg. Nonlinear inelastic responses of both the ship and tubular structures are considered. The numerical results are checked against tests by other authors to verify the accuracy of the model. The verified model is used to
perform parametric simulations. Different geometrical parameters such as member length, wall thickness and diameter of the tubular members, as well as the boundary conditions, axial preloading and dynamic aspects such as the impact velocity and the strain rate effects are considered in the analyses to examine the performance of the platform under vessel impact. Based on intensive numerical results, discussions are made with respect to the accuracy of the current code of practice in offshore platform design to resist possible vessel impact.

3.2 Background

The effects of a ship impact can result in several scenarios regarding the integrity of the platform. These can range from moderate to catastrophic, depending on the ship involved, its velocity, as well as the stiffness and strength of the platform and the location of impact. For an accurate prediction of the performance of a platform under ship impact, the relative strength of both ship and platform structures should be accounted for when analysing how the kinetic energy of the system is converted into strain energy.

The classical way the collision problem has been being approached is through quasi-static analyses, where the collision effects are assessed by following the laws of conservation of momentum and conservation of energy. In most previous studies, to simplify the problem, many researchers consider only one impact structure is deformable and assume the other one is rigid. As a result, the impact loads are related to only the damage of the ‘deformable’ structure. However, it is commonly agreed that neglecting the deformation and damage of the other structure, either the ship or the platform, which may absorb significant amount of impact energy and hence affects the ship-platform structure interaction, may lead to inaccurate predictions of impact loads and structural responses.

Since head-on collisions from a ship can represent the most harmful cases for the integrity of the steel platform, it is of interest to assess with a certain level of detail how platforms are locally affected, and more precisely the hit tubular members by ship bows by considering deformation and damage of both ship bow and platform structure. The results will be used to check the accuracy and scope of the current design practice developed based primarily on the assumption that either ship or platform structure is
rigid, and lead to better understanding of ship-platform structure interactions upon ship impacts.

Often the effects of ship collisions on offshore platform structures are determined by impact energy together with simplified methods of structural analysis. The Norwegian Standard classifies the design principles according to the amount of strain energy dissipated through each of the structures, where the dissipated energy is a function of the relative strength between installation and ship.

### 3.2.1 Bow deformation forces

The quantification of the force that can be exerted by a ship bow with an initial velocity on a platform structure is usually made by assuming the ship crushing on a rigid wall that could be representative of relatively stiff bridge piers, large diameter columns or other ships or floating platforms insusceptible of any large deformation and yield. Minorsky [24] estimated the relationship between the deformed steel volume and the absorbed impact energy based on investigation of various ship-ship collisions. Ship types can vary in size, shape and structure. Curves derived from numerical study performed by Pedersen et al. [79] on ship impacts on rigid bridge pier or offshore platform estimate, for ships varying between 500 DWT coasters and 150 000 DWT bulk vessels, that crushing loads can reach 700 MN with initial speeds of up to 9.3 m/s. Defining a model to be representative for all the ship types and that could come across a jacket platform is not possible. Therefore, sorting ships with a certain degree of structural similarity into the same category would reduce the number of different models needed in numerical works.

Among the authors who have developed methods to analyse bow impacts, the method of Amdahl [81] is discussed in detail here. Amdahl correlated model test results with theoretical considerations. The energy which is dissipated during the plastic deformation of structural elements such as angles, T-sections and cruciforms is considered (see Figure 3-1).
The average crushing strength according to Amdahl’s procedure is given by:

$$\sigma_c = 2.42 \cdot \sigma_u \cdot \left[ \frac{n_{AT} \cdot t^2}{A} \right]^{0.67} \cdot \left[ 0.87 + 1.27 \frac{n_c + 0.31 \cdot n_T}{n_{AT}} \left( \frac{A}{(n_T + 0.31 \cdot n_T) \cdot t^2} \right)^{0.25} \right]^{0.67}$$  \hspace{1cm} (3-1)

and the total crushing load is obtained by multiplying $\sigma_c$ by the associated cross-sectional area of the deformed steel material, $A$. Here, $\sigma_c$ is the average crushing strength of bow, $\sigma_u$ the ultimate strength of steel (including strain rate effects), $t$ the average thickness of the cross-section under consideration, $A$ the cross-sectional area of the deformed steel material, $n_c$ the number of cruciforms in the cross-section, $n_T$ the number of T-sections in the cross-section and $n_{AT}$ the number of angle and T-sections in the cross section.

The magnitude of the dynamic flow stress $\sigma_u$ [82] is calculated as follows:

$$\sigma_u(\dot{\varepsilon}) = 1.29 \cdot \sigma_{us} \cdot \dot{\varepsilon}^{0.037}$$ \hspace{1cm} (3-2)

where $\sigma_{us}$ is the static ultimate stress of the steel material; and the strain rate is taken as:
\[ \dot{\varepsilon} = \frac{v_x}{S_f} \quad (3-3) \]

where \( v_x \) is the velocity in longitudinal direction during impact and \( S_f \) the frame spacing.

This model proposed by Amdahl makes use of Wierzbicki’s folding mechanisms [83]. Yang and Caldwell [50] assumed a similar way of energy dissipation during deformation of the structure. It differs from Amdahl’s that the longitudinal stiffeners may be included through an equivalent thickness of the shell plating, equalizing the plastic bending moment of the equivalent plating and the plastic bending moment of the shell plating with longitudinal stiffeners. Furthermore, while Amdahl determines the folding length and crushing load by minimizing the deformation energy absorbed during the folding process, Yang and Caldwell take the folding length \( H \) the same as the space between the transverse frames, provided that the frame spacing is less than the theoretical folding length.

The numerical predictions by Amdahl and Yang and Caldwell are compared in [80], who derives his expression based on six different ships, ranging from a small coaster to a large bulk carrier. To estimate the maximum bow collision load, Pedersen considers the influences of the bow vessel size, loading condition, speed and strain rate effects. The expression becomes:

\[
P_{\text{bow}} = \begin{cases} 
P_0 \cdot \frac{E_{\text{bow}}}{E_{\text{imp}}} \left( \frac{5.0}{L} \right)^{1.6} & \text{if } L \leq L_{pp} / 2.75 \\
2.24 \cdot P_0 \left[ \frac{E_{\text{bow}}}{E_{\text{imp}} \cdot 1425} \right]^{0.5} & \text{if } L > L_{pp} / 2.75 
\end{cases}
\quad (3-4)
\]

where

- \( L = L_{pp} / 275 \) [m];
- \( E_{\text{bow}} = E_{\text{bow}} / 1425 \) [MJ];
- \( E_{\text{bow}} = \frac{1}{2} \cdot 1.05 \cdot m_{\text{ship}} \cdot v_0^2 \) [MJ].

and \( P_{\text{bow}} \) is the maximum bow collision load, \( P_0 \) the reference collision load equal to 210 MN, \( E_{\text{imp}} \) energy to be absorbed by plastic deformations of the bow, \( L_{pp} \) the length of the vessel (m), \( m_{\text{ship}} \) the total mass \( (10^6 \text{ kg}) \) with respect to the longitudinal motion and \( v_0 \) is the initial speed of the vessel \( (\text{ms}^{-1}) \).

With respect to the bow deformation, the maximum indentation \( s_{\text{max}} \) is estimated from the equation of motion by approximating the load-indentation curves into sinusoidal curves:

3-5
and the associated impact duration derived from the equation of motion is estimated by

\[ td \approx 1.67 \cdot \frac{s_{\text{max}}}{v_0} \]  

(3-6)

### 3.2.2 Deformation of tubular members subjected to lateral loads

Offshore platform structures are often made of tubular members and tubular members often have very thin walls when compared to the diameter. Therefore their deformations under ship impact, which will contribute to energy dissipation, are often significant and cannot be neglected. In previous studies, the tubular members are usually treated as thin-walled structures, allowing simplified equations to be employed to predict their responses in terms of the cross-sectional area, section modulus or plastic moment. The behaviour of tubular members under lateral loads has been studied either via numerical, experimental or analytical approaches. Furnes and Amdahl [84] defined the relationship between the indenting force and the depth of penetration and energy as:

\[ P = 15 \cdot m_p \left( \frac{D}{t} \right)^{1/2} \left( \frac{2X}{D} \right)^{1/2} \]  

(3-7)

\[ E_s = 14 \cdot m_p \frac{X^{1.5}}{\sqrt{t}} \]  

(3-8)

where \( m_p \) is the plastic moment of the tube wall \( (m_p = [t^2 \sigma_y]/4) \), \( D \) the tube diameter, \( t \) the thickness of the tube wall, \( X \) the indentation depth and \( \sigma_y \) the yield stress. Figure 3-2 describes the approximated deformed surface caused by ship sideway impacts, where the dent differs from that of Figure 2-3 in chapter 2.

Figure 3-2 Plastic mechanism for sideway impact by supply vessel [84] – \( b \) defines the length of the flattened area in contact with the indenter.
A series of experiments by Ellinas and Walker [85] were performed to study the deformation phenomenon, including a constant coefficient, $K$, which stands for the shape of the indenter (normally assumed as 150):

$$ P = \frac{K}{4} \sigma_t r^2 \left( \frac{X}{D} \right)^{1/2} $$  \hspace{1cm} (3-9)

$$ E_s = 100 \cdot m_p \cdot \frac{X^{1.5}}{\sqrt{D}} $$  \hspace{1cm} (3-10)

The above two relations for the indenting force match one another when $D/t$ is equal to 50. The main difference resides on the shape of the indenter which is a wedge-shaped or a rigid beam. It is assumed that the motion of the indenter towards the tube makes a right angle. It should be noted that the indenting force defined by Equations 3-7 and 3-9 are derived without considering the interactions between ship structure and tubular members during the collision.

One of the first reports that can be found on the behaviour of tubular members under dynamic lateral load belongs to Soreide and Kavlie [86], who performed tests on tubes with time-dependent lateral concentrated loads. The response of the preloaded tubular members under lateral dynamic loads was studied by Zeinoddini et al [87], through a series of numerical and experimental tests. All these studies had in common the fact that the lateral loads are applied at the mid-span of member. The effect of impact location along different points of the tube length on tubular member responses was addressed by Khedmati and Nazari [88] by numerical simulations. These works that deal with the behaviour of dented tubular members have two things in common: the indenters used to penetrate the tubes are rigid and the contact is made through a continuous contact surface. Other studies regarding the denting of steel members can be found in [89-94].

When a tubular member subjects to relatively low velocity ship impact, its deformation consists of local denting and global bending. For tubular beams undergoing large deformations the global bending response is estimated using the three-hinge mechanism [95], which is based on the principle of virtual works. The load is modelled as a concentrated load and the method is best applied to braces due to their oblique position and thus an only single zone of contact. This assumption is also acceptable for stern impacts or early stages of broadside impacts where the contact with the platform happens at the ship deck or bilge due to rolling. The relation between concentrated impact load and the mid span deflection is:
\[
\frac{P}{P_u} = \begin{cases} 
1 - \left(\frac{U}{D}\right)^2 + \frac{U}{D} \sin^{-1} \frac{U}{D} \Rightarrow \frac{U}{D} \leq 1 \\
\frac{\pi}{2} \frac{U}{D} \Rightarrow \frac{U}{D} > 1
\end{cases}
\] (3-11)

where \( U \) is the central deflection at the point of impact and \( P_u \) the plastic collapse load of a circular tube with wall thickness \( t \) in pure bending, given by

\[
P_u = \frac{8M_u}{l} = \frac{8\sigma_u D^2 t}{l}
\] (3-12)

This assumption is valid as long as there is no buckling of the tube wall and the full plastic capacity of the cross section can be achieved during the deformation.

The DnV [3] current design practise makes use of Amdahl’s methodology, which takes into account the effects of axial flexibility and the strength of the connections. The axial stiffness of the adjacent structure is replaced by discrete elements (Figure 3-3). When collisions do not occur at the centre of the member, the force-deformation relationship can also be easily worked out. The same approach can also be addressed for cross bracings (Figure 3-4), where for central impact the capacity in comparison with a single brace is naturally doubled. For impacts at a certain distance from the centre, the brace response/deformation (global and local deformation modes) is also given from the equality of internal and external work [96].

Figure 3-3 Bending response for tubular beams with axial flexibility [3]
The local effects such as capacity reductions due to indentation and local buckling of tube wall can be considered by a deformation model at the point of impact. This modification is necessary when the wall thickness is not sufficient to avoid local denting (thinner walls may also allow the occurrence of local buckling). It is based on the assumption that the indented area is flat and the remaining part of the cross section has a constant radius of curvature. The reduced plastic section modulus can be derived by simple integration over the deformed cross section.

As discussed, all the above approaches neglect interactions between ship and tubular structure during impact, and either ship structure or tubular structure is assumed as rigid when calculating the responses of another structure. Recent studies of barge impacts on RC bridge piers found that the impact force and structural response depend on the bridge pier stiffness and geometry since dynamic interaction during barge impact and plastic deformations of the barge bow structure and bridge pier have significant influences on the impact loadings and structural responses [97-101]. Since tubular structures in offshore platforms are usually more flexible as compared to RC bridge piers, relatively large deformation is expected under ship impact. Therefore it is important to take into consideration the deformation and interaction of both ship and platform structures to derive more accurate predictions of impact loadings and dynamic responses and damage of platform structures. In this chapter detailed numerical model is
developed in LS-DYNA and intensive numerical simulations are carried out to investigate the influences of various structural parameters on impact loads and structural responses.

3.3 FEM description

3.3.1 Ship

The Norwegian Petroleum Directorate used to require platforms to be designed for impacts from supply vessels of 5000 ton of displacement with a speed of 2 m/s (added mass effects are included). This yields a kinetic energy of 11 MJ (added mass estimated in 10% for surge) for bow/stern impacts when specified values for hydrodynamic added mass are taken into account. According to Ref. [96], vessel displacements have been increased since 1985 to 6000 ton displacement and Central North Sea and Southern North Sea structures have been subjected to collisions involving ships up to 10000 ton. Amdahl and Johansen [100] obtained force deformation for strength design of jacket legs against bow collisions of vessels with displacements comprised between 2000-5000 tons and the kinetic energy up to approximately 50-55 MJ (velocities up to 6m/s displacement of 3000 tons), which is currently used by the DnV code for the design of bow deformation and energy absorption within the mentioned displacement range. By assuming the inertia effects to be too small for the vessel speeds considered, the impact force for strength design could be estimated according to the crush depth of the bow.

In this study, the model of the ship proposed for numerical experiments is developed to give a good characterization of loading, deformation behaviour and energy distribution that could be expected in a real impact scenario. Therefore, it is essential that the bow structure has a reasonable level of detail. Aspects such as crushing and rupture as well as buckling and plasticity are of important concerns. The finite element model of the bow is shown in Figure 3-5 which replicates to some extent the model of Amdahl and Johansen. It represents a generic model that can be considered representative for vessels in the range of 2000-5000 ton displacement. Since any possible kinetic energy to be absorbed by the ship is dissipated through the deformation of the very front part, the rest of the body has just been given the real outer shape in order to keep the ship dimensions and respective inertia properties in the numerical model (one must note that Equation 3-4 takes the vessel length into account for the estimation of the bow force).
The bow is modelled by a set of shell elements. Thickness values of 11 mm and 9 mm are in a first stage assumed for the shell plating and deck plating (average thickness) respectively. Interior scantlings such as forecastle deck, inner decks, frames and girders have been modelled accurately. The stiffness of the ship bows can vary between different manufacturers and ship types because of different layouts of internal reinforcement and variation of plate thickness. Nevertheless, the detail modelled part is expected to provide a reasonable accuracy on the estimation of the crushing forces. In [101], the crushing and rupture behaviours in ship collisions are analysed through FE simulations. A good agreement is later shown in [102] between the FE results and experimental test models. In Figure 3-6 the true stress strain curve of a piece-wise linear, isotropic hardening material model, representative of the mild steel the ship can be made of, is estimated from the engineering stress-strain curve using the material model III defined in [101]. This model is adopted in the present study to model the stress-strain relation of the steel material. The true stress-strain relationships can model the physical process with better accuracy than the engineering stress-strain relationships. This is because in situations such as the tensile test, the cross sectional area can change substantially and the engineering stress-strain definition ceases to be an accurate measure. Other values that characterize the mechanical properties of the steel used are $E=200$ GPa, $\rho=7800$ kg/m³ and $\nu=0.3$. The rear portion is modelled with solid elements with density in accordance with the intended total mass and mass distribution to be given to the ship. The description of the internal structure can be seen in Table 3-1.
Figure 3-5 Bow model and internal layout (dimensions in meters)

Table 3-1 Bow plate intersections

<table>
<thead>
<tr>
<th>Cross sections*</th>
<th>Number of angles</th>
<th>Number of T-sections</th>
<th>Number of cruciforms</th>
</tr>
</thead>
<tbody>
<tr>
<td>Section 1 (s=0.55m)</td>
<td>2</td>
<td>9</td>
<td>2</td>
</tr>
<tr>
<td>Section 2 (s=1.10m)</td>
<td>2</td>
<td>11</td>
<td>3</td>
</tr>
<tr>
<td>Section 3 (s=1.65m)</td>
<td>4</td>
<td>12</td>
<td>4</td>
</tr>
<tr>
<td>Section 4 (s=2.20m)</td>
<td>2</td>
<td>13</td>
<td>5</td>
</tr>
<tr>
<td>Section 5 (s=2.75m)</td>
<td>2</td>
<td>14</td>
<td>1</td>
</tr>
<tr>
<td>Section 6 (s=3.30m)</td>
<td>2</td>
<td>9</td>
<td>1</td>
</tr>
<tr>
<td>Section 7 (s=3.85m)</td>
<td>2</td>
<td>11</td>
<td>1</td>
</tr>
<tr>
<td>Section 8 (s=4.40m)</td>
<td>2</td>
<td>11</td>
<td>1</td>
</tr>
</tbody>
</table>

*Number of plate intersections according to Amdahl's procedure (as in Figure 1)
The Cowper-Symonds relation [103] can be adopted by the solver for the piecewise linear plasticity material to model the strain rate effect, which is given as follows:

\[
\frac{\sigma^d_0}{\sigma_0} = 1 + \left( \frac{\dot{\varepsilon}}{C} \right)^{1/P}
\]

(3-13)

where \(\sigma^d_0\) is the dynamic flow stress, \(\sigma_0\) is the static flow stress and \(C\) and \(P\) are constants in the strain rate hardening law. The way how these parameters affect the system response is discussed in 3.5.5.

The Overseas Coastal Area Development Institute of Japan [104] reports the following relationship between the displacement tonnage (DT) and the deadweight tonnage (DWT):

\[
\begin{align*}
C & \text{arg o ships} (\leq 10000DWT): \log(DT) = 0.550 + 0.899 \log(DWT) \\
C & \text{arg o ships} (\geq 10000DWT): \log(DT) = 0.511 + 0.913 \log(DWT) \\
Container & \text{ ships}: \log(DT) = 0.365 + 0.953 \log(DWT) \\
Roll & \text{ on roll off vessels}: \log(DT) = 0.657 + 0.909 \log(DWT) \\
Oil & \text{ tanker s}: \log(DT) = 0.332 + 0.956 \log(DWT)
\end{align*}
\]

(3-14)

According to the above relations, a mass of 3000 ton displacement and the bow dimensions of the chosen model nearly matches a 2000 DWT vessel which, according to the same source, would have an overall length \((Lpp)\) estimated in 83 m for a cargo ship or 76 m for an oil tanker.
3.3.2 Tubulars

Tubular members used in offshore steel structures have their response to lateral loading influenced by their geometrical and mechanical properties and external conditions such as axial pre-loading or stiffness of the adjacent members and connections. The pipe members used for numerical calculations are modelled with shell elements of size 100 mm. The mechanical and geometrical properties as well as other parameters regarding the tubes external conditions considered in the numerical tests are shown in Table 3-2 and Table 3-3.

Table 3-2 Mechanical properties of the steel tubular members used in numerical tests

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value/Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>E (GPa)</td>
<td>200</td>
</tr>
<tr>
<td>σ_y (MPa)</td>
<td>250</td>
</tr>
<tr>
<td>σ_u (MPa)</td>
<td>385</td>
</tr>
<tr>
<td>ε_u</td>
<td>0.15</td>
</tr>
<tr>
<td>ν</td>
<td>0.3</td>
</tr>
<tr>
<td>ρ (Kg/m³)</td>
<td>7800</td>
</tr>
</tbody>
</table>

Table 3-3 Geometric dimensions and external conditions of tubes numerically tested

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values/Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter, D (m)</td>
<td>1.5, 2</td>
</tr>
<tr>
<td>Thickness, t (mm)</td>
<td>40, 50, 60, 80</td>
</tr>
<tr>
<td>Length, L (m)</td>
<td>15, 18, 18.5, 21, 22, 24, 26, 27, 30</td>
</tr>
<tr>
<td>Axial preloading</td>
<td>0%, 50% F_a</td>
</tr>
<tr>
<td>Boundary conditions</td>
<td>Fixed ends, Pinned ends</td>
</tr>
<tr>
<td>Strain rate effects</td>
<td>Neglected/Included</td>
</tr>
</tbody>
</table>

3.4 Numerical model calibration

3.4.1 Bow mesh size

In order to verify the reliability of the bow model, the elements comprising the two parts of the ship (bow and rigid blocks) have been developed in LS-DYNA and performed impacting simulations by assuming an impacting speed of 6.0 m/s towards a rigid cylinder with a diameter of 2 m and a rigid wall, respectively. The density of the solid blocks has been adjusted such that the whole mass of the rear part could make up 3000 ton. Together with the ship bow structural elements, the initial kinetic energy is estimated as 54~55 MJ.

Convergence tests are performed to verify the mesh density of the shell elements. These tests are performed assuming the impact with the rigid tube. The explicit dynamic
analyses are carried out using the computer code LS-DYNA. To model the impact and the deformations, two contact algorithms are chosen: the keywords AUTOMATIC_SURFACE_TO_SURFACE, for the slave surface (ship bow elements) hitting the master surface (cylinder), and AUTOMATIC_SINGLE_SURFACE, that prevents the bow elements free penetration when large deformations occur. This is because when the penetration of the cylinder into the ship bow takes place, it is likely that some of the structural elements of the ship will touch one another and so the secondary stiffness of the bow is restored. If this is disregarded, the crushing force could have been underestimated. The shell elements have initially been tested with sizes of 0.1m and 0.05m (see Table 3-4). As shown in Figure 3-7, the both element sizes lead to very similar predictions, but simulation time is greatly reduced if 0.1m element is used although the required simulation time is still substantial. Because the time-step of the explicit numerical analysis drops during the contact, making the total calculation time very long, to further reduce the calculation time the mass scaling technique is adopted. The addition of a non-physical mass to the structure to achieve a large time-step affects the results owing to additional inertial force (\( F=m*a \)). By trying to limit the time step to 1E-6 for a shell element size of 0.1m, it is observed that (see 0.1’ case in Figure 3-7) the error in the calculated impact force owing to the additional non-physical mass never goes over 0.1% as compared to that without using mass scaling technique, but using this technique leads to further significant reductions in computation time. Considering the simulation time and CPU costs, the use of mass scaling techniques and 0.1m shell elements is adopted in this study.

<table>
<thead>
<tr>
<th>Shell element size (m)</th>
<th>Total number of shell elements</th>
<th>Calculation time factor (approximation)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1’ (mass scaling)</td>
<td>64620</td>
<td>1 (between 7-10 hours)</td>
</tr>
<tr>
<td>0.1</td>
<td>64620</td>
<td>3</td>
</tr>
<tr>
<td>0.05</td>
<td>238012</td>
<td>60</td>
</tr>
</tbody>
</table>
3.4.2 Bow crushing force

The comparison between deformations on bows caused by a ‘rigid jacket leg’ and a rigid wall and Equation 3-1 are shown in Figure 3-8. The design curve currently used in the DnV design code recommended for vessels within the range of 2000-5000 tons against jacket legs with diameters between 1.5 m and 2.5 m and kinetic energy up to 60 MJ for bow collisions, is also plotted.

Figure 3-8 Force-Indentation curves of theoretical approaches and LS-DYNA explicit dynamic simulations
It can be seen that, despite a bit conservative, Amdahl’s prediction based on Equation 3-1 shows the same trend as the FE results. If compared to the design curve used in the DnV code, the values of the FE calculations are higher until the maximum force point. It must be noticed that several types of ships can come across offshore platforms and their internal structure varies in many ways and so does their geometry. Equation 1 takes into account parameters such as the spacing between longitudinal frames, thickness of structural elements or areas of the cross sections, but not the colliding structure. Between the collisions with a rigid cylinder and a rigid wall, the expected small reduction in the contact force when colliding with a cylinder can be observed in the figure. The snapshots of the two cases are illustrated in Figure 3-9 at different stages of impact:

![Figure 3-9 Deformation of bulbous bow at different time-steps](image)

The 2000 DWT tanker model used in [80] to estimate bow crushing loads on a rigid wall, at an initial speed of 7 m/s, is compared in Figure 3-10. The contact force for bow with bulb from the DnV is also plotted:
As shown, the use of Pedersen’s tanker model results in higher prediction of impact force. The peak impact force registered of nearly 30 MN before the denting reaches 1 m with Pedersen’s model can be explained by a possible different internal layout of the model. Also, the plate thickness of the 2000DWT ship hull ranges from 8.5 to 18.0 mm, giving a higher plate thickness average than the 11.0 mm used in the current model. This consequently results in a more rigid bow structure, less plastic deformation, less energy absorption and greater exerted force by the bow.

Following Equation 3-4, the maximum bow force \( P_{bow} \) of a 2000 DWT cargo ship colliding with a rigid wall with an initial speed of 6 m/s would be 51.5 MN, dropping to 46.5 MN in the case of the 2000 DWT oil tanker, substantially higher than the 20.2 MN predicted by the FEM calculation or the 24.3 MN predicted by Equation 3-1. Although Equation 3-4 accounts for effects such as the strain rate of the steel, and the size of the ship is similar to the model considered in this study, the ship models considered in deriving the equation are made of thicker plates and different steel mechanical properties. These differences result in different predictions of impact forces.

3.4.3 Tube deformation

Tubular members used in offshore steel structures have their response to lateral loading influenced by their geometrical and mechanical properties and external conditions such as axial pre-loading or stiffness of the adjacent members and connections. Their response to lateral impacts also depends on the type of the indenter they are subjected to during penetration. To compare the response of steel pipes when impacted by rigid
indenters of different sizes (different contact areas are implied), three clamped tubes modelled with shell elements are considered. The three cases are described in Table 3-5. The mechanical properties are the same as those described in Table 3-2. For all the cases the behaviour of the tubes after yielding is defined as fully plastic and the influence of the strain hardening is compared for case C. Two different rigid indenters of different sizes are modelled for penetration. Both indenters have cubic shapes, with the larger one being ten times wider than the thinner one. They give a contact length along the longitudinal direction of the pipe of 0.2 m and 2m respectively, or 2% and 20% of the full length of the tube A. In simulations, a vertical prescribed motion of 0.5 m/s is given to the indenters, downwards to the tube that is horizontally positioned (see Figure 3-11).

<table>
<thead>
<tr>
<th>Tube</th>
<th>L [m]</th>
<th>D [m]</th>
<th>t [mm]</th>
<th>L/D</th>
<th>D/t</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>10</td>
<td>1.5</td>
<td>30</td>
<td>6.7</td>
<td>50</td>
</tr>
<tr>
<td>B</td>
<td>10</td>
<td>1.5</td>
<td>43</td>
<td>6.7</td>
<td>34.9</td>
</tr>
<tr>
<td>C</td>
<td>20</td>
<td>1.5</td>
<td>43</td>
<td>13.3</td>
<td>34.9</td>
</tr>
</tbody>
</table>

Figure 3-11 Deformation shape of tubular members subjected to impacts from different indenters

Experimental impact tests reported in [105] conducted on pressurized mild steel pipes with clamped ends have shown, for $D/t$ ratios of 35.3 and a $L/D$ proportion of 10, that the semi-empirical predictions from Equation 3-9 would better agree with their experimental results if $K$ was taken as twice as the empirical value adopted in [85], i.e. $K=300$. However, in [106] $K=150$ is considered valid for $X/t$ values up to 5.8, which approximately corresponds to the case when the local denting deformation stops and the global displacement takes place. The displacement values of the numerical tests are
recorded and plotted in Figure 3-12 until failure. As shown, for the ultimate strain defined in Table 3-2, the total deflection values obtained prior to failure of the shell elements are relatively small when compared to those predicted by Equation 3-9 with $K=300$. It is obvious in Figure 3-12 that the prediction with $K=150$ shows a reasonable agreement for both $D/t$ values for the thin striker. In fact, neither in case A nor in case B global deflection has been detected. For tube C, the increase of the tube length provokes a very small drop in the contact force for the same deformation value, and very small displacements of the lower membrane are observed. The effects of the strain hardening are also compared in Figure 3-13, where it is clear that their inclusion in the material properties will not significantly affect the tube response, except for the tube failure at higher deflection values possibly due to the higher capacity of rotation of the plastic zone.

Figure 3-12 Force deformation curves, comparison with Equation 3-9 (Left: D/t=50; Right D/t=35)

Figure 3-13 Force deformation curves for different tube length
The results for the penetration using the larger indenter lie in between the curves for $K=150$ and $K=300$, indicating the indenter size will influence the response as it changes the contact area between striker and struck tube.

Currently, the DnV [3] resistance curves for local denting predict the variation of the flattened contact area through the use of constants that take into account the ratio between the length of the flattened surface and the diameter of the tube. The comparisons of the DnV curves with the numerical tests for tubes A and B using the two indenters show an acceptable agreement in Figure 3-14.

To examine what the force-deformation behaviour of a steel tube hit by a striker with an irregular shape such as a ship bow could be, the shell of the bow model is set as rigid with the rigid blocks that make up the ship rear part attached. The hit tube is given a diameter of 2m, with thickness and length of 40 mm and 10 m respectively. The steel properties are taken from Table 3-2 as well.

From the contact with a bulbous bow there are two main points of contact that may locally deform the tube wall: the region where the tube wall is hit by the bulb and the zone at the level of the forecastle deck. Due to the geometry and the areas in contact the bulb is responsible for the higher portion of the total contact force. The curve from the FE simulation is compared with the force-deformation relationships for denting [3] in Figure 3-15.
As shown, the DnV estimation agrees well with the FE simulation until the denting reaches approximately 12-13% (0.24 to 0.26 m) of the tube diameter. At this point, shear failure occurs at the upper contact point. This is justified by the sharpened bow shape at the top. This first failure is responsible for the sudden rise in the FE curve. In this particular case, the failure of the lower contact point happens when the indentation reaches around 75% of the diameter. It should be noted that these results are obtained with rigid ship bow assumption. In real case, the bows are not rigid and so the estimation of the contact force has also to consider the bow deformations. Nonetheless this observation demonstrates that the applicability of the design curves has to take into account two aspects: the indenter strength and the denting configuration. It must be noted that the scope of Equations 3-7 to 3-10 is based on broadside impacts that are likely to take place at mooring of supply vessels. The fact that head-on collisions involving bulbous bows can result in two distinct indentation areas relates the penetration force to a different denting configuration. The current practice assumes the deflection mechanism of the tube membrane similar to a nonlinear spring that considers the geometrical parameters referred in Equations 3-7 – 3-10. In Figure 3-15 the DnV design curve is obtained by considering the sum of the two contact points as the total contact area. It is not clear, however, what the error could be for different distances between the two contact points, which in some cases could result in interference between the two denting mechanisms. This issue, however, is not investigated in the present work, as the primary objective of the present study is to investigate the
influences of ship and platform structure interaction during impact and the
deformations of both ship and platform structures on impact loads and structural
response and damage.

3.5 Parametric investigation

3.5.1 Tube geometry

The way how the energy is absorbed through the tubes depends on the nature of the
deformation they are subjected to. Local denting and bending can interfere with one
another during the deformation process. The damage caused by the local denting
deformation of the tube wall is influenced by the global bending deformation and hence
by the cross section geometry, tube length and boundary conditions. From Equations 3-7
and 3-9, it is obvious that the wall thickness of the tubes has a higher influence on the
denting capacity than the diameter of the cross section. Considering the tube cross
section diameter as 1.5m and 2.0m respectively, the denting resistance varies, according
to Equation 3-9, by 15.5% for the same wall thickness and yield strength while
according to Equation 3-7 it is not affected. On the other hand, the influence of
thickness is included in both the expressions, being the resistance increased by factors
of 2.83 (Equations 3-7 and 3-8) and 4 (Equations 3-9 and 3-10) with the thickness
increasing from 40 to 80 mm for a given section diameter, implying significant
differences in those formulae. The same mentioned factors obviously also affect the
energy absorptions and according to these expressions the energy absorption through
local denting is mainly related to the plastic moment of the thin walled tubes.

The distinction between local and global deformations on tubular members due to
impact has rarely been reported in experimental studies, where the overall displacement
is assessed rather than the contribution of each of the deformation modes. In [106] the
local and global displacements are estimated from an idealized deformed cross section
configuration described in Figure 3-16.
The following relationships can be assumed and compared to measurements from the impact test:

\[ r_0 = T_r \cdot \left[1 + \left(\frac{D_m}{2T_r}\right)^2\right]/2 \]  \hspace{1cm} (3-15)

\[ \beta = \pi D/4r_0 \]  \hspace{1cm} (3-16)

\[ \cos \phi_0 = 1 - T_r/r_0 \]  \hspace{1cm} (3-17)

\[ \delta = r_0(\cos \beta - \cos \phi_0) \]  \hspace{1cm} (3-18)

\[ X = D/2 - \delta \]  \hspace{1cm} (3-19)

\[ U = \Delta - X \]  \hspace{1cm} (3-20)

in which \( r_0 \) is the radius of the deformed cross section, \( T_r \) the local permanent thickness of the deformed cross section and \( D_m \) the maximum width of the deformed cross section.

The graphic in Figure 3-17 relates the tube dimensions with the dominant deformation mode for different tubes with fixed ends after impact of the ship model described in section 3.3.1. As a criterion, it has been considered that the cross sections undergoing deformations of less than 2% of their diameter is assumed as fully effective or rigid. Thus, for shorter and thicker members, the behaviour can be assumed as practically...
rigid section because of small local denting and global beam flexural deformations. This is verified by the numerical simulations of the cases where the member wall is 60 mm thick, and where the energy locally absorbed by the steel tubular wall is not higher than ~2.5% of the impact energy (~55 MJ) of the full system (ship and platform) system, i.e. less than ~1.3 MJ. As shown in the figure, when $L/D$ ratio is large, flexural bending tends to dominate the tubular member deformation, while local denting deformation dominates when the $L/D$ ratio is small and member wall is thin.

![Deformation modes of tubes](image)

The cases shown in Figure 3-18 are compared with those calculated by Equations 3-15 – 3-20 in Table 3-6. The calculated deformation value $X$ obtained by Equation 3-19 from the values $D_m$, $A$ and $T$, measured from the numerical models after the impact, is compared with the same parameter $X$ directly taken from the numerical simulations. In Table 3-6, $D$ represents tube diameter, $t$ thickness and $L$ length. For example, D2t40L15 represents a tube of diameter 2 m, thickness 40 mm and length 15 m.
Table 3-6 Numerical results of fixed-ends steel pipe deformation after ship bow impact

<table>
<thead>
<tr>
<th>Test</th>
<th>Test (D[m][m] L[m])</th>
<th>Tl [m]</th>
<th>D0 [m]</th>
<th>r0 [m]</th>
<th>β [rad]</th>
<th>Δ0 [rad]</th>
<th>δ [m]</th>
<th>X [m]</th>
<th>D [m]</th>
<th>Δ [m]</th>
<th>U [m] (measured)</th>
<th>X [m]</th>
<th>Es [MJ]</th>
</tr>
</thead>
<tbody>
<tr>
<td>D2t40L15</td>
<td>1.42</td>
<td>2.14</td>
<td>1.11</td>
<td>1.41</td>
<td>-0.28</td>
<td>0.48</td>
<td>0.52</td>
<td>2.00</td>
<td>0.55</td>
<td>0.03</td>
<td>0.58</td>
<td>5.57</td>
<td></td>
</tr>
<tr>
<td>D2t40L21</td>
<td>1.09</td>
<td>2.34</td>
<td>1.17</td>
<td>1.34</td>
<td>1.50</td>
<td>0.19</td>
<td>0.81</td>
<td>2.00</td>
<td>1.09</td>
<td>0.28</td>
<td>0.91</td>
<td>10.44</td>
<td></td>
</tr>
<tr>
<td>D2t40L30</td>
<td>1.14</td>
<td>2.41</td>
<td>1.21</td>
<td>1.30</td>
<td>1.52</td>
<td>0.26</td>
<td>0.75</td>
<td>2.00</td>
<td>1.73</td>
<td>0.99</td>
<td>0.86</td>
<td>15.67</td>
<td></td>
</tr>
<tr>
<td>D1.5t40L15</td>
<td>1.12</td>
<td>1.54</td>
<td>0.83</td>
<td>1.43</td>
<td>1.93</td>
<td>0.41</td>
<td>0.34</td>
<td>1.50</td>
<td>0.50</td>
<td>0.16</td>
<td>0.39</td>
<td>4.61</td>
<td></td>
</tr>
<tr>
<td>D1.5t40L22</td>
<td>1.46</td>
<td>1.50</td>
<td>0.92</td>
<td>1.28</td>
<td>2.19</td>
<td>0.81</td>
<td>-0.06</td>
<td>1.50</td>
<td>0.29</td>
<td>0.35</td>
<td>0.04</td>
<td>3.20</td>
<td></td>
</tr>
<tr>
<td>D1.5t40L30</td>
<td>1.48</td>
<td>1.50</td>
<td>0.93</td>
<td>1.27</td>
<td>2.20</td>
<td>0.83</td>
<td>-0.08</td>
<td>1.50</td>
<td>0.96</td>
<td>1.04</td>
<td>0.02</td>
<td>11.97</td>
<td></td>
</tr>
<tr>
<td>D2t50L15</td>
<td>1.78</td>
<td>2.03</td>
<td>1.18</td>
<td>1.33</td>
<td>2.11</td>
<td>0.88</td>
<td>0.12</td>
<td>2.00</td>
<td>0.02</td>
<td>-0.10</td>
<td>0.22</td>
<td>1.10</td>
<td></td>
</tr>
<tr>
<td>D2t50L21</td>
<td>1.71</td>
<td>2.07</td>
<td>1.17</td>
<td>1.34</td>
<td>2.06</td>
<td>0.81</td>
<td>0.19</td>
<td>2.00</td>
<td>0.28</td>
<td>0.09</td>
<td>0.29</td>
<td>1.71</td>
<td></td>
</tr>
<tr>
<td>D2t50L30</td>
<td>1.53</td>
<td>2.17</td>
<td>1.15</td>
<td>1.37</td>
<td>1.91</td>
<td>0.62</td>
<td>0.38</td>
<td>2.00</td>
<td>0.47</td>
<td>0.08</td>
<td>0.47</td>
<td>6.64</td>
<td></td>
</tr>
<tr>
<td>D1.5t50L15</td>
<td>1.49</td>
<td>1.51</td>
<td>0.94</td>
<td>1.26</td>
<td>2.20</td>
<td>0.84</td>
<td>-0.09</td>
<td>1.50</td>
<td>0.01</td>
<td>0.10</td>
<td>0.12</td>
<td>0.72</td>
<td></td>
</tr>
<tr>
<td>D2t60L18</td>
<td>1.21</td>
<td>2.30</td>
<td>1.15</td>
<td>1.36</td>
<td>1.62</td>
<td>0.29</td>
<td>0.71</td>
<td>2.00</td>
<td>0.79</td>
<td>0.09</td>
<td>0.83</td>
<td>7.80</td>
<td></td>
</tr>
<tr>
<td>D2t40L24</td>
<td>1.03</td>
<td>2.40</td>
<td>1.21</td>
<td>1.30</td>
<td>1.42</td>
<td>0.15</td>
<td>0.86</td>
<td>2.00</td>
<td>1.40</td>
<td>0.55</td>
<td>0.97</td>
<td>14.59</td>
<td></td>
</tr>
<tr>
<td>D2t40L27</td>
<td>0.88</td>
<td>2.50</td>
<td>1.33</td>
<td>1.18</td>
<td>1.23</td>
<td>0.06</td>
<td>0.95</td>
<td>2.00</td>
<td>1.94</td>
<td>0.99</td>
<td>1.12</td>
<td>20.55</td>
<td></td>
</tr>
<tr>
<td>D1.5t50L18.5</td>
<td>1.24</td>
<td>1.52</td>
<td>0.85</td>
<td>1.38</td>
<td>2.04</td>
<td>0.54</td>
<td>0.21</td>
<td>1.50</td>
<td>0.26</td>
<td>0.06</td>
<td>0.17</td>
<td>1.99</td>
<td></td>
</tr>
<tr>
<td>D2t50L27</td>
<td>1.65</td>
<td>2.10</td>
<td>1.16</td>
<td>1.35</td>
<td>2.01</td>
<td>0.74</td>
<td>0.26</td>
<td>2.00</td>
<td>0.47</td>
<td>0.21</td>
<td>0.35</td>
<td>3.61</td>
<td></td>
</tr>
<tr>
<td>D2t50L18</td>
<td>1.74</td>
<td>2.05</td>
<td>1.17</td>
<td>1.34</td>
<td>2.08</td>
<td>0.84</td>
<td>0.16</td>
<td>2.00</td>
<td>0.25</td>
<td>0.09</td>
<td>0.26</td>
<td>1.39</td>
<td></td>
</tr>
<tr>
<td>D2t50L24</td>
<td>1.70</td>
<td>2.06</td>
<td>1.16</td>
<td>1.35</td>
<td>2.05</td>
<td>0.79</td>
<td>0.21</td>
<td>2.00</td>
<td>0.35</td>
<td>0.14</td>
<td>0.30</td>
<td>2.32</td>
<td></td>
</tr>
<tr>
<td>D1.5t60L26</td>
<td>1.33</td>
<td>1.57</td>
<td>0.90</td>
<td>1.32</td>
<td>2.08</td>
<td>0.66</td>
<td>0.09</td>
<td>1.50</td>
<td>1.06</td>
<td>0.98</td>
<td>0.17</td>
<td>12.13</td>
<td></td>
</tr>
</tbody>
</table>

As can be noted from the table, the idealized deformation shape defined in Equations 3-15 – 3-20 does not necessarily give reliable prediction of the tube deformation by ship bow impact. This is because the contact areas of ship bow with the tube are irregular and not flat and therefore the dented section will not be plane. Nevertheless, the calculated results from Equations 3-15 – 3-20 show a reasonable accuracy when the deformation is relatively large, e.g., at least 30% of the diameter of the initial cross section. When the deformation is small, the prediction of these equations is not accurate, indicating the deformation shape and the contributions from denting and global bending are difficult to be idealized.

In regards to the energy assessment, the energy can be non-dimensionalized with respect to the static collapse load for bending $P_a$ given in Equation 3-12. In [107] the dimensionless energy is first defined, considering specimens with varying $D/t$ ratios, as:
where $P_u t$ represents the external work of the concentrated collapse load $P_u$ that produces a transverse displacement $t$ right underneath the load application point. To account for the different material properties of tubes of similar geometry, Equations 3-21 and 3-22 are re-written as:

$$\lambda = \frac{E_k}{P_u t}$$  \hspace{1cm} (3-21)

$$\lambda = \frac{LE_k}{8\sigma_y D^2 t^2}$$  \hspace{1cm} (3-22)

using $\sigma_u$ instead of $\sigma_y$ in the calculation of $P_u$. In the above equations, the initial kinetic energy of the striker, $E_k$, is used.

The maximum transverse displacements, in turn, can be written in a dimensionless form with respect to the wall thickness or the tube diameter. In Equations 3-21 to 3-24, however, the dimensionless energy is obtained from the initial kinetic energy. From the reported tests using a rigid striker, the remaining energy after the impact is divided between the plastic deformation energy of the tube and the kinetic energy due to rebound, being that the kinetic energy from the rebound represents a very small amount of the initial kinetic energy from the impact. This is consistent with the numerical simulations that at least 85% impact energy is converted to plastic deformation energy.

The numerical results are compared with experimental results reported in [107] in Figure 3-18. The dimensionless energies are calculated based on the plastic deformation energy. Since in impacts with deformable strikers the plastic deformation energy of the tube can decrease significantly in terms of percentage of the total energy since the striker will also absorb energy, it becomes plausible that the dimensionless energies take into account only the plastic deformation energy of the tube. In Figures 3-18 and 3-19 both dimensionless displacements and energies are considered. It is found that if the plastic deformation energy of the experimental tests with various $D/t$ ratios is assumed rather than the initial kinetic energy, the results will nearly follow a linear trend without significant dispersion for $\Lambda$ vs $\Delta/D$ and $\lambda$ vs $\Delta/t$. For $\lambda$ vs $\Delta/D$ an approximate linear behaviour can also be observed, and the numerical tests are better
correlated by using a ratio of 60 for $D/t$. It shall be referred that these relationships take into account the total displacement of the membrane, regardless of the deformation modes. It should also be noted that the way the absorbed energies made dimensionless by a single static load at collapse might become inappropriate for the case of ship collision with a wider contact area since for similar penetration depth it will result in greater amounts of plastic deformation energy. The numerical results, divided into the different deformation modes are plotted separately in Figure 3-19. The main difference is observed for ‘bending without denting’ cases mainly for $\Lambda$ vs $\Delta/t$, which suggests that the dimensionless energies and the dimensionless displacements are better correlated if very long tubes with thick walls would not be considered. In [107] it is also suggested, based on two sets of data, that geometrically similar scaling could be assumed for design purposes. For easier comparison, approximate values of $L/D$ are chosen with $D/t$ equal to 40. The scale factors, $\chi$, for diameter and thickness have very close range of values of 0.900-1.922 for $t$ and 1.001-2.135 for $D$. Even though it is not totally clear due to a gap and possibly some lack of sufficient data, it appears that the results plotted in Figure 3-20 seem to agree with the experimental tests as a linear trend could be projected over the chart. Therefore, the principles of geometrically similar scaling can be assumed to be appropriate also for impacts using deformable strikers.
Figure 3-18 Comparison of maximum dimensionless strain energy versus maximum non-dimensional displacement for fixed tubes for numerical and experimental tests

Figure 3-19 Maximum dimensionless strain energy versus maximum non-dimensional displacement for fixed tubes considering different deformation modes

Figure 3-20 Scaled maximum membrane displacement ($\Delta$) versus scaled strain energy of tubes ($E_s$) – Scaled tests from [107]
Figure 3-21 shows the amount of kinetic energy absorbed by the leg with respect to the plastic moment of the tube wall and the tube slenderness. It is obvious that bending response absorbs more energy than local denting even for shorter members with thinner walls, for which local denting also contributes to significant amounts of energy absorption. As expected, the maximum energy absorption results from the combination of the two deformation modes when the tube is slender with large bending deformation and the wall is thin with considerable denting deformation (see also Table 3-6).

![Figure 3-21 Geometry vs. absorbed energy of tubes](image)

The energy absorption by plastic deformation of the platform steel member therefore depends on both plastic hinge formations due to bending and plastic wall deformation owing to local denting. For structure integrity, it is important that denting should be minimized. According to the API [1], the maximum value that the ratio $D/t$ shall assume in the design in order to maintain the full capacity of the cross section through plastic deformation is $9000/\sigma_y$. It can be seen from Figure 3-17 that the tubes with wall thickness of 60 mm or higher will not experience any significant local denting at any point along their length when crashed by the ship of section 3.1. Although the API data are for broadside collisions, the analyses for bow collision with appropriate parameters seem to agree with the same conditions.

The 8.5 MN of a concentrated load obtained by Amdahl and Johansen [100], as the maximum admissible concentrated load to prevent a tube wall of 80 mm thick from
denting, reveals to be much larger than those obtained in the current study. Table 3-7 gives the maximum forces to prevent local denting obtained in this study. It should be noted that these results are obtained with the two types of contact areas illustrated in Table 3-7 and the contact force assumed to be uniformly distributed in the contact area, and the tube diameter is 2 m and wall thickness of at least 60 mm.

Table 3-7 Local collision forces evenly distributed over regular areas

<table>
<thead>
<tr>
<th>Contact area with bulb</th>
<th>Contact area with superstructure</th>
</tr>
</thead>
<tbody>
<tr>
<td>a (m)</td>
<td>0.6</td>
</tr>
<tr>
<td>b (m)</td>
<td>0.5</td>
</tr>
<tr>
<td>Force (MN)</td>
<td>1.94</td>
</tr>
<tr>
<td>c (m)</td>
<td>0.3</td>
</tr>
<tr>
<td>d (m)</td>
<td>0.6</td>
</tr>
<tr>
<td>Force (MN)</td>
<td>0.85</td>
</tr>
</tbody>
</table>

For the ‘only denting’ cases, the tube absorbed energy is listed in Table 3-8. As can be noticed, denting only absorbs a small amount of the total energy. Most impact energy is absorbed by the global bending deformation of the tubular member and ship structure deformation as will be discussed in more detail later.

Despite the same total impact energy from ship considered in all the cases, the absorbed energy owing to plastic deformation of platform member and ship differ slightly, indicating different amounts of kinetic energy of rebound that can take place after the impact. However, the energy absorbed by the platform member denting deformation differs significantly, indicating very different plastic deformations for different structural conditions. The maximum absorbed energy of 7.80 MJ registered on dented tubes corresponds to a larger damaged area, not necessarily the deepest ship penetration. The differences are noticeable when the wall thickness changes. It should be noted that bending of the steel members will absorb additional amounts of the impact energy, which are not included in Table 3-8. In fact, the clamped tubes undergoing bending with or without dent of the cross section can absorb great amounts of impact energy due to
the flexural deflection, depending on the tube length, usually larger than that due to the membrane displacement (denting deformation) as will be discussed below.

<table>
<thead>
<tr>
<th>Impacted tube (D[m]t[mm] L[m])</th>
<th>Platform dented member</th>
<th>Total Deformation Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>D2t40L15</td>
<td>5.37</td>
<td>53.91</td>
</tr>
<tr>
<td>D2t50L15</td>
<td>1.09</td>
<td>53.84</td>
</tr>
<tr>
<td>D2t50L21</td>
<td>1.71</td>
<td>54.04</td>
</tr>
<tr>
<td>D1.5t50L15</td>
<td>0.72</td>
<td>53.88</td>
</tr>
<tr>
<td>D2t40L18</td>
<td>7.80</td>
<td>55.70</td>
</tr>
<tr>
<td>D1.5t50L18.5</td>
<td>1.99</td>
<td>55.61</td>
</tr>
<tr>
<td>D2t50L18</td>
<td>1.39</td>
<td>55.56</td>
</tr>
<tr>
<td>D2t50L24</td>
<td>2.32</td>
<td>55.73</td>
</tr>
</tbody>
</table>

Considering the flexural response, it is found that the maximum displacement of the tube elements on the compressive side of the tube in direct contact with the ship bulb occurs slightly below the mid-length of the tube element. The non-dimensional displacement, representing the horizontal displacement of the tube membranes regardless of the governing response, when related to the strain energy, increases linearly. Figure 3-22 confirms, from all the numerical simulations considering fixed-ends tubes, that the energy absorption is proportional to the total deformation of the tubular member irrespective if it is local denting or global bending deformation.

![Figure 3-22 Energy vs. non-dimensional displacement of tube membrane (Δ – total displacement of membrane; D – diameter of the tube)](image)

**3.5.2 Bow strength**

Within a design scenario where the strain energy is dissipated over the two colliding bodies, it is of interest to determine the amount of energy dissipation by each body with
varying parameters. As said earlier, the bow model geometry defined in section 3.3.1 is a replica of the model used in [100] to define the bow deformation behaviour in high energy collisions. The vessels analysed in [80] have the plate thickness, for the 2000 DWT, ranged from 8.5 to 18.0 mm, meaning a higher plate thickness average than the 11.0 mm assumed in this study. To study the influences of bow plate thickness on ship-platform interaction, similar simulations as those presented above are repeated with the shell thickness of the bow model increased to 14.0 mm. The peak of the contact forces are obtained as 27.72 MN and 20.32 MN from the bow crushing against a rigid wall and a rigid cylinder of 2 m diameter, respectively, higher than the cases from the previous simulation of 17 MN peak force when the ship plate thickness is 11 mm. Figure 3-23 shows the energy absorption with respect to the bow crushing distance. As expected, a thicker bow plate corresponds to less crushing distance at the same amount of energy absorption.

To examine the influences of the bow and tubular structure interaction on impact forces, numerical simulations are carried out with bow plate thickness of 11 mm and 14 mm, and tubes of different length, diameter and wall thickness. The simulated impact force time histories are shown in Figure 3-24. As shown, in general increasing the stiffness of bow structure and tube structure leads to a larger impact force but shorter force duration. The detailed simulation results, which are not shown here, also demonstrate that the 60 mm thick wall of the tube experiences only small local denting deformation under the collision of ship with bow plate thickness of 14 mm and $X/D$ not greater than 8%, as compared to almost no deformation when the bow plate thickness is 11 mm as
discussed in 3.5.1. When the tube wall is 80 mm thick, no obvious local denting deformation is generated even the bow plate is 14 mm thick. These results indicate that the relatively weaker structure, either bow or tube, will experience larger plastic deformation and absorb more energy during the collision.

![Force-time history of contact forces from ship bows of different strength](image)

Figure 3-24 Force-time history of contact forces from ship bows of different strength

The reduction of the impact duration of about 0.25-0.5 seconds with the increase of the bow strength (plate thickness here) is observed in Figure 3-25. The higher difference occurs in the cases of the tubes behaving rigidly and undergoing bending deformation, implying the tube deformation and particularly bending deformation prolongs the ship collision process. The impact duration predicted as 0.49 seconds by Equations 3-5 and
3-6 in a strength design scenario is far shorter than what is obtained in the present calculations because those equations were derived without considering interactions between ship and tube structures during impacting. By assuming the maximum penetration of a rigid obstacle with unlimited width, the duration would ascend to 0.84 seconds in Equations 3-5 and 3-6, still less than the 1.26 seconds obtained when colliding of the stronger bow against a rigid wall, i.e. the same impacting scenario considered in [80]. These observations indicate those equations significantly underestimate the impact loading duration.

3.5.3 Boundary conditions

In the above simulations, the tubes are assumed to be clamped at both ends. Such assumption, however, does not necessarily represent the true boundary conditions of tubular members in a steel offshore platform. Since the boundary conditions of the tubular member affect its bending deformation and ability to resist impact loads, for example the longitudinal membrane forces and plastic hinges are not developed at the ends of a pinned tube, and the value of the plastic limit load defined in Equation 3-12 is reduced to half when the ends switch from fixed to pinned, the influences of boundary conditions are investigated in this section.

To model the free rotation condition at the tube ends, an extra node at the centre of each tube end cross section is defined. The translational degrees of freedom of this additional node are restrained but it can rotate freely. All the nodes at the tube end cross sections are rigidly connected to this centre node. Thereby the tube end can rotate with respect to the centre node. It should be noted that for a real tubular member in a platform structure, its boundary conditions are neither fully fixed nor pinned. The two boundary conditions considered in the present study represent the two idealized cases. The true behaviours are in between these two extreme boundary conditions.

Six cases have been considered for the different deformation scenarios in section 3.5.1. The comparison between the same cases for the pinned and fixed ends is given in Table 3-9. As can be noticed, although the ultimate plastic load of a beam with pinned ends is reduced to half, this does not mean that the total impact loads for which the contact occurs until the collapse of the jacket leg would be reduced by the same amount. The contact force is affected by the deformations of the ship bow as well. The peak values of
the contact forces obtained for the cases under consideration are actually very close for the two tube boundary conditions, as can be noted in Table 3-9.

Table 3-9 Data for hit tubes with different boundary conditions

<table>
<thead>
<tr>
<th>Tube (D[m][mm] L[m])</th>
<th>Ends</th>
<th>Impact duration (s)</th>
<th>Energy absorbed (MJ)</th>
<th>Estimated $P_u$ from Eq. 3-12 (MN)</th>
<th>Maximum contact force (MN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>D1.5t40L18.5</td>
<td>fixed</td>
<td>0.92*</td>
<td>9.6*</td>
<td>9.7</td>
<td>15.1*</td>
</tr>
<tr>
<td></td>
<td>pinned</td>
<td>1.01*</td>
<td>15.4*</td>
<td>4.9</td>
<td>16.3*</td>
</tr>
<tr>
<td>D1.5t50L22</td>
<td>fixed</td>
<td>1.02*</td>
<td>1.0*</td>
<td>10.2</td>
<td>14.8*</td>
</tr>
<tr>
<td></td>
<td>pinned</td>
<td>0.94*</td>
<td>12.1*</td>
<td>5.1</td>
<td>16.0*</td>
</tr>
<tr>
<td>D1.5t80L30</td>
<td>fixed</td>
<td>1.90</td>
<td>12.0</td>
<td>12.0</td>
<td>16.3</td>
</tr>
<tr>
<td></td>
<td>pinned</td>
<td>2.05</td>
<td>11.3</td>
<td>6.0</td>
<td>16.6</td>
</tr>
<tr>
<td>D2t50L21</td>
<td>fixed</td>
<td>1.90</td>
<td>1.7</td>
<td>19.0</td>
<td>16.4</td>
</tr>
<tr>
<td></td>
<td>pinned</td>
<td>1.90</td>
<td>12.3</td>
<td>9.5</td>
<td>15.4</td>
</tr>
<tr>
<td>D2t40L27</td>
<td>fixed</td>
<td>1.78</td>
<td>20.6</td>
<td>11.9</td>
<td>16.2</td>
</tr>
<tr>
<td></td>
<td>pinned</td>
<td>1.82</td>
<td>23.2</td>
<td>5.9</td>
<td>16.3</td>
</tr>
<tr>
<td>D2t60L30</td>
<td>fixed</td>
<td>1.90</td>
<td>1.3</td>
<td>16.0</td>
<td>17.2</td>
</tr>
<tr>
<td></td>
<td>pinned</td>
<td>2.15</td>
<td>13.6</td>
<td>8.0</td>
<td>13.6</td>
</tr>
</tbody>
</table>

*Tube collapse due to failure of steel

The duration of the impact is naturally shorter for the clamped tubes and the occurrence of the peak of the contact force in general takes place earlier as compared to those of the pinned tube because the different structural stiffness corresponding to the two boundary conditions. Simulation results, which are not shown here, also indicate different deformation modes when the tubes have different boundary conditions. For example, as presented above, when the ship with 11 mm bow plate thickness collides with the fixed ends tubular member with tube wall thickness 60 mm, it induces almost no local denting deformation in the tube wall. However, very small local denting deformation that occurs due to the absence of axial membrane forces can be observed under the same colliding condition when the tubular member has pinned boundaries and undergoes global deflections. The percentages of energy absorption also vary with the different deformation modes. Nonetheless, most of the energy is still absorbed by the tube global bending deformation, the same as that discussed above for the case with fixed ends tube structure. Because the pinned boundary makes the tube structure more flexible than the fixed boundary, the impact duration increases but the plateau of the maximum average contact force reduces (Figure 3-25).
which the steel strain reaches the predefined value of 0.15 in this study, leading to erosion of the finite elements and consequent rupture, is also different. For the tubes with pinned ends, the rupture occurs at mid length of the member where a plastic hinge is formed and the contact stress is greater, whereas for clamped members the rupture occurs at the lower end of the member.

Although no general conclusion can be drawn for the amplitude of impact force with respect to the tube boundary conditions, the boundary condition has consistent influences on absorbed energy of the tube. In Figure 3-26 the results corresponding to the cases presented in Table 3-9 are plotted and compared, similar to the discussions presented in section 3.5.1. The cases with failure occurrence are also included. As shown, it is possible to establish a connection between the dimensionless absorbed energy and displacement. It should be noted that the values of the non-dimensional energy for pinned members are obtained from Equations 3-21 to 3-24 by considering $P_u$ as half of the expression given in Equation 3-12 for fixed-ends members.
3.5.4 Axial preloading

Since the offshore platforms are carrying their design loads during the ship impact, it is of significance to analyse how the tube response is affected when subjected to axial stress. The combination between the axial prestress and the lateral impact load may increase the chance of failure due to buckling of the tubular member and consequently the failure of the whole platform. To examine the preload effects on the legs laterally impacted by the bow, some of the previous cases are analysed again with the axial pre-compression added. To study the coupled effects of boundary conditions and axial loading, besides the geometric properties of the tubes, the end restraints are also alternated between fixed and pinned conditions.

The numerical calculations involving axial preloading consist of two steps. The “dynamic relaxation” is applied at the beginning with the application of the design loads necessary for the desired axial stress to be achieved prior to the second-step impact analysis. The levels of preloading that have been introduced in this study correspond to the 50% of the allowable axial compressive stress, $F_a$, obtained from the API code.

The tube parameters for the analysis of the axial preloading effects are given in Table 3-10.
Table 3-10 Parameters of the preloaded tubes analysed

<table>
<thead>
<tr>
<th>Case</th>
<th>Tube (D[m] t[mm] L[m])</th>
<th>Ends</th>
<th>50% $F_a$ (MPa)</th>
<th>$P_u$ (MN)</th>
<th>Maximum contact force recorded (MN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>D1.5t60L26</td>
<td>Fixed</td>
<td>63.3</td>
<td>10.4</td>
<td>9.3*</td>
</tr>
<tr>
<td>2</td>
<td>D1.5t80L30</td>
<td>Pinned</td>
<td>60.6</td>
<td>6.0</td>
<td>10.1*</td>
</tr>
<tr>
<td>3</td>
<td>D2t50L22</td>
<td>Fixed</td>
<td>68.7</td>
<td>18.1</td>
<td>9.8*</td>
</tr>
<tr>
<td>4</td>
<td>D2t60L30</td>
<td>Fixed</td>
<td>65.5</td>
<td>16.0</td>
<td>7.9*</td>
</tr>
<tr>
<td>5</td>
<td>D2t80L30</td>
<td>Fixed</td>
<td>65.4</td>
<td>21.3</td>
<td>16.5</td>
</tr>
<tr>
<td>6</td>
<td>D2t80L30</td>
<td>Pinned</td>
<td>65.4</td>
<td>10.6</td>
<td>12.5*</td>
</tr>
<tr>
<td>7</td>
<td>D2t50L15</td>
<td>Fixed</td>
<td>71.1</td>
<td>26.6</td>
<td>16.1</td>
</tr>
<tr>
<td>8</td>
<td>D1.5t80L15</td>
<td>Pinned</td>
<td>69.2</td>
<td>12.0</td>
<td>16.6</td>
</tr>
<tr>
<td>9</td>
<td>D2t60L15</td>
<td>Pinned</td>
<td>71.1</td>
<td>16.0</td>
<td>15.7</td>
</tr>
</tbody>
</table>

*Tube collapse due to failure of steel ($P_u$ calculated according to Equation 3-12)

As the $D/t$ ratio for the selected tubes is greater than 60, it is not expected according to API code as well as the FE simulation results presented above that significant local buckling would take place. Therefore, only global buckling can lead to the member failure.

A previous study demonstrated that the application of axial preloading affects the force versus tube deformation, namely local denting, relationships [88]. Both the maximum load capacity of the tubes and tube maximum deformation can diminish substantially with the increase of preloading. For the case with a preloading of 50% of the maximum allowable stress according to the API, the maximum impact load that is reached prior to the tube collapse drops down to nearly half for the same tube with no pre-compression applied. Likewise, the lateral displacement of the tube membrane in contact with the striker also reduces substantially. This observation is, however, different from those obtained in the present study. The results presented in Table 3-10 indicate that despite axial preloading reduces the lateral load-carrying capacity, it has little influence on the peak impact loads on the tube from ship impact. This can be attributed to the short impact duration. In other words, the peak of impact load is reached before the tube experiences significant deformation and failure. Therefore the axial preloading has little effect. The conclusion made in [88] was based on static simulation that neglects the dynamic effect. The results obtained in the present study demonstrate that dynamic effect may have a profound influence on the ship-tube structure interaction (Figure 3-27). Those based on static analysis do not necessarily provide reliable predictions.
3.5.5 Dynamic parameters

The majority of the available methods that are employed to assess the platform-ship collisions make use of quasi-static solutions due to the relatively low velocity of the ships during the impact. This seems acceptable as the local inertia effects of the structures are very low. However, the ship and tubular structure interaction during impact is a dynamic process. The material strain rate effects can be noticeable in some regions reaching plasticity even for relatively low velocity impact. The dynamic flow stress becomes larger than the static yield stress. This might lead to higher peaks of the
contact forces and shorter duration of the impact. In this section, the dynamic effects, in particular the strain rate effect on ship-tube interaction are investigated.

To model the material strain rate effect, the coefficients $C$ and $P$ from Equation 3-13 are usually set as 40.4 s$^{-1}$ and 5, respectively [103]. These values are suitable for mild steel under uniaxial tensions with small strains. In [101] it is stated that those coefficients result in a very small value for the fracture strain. Several authors have conducted many research works to model the strain rate effects and suggested different parameters. Abramowicz and Jones [109] suggested the values of $C = 6844$ s$^{-1}$ and $P = 3.91$ to be used for dynamic crushing of mild steel square tubes, whereas Yang and Caldwell [50] defined $C = 500$ s$^{-1}$ and $P = 4$ to assess the energy absorption of ship bow structures. Values in the range of 7000-10000 s$^{-1}$ and 2 to 4 respectively for $C$ and $P$ were also recommended to avoid too small values of the fracture strain [101, 110]. High strength steels might assume values of 3200 s$^{-1}$ and 5 for $C$ and $P$ [108].

These significant differences in those constants recommended by different researchers are because Equation 3-13 was derived based on uniaxial tensile tests with constant strain rate. These values are likely to be different when it comes to bending of beams or even local denting. By integrating Equation 3-13 over the cross section and deriving the dynamic bending moment, the curvature rate of the structure could be estimated [111]. Nevertheless, most of the researchers have chosen to apply Equation 3-13 to directly estimate the strain rate effects as they are most commonly adopted by different researchers. Figure 3-28 compares the force-deformation relationships of a bow crushed by a rigid column of 2 m diameter for impact speeds of 3 ms$^{-1}$ and 6 ms$^{-1}$ with and without the influence of the strain rate effects ($C = 40.4$ s$^{-1}$ and $P = 5$). As shown, neglecting the strain rate effects gives very different predictions of the impact force time history, especially when the ship velocity is relatively high. These results demonstrate the importance of considering the strain rate effect in numerical simulations.

To further study the strain rate effects, some numerical simulations are conducted here for collision with two deformable structures with different strain rate constants as listed in Table 3-11. The simulation results are shown in Figure 3-29.
Figure 3-28 Impact velocity analysis with and without strain rate effects

Table 3-11 Parameters for strain rate numerical tests

<table>
<thead>
<tr>
<th>Case</th>
<th>Tube Strain rate values</th>
<th>Bow Strain rate values</th>
<th>Impact velocity (ms(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>C (s(^{-1}))</td>
<td>P</td>
<td>C (s(^{-1}))</td>
</tr>
<tr>
<td>A</td>
<td>40.4</td>
<td>5</td>
<td>40.4</td>
</tr>
<tr>
<td>B</td>
<td>40.4</td>
<td>5</td>
<td>6844</td>
</tr>
<tr>
<td>C</td>
<td>40.4</td>
<td>5</td>
<td>500</td>
</tr>
</tbody>
</table>
Figure 3-29 Influence of strain rate model on simulation results

By comparing the cases in Figures 3-28 and 3-29, it can be concluded that it is important to correctly model the strain rate effect in numerical analysis. Using different strain rate effect models may result in different predictions. For example, the use of a low value for $C$ results in different predictions from the other cases, mainly resulting in higher peaks of the contact force and shorter duration of the impact.
3.6 Discussion and recommendations

The current design practise of offshore platforms against accidental loads such as ship impacts normally considers the events of high probability of occurrence. The deformation caused by ship impacts on jacket legs usually considered is therefore due to broadside collisions that can occur during berthing at very low speeds. For this reason no deformation on the ship structure is considered. When assessing the global deformation of tubular beams the contact area is usually assumed very small and takes place only at one single point.

In reality, the collision risk with offshore platforms might not be limited to broadside collisions. Some studies of bow collision with offshore platform structures have also been reported in the literature. The previous studies with respect to bow collisions with jacket legs at relatively high speed, however, usually neglect the interactions of bow and platform structures [100]. The platform structure is commonly assumed to be rigid and impact force is related with the bow crushing distance. Consequently, the expressions previously adopted to predict the exerted force from the ship impact might not be accurate. As presented above, the current numerical simulations that consider the ship and platform structure interaction have demonstrated the possible significant dissipation of the strain energy over the two bodies in contact, and the geometry of the tube wall deformation depends also on the bow shape and stiffness, as well as the tube geometry and stiffness, and all these parameters affect the ship-tube interaction and the impact force.

This section discusses the limitations of the current practice in predicting the impact force, deformation and energy absorption associated with ship bow impacting on tubular jacket legs, and suggests possible improvements that consider the ship and platform structure interaction for better design and protection of offshore platform structures against ship impacts.

3.6.1 Bow response

The generic bow structure considered in the study has been used by other researchers and represents the general bow structure designs in a cargo or oil tank. Therefore the numerical results presented in this study give a more precise understanding of what to expect from a jacket leg, and to some extent from a jacket platform, when subjected to
collisions involving ships of similar sizes and initial kinetic energies than those in the current design practise.

The current FE analysis results show the importance of the ship-platform interaction. The design curves currently adopted in the DnV code were obtained with the assumption of ship collision on a rigid wall, which is proven in this study overestimates the contact force, especially when large deformation of the tube and the ship bow occurs. It is also known that the collision with more flexible tubes can result in lower peak values of the contact force and longer impact duration. The numerical tests carried out in this study have shown these variations, despite small, to be more noticeable when different boundary conditions are assumed. The comparisons of the numerical simulation results with the current design curves from DnV on force and bow crushing distance are shown in Figure 3-30. It shows that, if the 'shared deformation' principle is assumed, the contact force must be related to the plastic deformations of the both bow and tube structure. Therefore, the value of the contact force that is reached until the bow starts to deform will be greater than that estimated by impacting with an idealized rigid cylinder. This value will also depend on the stiffness of the bow. The curves relating the strain energy to the bow deformation are shown in Figure 3-31. It should be noted that the suggested curves are based on the tube structures considered in the present study. They might not be extrapolated to other tubes in other offshore platform structures.

Figure 3-30 Force-deformation relationship for bulbous bow (ship displacement: 2000-5000 tons)
Nevertheless, it is recommended that the response to impact when none of the structures behaves as ideally rigid be made in terms of strain energy dissipation. Obtaining the contact force from the deformations occurred simultaneously on both tube and bow structure is a very complex, whereas the energies can be easily related to the plastic deformations of the deformable bodies. The percentages of the initial kinetic energy to be dissipated through the different structures in contact depend on the relative stiffness of the two parts. Hence, for the impact scenario under consideration, particular care should be taken in order to correctly capture the deformation modes, rupture and plasticity that influence the stiffness variation during the plastic deformation and consequently the energy dissipation.

3.6.2 Leg response

For all the cases with different tubular structures and different wall thickness, diameter, length and boundary conditions considered in the present study, it is shown that the bow shape never remains unaffected after the impact. For the same reasons mentioned in section 3.6.1, the tube response is better predicted by means of strain energy. It is confirmed that the change of geometric parameters of the pipes, as well as boundary conditions or material properties can be treated through the normalization of the energy that is absorbed by the tube. The fact that a striker, a ship bow in this case, undergoes significant deformations does not seem significantly affect the expressions already in

Figure 3-31 Energy-deformation relationship for bulbous bow (ship displacement: 2000-5000 tons)
use for the assessment of energy dissipation. On the other hand, the normalization of the strain energy by the work done by an ultimate static load must take into account the nature of the load i.e., ship impacts are better characterized by distributed loads rather than a single point load and this will reflect on the deformation behaviour/configuration of the tube. Since the DnV code [3] takes into account the length of the dented (flattened) area, and the damage assessment can be done in terms of absorbed energy, its adequacy is therefore compared here. The DnV expression for the dissipated energy can be obtained from integrating the force-deformation relationship for denting, giving:

\[
\frac{E_d}{R_c} = c_1 \cdot \left( \frac{x^{c_1+1}}{c_2+1} \right) \cdot \left( \frac{1}{D} \right) \cdot D c_1
\]

(3-25)

\[
c_1 = \left( 22 + 1.2 \cdot \frac{b}{D} \right) \cdot c_3
\]

(3-26)

\[
c_2 = \frac{1.925}{3.5 + \frac{b}{D}}
\]

(3-27)

\[
c_3 = 1.0, \quad \frac{N_{sd}}{N_{RD}} \leq 0.2
\]

(3-28)

\[
R_c = \sigma_y \cdot t^2 \cdot \frac{D}{4 \sqrt{t}}
\]

(3-29)

where \( b \) is the dent length along the tube longitudinal direction. A typical comparison for a particular tube is given in Figure 3-32. As shown the DnV equation gives very good predictions of energy absorption with respect to the local denting deformation. However, it should be noted that it is difficult to apply Equation 3-25 to predict the tube damage because the bow surface is not flat such that the contact area often cannot be properly defined. Moreover, the dented area can have not only different depths, but also discontinuities, especially at early stages of contact/bow deformation. For the studied case the estimation has been done by using an average depth derived from numerical simulation, which is usually not available in practice without performing numerical simulations.
In regards to the different deformation modes and their interaction, it has had already been verified that the absorbed energy increases nearly linearly with the increase of the maximum horizontal displacement of the tube membrane on the compressive side. This still holds true regardless of the boundary conditions. Yet, in some cases involving very long tubes and high $L/D$ ratios, it is not clear of whether the large displacements measured could be related to the normalized absorbed energies in the same way the rest of the cases are treated. On the other hand, such members are less likely to be representative of typical jacket legs, which are usually shorter in length, and the ratio between the length of the leg and the ship depth is also small.

The numerical results also show that the obtained maximum contact force before rupture is always larger than the ultimate load $P_u$ estimated based on the assumption of a concentrated point load and static analysis defined in Equation 3-12. This is because the actual case is not a point contact besides dynamic effect, which increases the material strength, as well as the ship-structure interaction. For instance, the clamped members with $P_u$ estimated as 12 MN do not collapse after the contact force has reached to a peak of 16.3 MN. The same is obtained for a pinned member with $P_u$ estimated as 5.9 MN, but the peak contact force reached 16.3 MN, indicating again that the peak force from ship impact is different from the static capacity of the tube structure owing to dynamic effect as discussed above when considering the influence of preloading on peak impact force.
Chapter 3

3.7 Summary

This chapter presents numerical simulation results of ship impacts on offshore platform tubular structures. Compared to most previous studies of ship impacts on offshore structures that assumed mainly the low velocity broadside impacts during berthing and neglected ship-platform structure interaction by assuming either ship or platform structure as rigid, the present study considered ship and platform structure interaction. It is found that both ship bow structure and platform tube structure might experience large plastic deformation during impact. Neglecting their interaction and plastic deformation of either structure might lead to inaccurate predictions of impact loads, deformation, failure modes of the tube, and energy absorption. Intensive parametric simulations are carried out to study the influences of ship bow stiffness, tube dimension, thickness, boundary conditions, preload and impact velocity on ship-platform structure interaction. The numerical results are also compared with the design equations and analytical formulae suggested by other researchers in the literature for predicting impact loads, energy absorption and structure deformation. The adequacy of the current design practice is discussed. Based on the numerical simulation results, design curves are also suggested. These design curves derived with consideration of ship-platform structure interaction are believed giving more accurate predictions of impact loads, deformation and energy absorptions.

In the next chapter the collision problem is treated for the inclusion of the entire offshore facility in the model. The main challenge resides in developing simplified analysis methods that can lead to a significant reduction in the computational requirements and easy application in engineering practice.
Chapter 4

Dynamics of Steel Offshore Platforms under Ship Impact

4.1 Introduction

The exposure of offshore platforms to ship collision is treated by the current design codes for relatively low amounts of energy that are normally related to accidental berth manoeuvring of supply vessels. Some statistics of reported incidents in Ref. [112] refer the transfer of cargo followed by vessels that approach the installation and unloading operations as the most common type of activities that lead to collisions with the offshore installations. However, incidents involving passing vessels have also been recorded in the same report. The DnV design against accidental loads [3] requires platforms to be designed for impacts with kinetic energies up to 14 MJ for side impacts or 11 MJ for head-on impacts. Recently ship collision forces from supply vessels have been estimated for energies of 55-60 MJ, owing to higher impact velocities for a typical vessel with displacement comprised between 2000 to 5000 ton. It is also possible to assume that collisions might involve heavier ships surrounding offshore installation areas [96]. The energy of the impacts is the factor normally considered for platform design. While for instance the Health and Safety Executive [113] recommends that the platform shall contribute to energy dissipation with amounts of at least 4 MJ, the DnV code follows the share of energy upon the relative stiffness of the two structures in contact. The mechanisms that contribute to energy dissipation can be divided into local denting, beam bending, frame deformation or ship deformation. Not much from the
design codes has been known with regards to beam bending due to axial loading, which represents another way of energy absorption. Some literature related to axial plastic energy of tubes can however be found in [114], [115], [116], and [117], providing some basis for hand-calculations that could meet the design requirements. The way how the different mechanisms contribute to the energy absorption will also depend on the impact point considered. The platform members that are affected by the impact, for reasons of simplicity and design purposes, are individually treated through hand calculations. While for local denting, bending or ship deformation the response is assumed to be governed by the plastic straining, for the global deformation of the platform the elastic straining might be significant. In order to resist higher energy collisions, platforms can be strengthened by increasing their members stiffness and increase the amount of deformation onto the ship as its energy absorption capacities are by far superior to the platform that the energy is mainly locally absorbed by dented legs or bent bracings, which lead to significant localized damage. For stiffer contacts with platform legs, the dynamic effects are expected to have some contribution to the platform response. The static approaches therefore have their own drawbacks. Amdahl and Eberg [118] compared the dynamic and the static effects for impacts between supply vessels and a 4-legged jacket and a jack-up platform. Some conclusions from their study pointed out the importance of the collision point as well as the impact duration and the strength of the adjacent members as factors that might differentiate the static from the dynamic approach. The considered collision cases do not include, however, joint impacts for collisions against jackets. The contact was simply modelled through a single point connected to a spring with an attached mass point, which has some limitations in terms of capturing the local deformations in the contact areas.

In this chapter, through the use of non-linear FE models, the dynamic effects of high energy ship impacts are studied and compared. Based on the intensive simulation results, simplified approaches that can be used for design purposes are suggested.

4.2 Development of an equivalent system for impact analysis

4.2.1 Background

It is common that in design structures are simplified to equivalent systems for efficient analysis. Usually some critical response quantities such as deflection of the structure are used as control parameters for deriving the equivalent system, i.e., these response
quantities calculated from the simplified equivalent system should be similar to those of the prototype structure obtained from more sophisticated analysis or experimental tests. Finite element codes represent a reliable and popular tool for structural modelling, but are very demanding in terms of computation resources and time. For complex structures subjected to impact loads, simplified models can be developed considering the specifications related to the impact energy, impact velocity or local stiffness. The deformation of the striker can also interfere with the system response in ship impacts against offshore platforms. Simplifications/specifications regarding the material behaviour are equally often assumed in the analyses.

A proper analysis of the global response of a platform subjected to ship impact is easier to be performed if the plastic strains in the contact area are not significant. In other words, the structure responses primarily in global bending should be much straightforward to be analysed than the one with significant local damage from the contact. In Ref. [119] the analysis of beams subjected to transverse impact loads from an external rigid striker is developed through pseudo-dynamic techniques, involving a SDOF system and elastic contact between the striker and the beam. For the scope of the present work, some limitations would however be found concerning the impact speed, where the propagation phenomena must be included for higher velocities since the participation of higher modes might lead to a more complex structure deformation configuration that requires the uses of a higher number of degrees of freedom for reliable analysis. The striker deformation and the possibility of small nonlinearities are some of other factors that are taken into account in the proposed simplified modelling in this study.

### 4.2.2 Formulation of equivalent system

The adopted methodology in the current work for deriving the simplified equivalent system for analysis of platform response to ship impact is based on the Rayleigh-Ritz principle [120]. The approach is widely used in assessing the structural response to blast loading [121]. The solution for the loaded structure in question needs an expected deformed shape to be selected which satisfies the specific boundary conditions relating to the displacement. The strain energy per unit volume of material is then evaluated from the deformed shape and curvature, and the total strain energy of the element is calculated by integration.
For structures formed by a significant number and diversity of elements, their somewhat high degree of complexity suggests that the structure could be represented by a reduced number of degrees of freedom in order to reduce the computation time. Fixed steel offshore platforms are among the most representative offshore structures ([7] and [122]) and they are usually simplified to a cantilever beam for design analysis to estimate their structural response to transverse loadings.

The contact between the striking ship and the fixed structure does mainly vary due to the ship loading conditions which make it drift at different vertical levels. Nonetheless, and by considering both heights of vessel and platform and the sea level, the impact load is usually taken as applied at the platform top (or very near to the top) which, for a cantilever case, corresponds to the free end. For a given cantilever beam of length $L$, subjected to a transverse load $P$ at its free end (Figure 4-1), the approximate configuration of the elastic deformation, neglecting the transverse shear and the rotary inertia, can be easily obtained by integration of the bending moment equation as:

\[
EI \cdot u(z) = - \int_0^L \int_0^L (P \cdot z - P \cdot L) dz dz \tag{4-1}
\]

where $EI$ represents the flexural stiffness of the beam ($E$ – Young Modulus; $I$ – second moment of area of the beam cross section), $z$ is the distance from the cantilever root and the constants resulting from the integration that are related to the boundary conditions of the beam are zero-valued for the illustrated case. The deformation of the beam, thus given as:

\[
u(z) = \frac{1}{EI} \left[ - \frac{Pz^3}{6} + \frac{PLz^2}{2} \right] \tag{4-2}
\]
can be normalized by its maximum displacement, measured at the free end

\[ u(L) = \frac{PL^3}{3EI} \]  

(4-3)

as:

\[ \phi(z) = \frac{u(z)}{u_{\text{max}}} = \frac{u(z)}{u(L)} = -\frac{1}{2} \left( \frac{z}{L} \right)^3 + \frac{3}{2} \left( \frac{z}{L} \right)^2 \]  

(4-4)

The equation of the elastic deflection can be re-written in the form:

\[ u(z) = \phi(z) \cdot u_0 \]  

(4-5)

in which \( u_0 \) is the free-end displacement.

The evaluation of the work done, strain and kinetic energy for the beam is then obtained as follows:

\[ W = \int_0^L p(z) \cdot u(z) \, dz = \sum_{i=1}^{n} P_i(z_i) \phi(z_i) u_0 = \alpha_1 \cdot Pu_0 \]  

(4-6)

\[ E_s = \int_0^L \frac{M(z)^2}{2EI} \, dz = \frac{EI}{2} \int_0^L \left[ \frac{d^2u(z)}{dz^2} \right]^2 \, dz = \frac{\alpha_2 Eiu_0^2}{L} \]  

(4-7)

\[ E_k = \frac{1}{2} \int_0^L \rho A \dot{u}(z)^2 \, dz = \alpha_3 L \cdot \rho A\dot{u}_0^2 = \alpha_3 M\ddot{u}_0^2 \]  

(4-8)

where \( p \) would represent the load distribution. As it will be shown later, the ship contact load can be accurately replaced by a set of single point loads \( P_i \) or even its resultant \( P \).

The coefficient \( \alpha_1 \) depends on the point where \( P \) is applied. Likewise, the values \( \alpha_2 \) and \( \alpha_3 \) would be \( \frac{1}{2} \) and 0.118 for this case. The parameters \( M, \rho, A, \dot{u} \) and \( m \) correspond to the bending moment, the material density, the area of the beam cross section, the velocity and the total mass of the system, respectively. For an equivalent single degree of freedom system (Figure 4-2) defined by its mass and stiffness \( m_e \) and \( K_e \), and with the same maximum displacement \( u_0 \) and maximum initial velocity \( \dot{u}_0 \) as the real structure, caused by the loading action \( P_e \), Equations 4-6 – 4-8 can be simply expressed as:
The factors for load, stiffness and mass can be obtained by equating Equations 4-6 – 4-8 to Equations 4-9 – 4-11, giving

\[ \Phi_L = \frac{P}{P} = \alpha_1 \]  
(4-12)

\[ \Phi_K = \frac{K_e}{K} = \frac{2 \cdot \alpha_2}{3} \]  
(4-13)

\[ \Phi_m = \frac{m_e}{m} = 2 \cdot \alpha_3 \]  
(4-14)

The response of the (assumed) undamped SDOF system is then obtained by solving the equation of motion:

\[ m_e \ddot{u} + K_e u = P_e(t) \]  
(4-15)

also expressed as

\[ m_e \ddot{u} + R_e(u) = P_e(t) \]  
(4-16)
where \( R_e \) stands for the equivalent resistance function and the stiffness \( K_e \) has a constant value.

### 4.2.3 Nonlinear numerical models

The consistence of the methodology previously described for application on impact problems is checked for a set of numerical non-linear finite element simulations in this section. Initially, a steel tube with diameter \( D = 1.7 \) m, length \( L = 25 \) m and wall thickness of \( t = 60 \) or \( 100 \) mm is considered. The tube is meshed with shell elements of \( 100 \) mm of size. The impacts are simulated by taking into account a rigid sphere of \( 6 \) m of diameter and a deformable bow-model with the same layout as that adopted in chapter 3 (see also the Appendix – ‘Ship S2’). The ship-bow is equally discretised with \( 100 \) mm size shell and rigid blocks for the rear part to comply with the inertia effects and the representation of the ship true-to-shape. The mechanical properties of both the tube and ship bow are given in Table 4-1, where for the ship model \( \sigma_u \) is increased and the strain rate effects are considered, as discussed in chapter 3. The calculated bow deformation behaviour upon the penetration of a rigid tube (\( D = 2.0 \) m and longer than the height of the bow) in line with the longitudinal axis of the ship at an initial speed \( v_0 \) of \( 6 \) ms\(^{-1} \) is plotted in Figure 4-3. It must be noted that due to the long period of the ship surge motion when compared to the duration of contact force the imparted contact force is mainly function of the bow crushing depth and hence governed by the initial kinetic energy of the ship structure (impulse). It should also be noted that for the range of critical fracture strain values \( \varepsilon_u \) defined for FE structural modelling the mesh size, plate thickness and strain rate parameters (\( C \) and \( P \) constants) are taken into account. The mesh is primarily scaled based on convergence tests, as discussed in chapter 3 that also take into consideration the contact type between the different surfaces of the bodies. The critical fracture strain values, calculated assuming material model III as defined in [101], [102] are therefore a few times larger than the nominal values usually assumed between 0.15-0.2 [3].
Table 4-1 Mechanical properties of steel materials used in numerical tests

<table>
<thead>
<tr>
<th>Part description</th>
<th>E [GPa]</th>
<th>$\sigma_y$ [MPa]</th>
<th>$\sigma_u$ [MPa]</th>
<th>$E_u$</th>
<th>$\nu$</th>
<th>$\rho$ [Kg/m$^3$]</th>
<th>C [s$^{-1}$]</th>
<th>P [10$^3$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship bow</td>
<td>200</td>
<td>285</td>
<td>500</td>
<td>0.22</td>
<td>0.3</td>
<td>7800</td>
<td>500 [103]</td>
<td>4 [103]</td>
</tr>
<tr>
<td>Tube</td>
<td>200</td>
<td>345</td>
<td>450</td>
<td>0.20</td>
<td>0.3</td>
<td>7800</td>
<td>40.4 [50]</td>
<td>5 [50]</td>
</tr>
</tbody>
</table>

Figure 4-3 Deformation behaviour of ship bow model (‘Ship S2’)

Because the resistance of the single tube is low due to its relative slenderness when compared to any point of a whole offshore platform model, a reduced amount of kinetic energy was set for the ship at this stage in order to avoid the tube collapse. The total mass of both the ship and the sphere strikers is adjusted according to the density of the solid elements that constitute the rigid sphere and the rigid rear part of the ship free of any deformation. The striking structures are then set a total mass of nearly 3000 ton and an initial speed of 2 ms$^{-1}$, making up an initial kinetic energy of approximately 6 MJ that can be converted into strain energy during the impact. From Figure 4-3 it can be predicted that the maximum peak force for a given initial energy amount of the system of 6 MJ is not expected to be larger than 13-14 MN so that the tube shall be capable of withstanding the impact without rupture. In fact, the flexibility of the deformable tube results in reduction of the peak force and increases the duration of the impact load. Similarly, the impact caused by the rigid sphere striker leads to higher amplitude but shorter duration of the impact force as compared to that from a deformable bow. The strikers were positioned relatively to the column in a way that they could be assimilated to a real impact scenario involving a real offshore facility, i.e., in the upper part of the single column, closer to its free end. It is known, for the deformable ship bow, that the
distributed contact load generates two concentrated stress areas in the hit structure that correspond to the bulb and the upper deck. The two contacting areas will contain respectively ~70% and ~30% of the total contact force as discussed in chapter 3. Thus, the centroid of the equivalent single ship loading $z_{P_{bow}}$ can be estimated as shown in Figure 4-4:

![Figure 4-4 Scheme for estimation of equivalent ship single point load position, $z_{P_{bow}}$](image)

The parameters of this set of numerical simulations are presented in Table 4-2. At the free end of the steel tube, with exception for the first test, a rigid block is placed to simulate the lumped mass supported by the tube, with the shell nodes from the top of the tube being rigidly connected to the top block. For the cases under analysis in this section, however, a very low value of the mass density of the block is set to minimize the inertia effects. Without loss of generality, the bottom of the tube is assumed to be fixed and the total length of the tube is 25 m.
Table 4-2 Data for cantilever tubes hit by different striker types

<table>
<thead>
<tr>
<th>Case</th>
<th>$t$ [mm]</th>
<th>$D$ [m]</th>
<th>Mass [kg]</th>
<th>Mass ratio ($m_{top}/m$)</th>
<th>Type</th>
<th>$v_0$ [m/s]</th>
<th>$E_{k0}$ [MJ]</th>
<th>$z_{PBW}$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>100</td>
<td>1.7</td>
<td>0</td>
<td>0</td>
<td>Sphere</td>
<td>2</td>
<td>5.78</td>
<td>16</td>
</tr>
<tr>
<td>1A</td>
<td>100</td>
<td>1.7</td>
<td>28576</td>
<td>0.29</td>
<td>Sphere</td>
<td>2</td>
<td>5.78</td>
<td>16</td>
</tr>
<tr>
<td>1B</td>
<td>100</td>
<td>1.7</td>
<td>28576</td>
<td>0.29</td>
<td>Ship bow</td>
<td>2</td>
<td>6</td>
<td>15.14</td>
</tr>
<tr>
<td>1C</td>
<td>60</td>
<td>1.7</td>
<td>28576</td>
<td>0.42</td>
<td>Ship bow</td>
<td>2</td>
<td>6</td>
<td>15.14</td>
</tr>
</tbody>
</table>

For the cases listed in Table 4-2, the normalized deformation shapes of the tubes corresponding to a single point load applied at the free end can be better approximated by defining the coefficients $\beta_1, \beta_2, \beta_3$ of Equation 4-17, in the form:

$$
\phi(z) = \beta_1 \left(\frac{z}{L}\right)^3 + \beta_2 \left(\frac{z}{L}\right)^2 + \beta_3 \left(\frac{z}{L}\right)
$$

(4-17)

It is also important to take into account that strong impacts will lead the structures to deform beyond their elastic limits. The elastic perfectly plastic material behaviour is therefore assumed for $R_e$ (Figure 4-5):

$$
R_e = \begin{cases} 
K_e u & u \leq u_y \\
\frac{P_y}{u} & u > u_y 
\end{cases}
$$

(4-18)

![Figure 4-5 Resistance function of SDOF system](image-url)
where $P_y$ defines the maximum resistance. The way how $P_y$ is defined depends on where in the structure yielding firstly occurs. For the cases being analysed in this section, it is expected that plastic hinges might be formed either from membrane yielding at the contact between tube and striker or at the base of the cantilever. Since the tube walls are given enough thickness to prevent significant crumpling that could be induced during the contact (estimations done in chapter 3), the maximum resistance is estimated based on the plastic moment of the section at the bottom of the tube.

The SDOF approach adopted to evaluate the response of a more complex structure subjected to an impact loading action suggests that the behaviour can be approximated by using a specific deformation shape that, multiplied by the mass distribution function $\bar{m}(z)$, becomes nearly proportional to the loading action. From Figure 4-5, it has:

$$y = y_e y_u \frac{K}{P_e}$$

(4-19)

or

$$P_y = K \phi(z/L) Y$$

(4-20)

by using the modal coordinates. The plastic moment of the cross section in this case is estimated as:

$$M_y = \frac{d^2 \phi(0)}{dz^2} EI \cdot Y$$

$$\Rightarrow M_y = \frac{d^2 \phi(0)}{dz^2} EI \cdot \frac{P_y}{K \phi(z/L)}$$

(4-21)

Several studies have addressed the quantification of the tubes resistance subjected to bending. By defining an elastic perfectly plastic behaviour for the material undergoing bending and transverse loading, to find the exact value for the ‘maximum resistance’ is not always straightforward. For instance, factors such as the $D/t$ ratio and energy dissipation characteristics are of significant importance. According to [123], the failure mode for $D/t$ higher than 35 will be an inward buckling on the compressive side of the pipe as the hoop strength will be too low if compared to the tensile strength, while for
lower $D/t$ ratios the failure initiates on the tensile side of the pipes due to stresses at the outer fibres that exceed the longitudinal stress limit. By conducting a theoretical study of hollow steel tubes with $D/t$ comprised between 20 and 40 and comparing them with test results, different plastic mechanisms are analysed in [124]. The bending capacity is regarded for the ovalisation plateau that follows the elastic behaviour during the bending process. In [125], the theoretical analysis of the elastic-plastic bending of tubes is carried out providing quantitative methods for predicting the spring-back behaviour and the residual stress distributions. The comparison with experimental data takes into account materials with different work hardening characteristics. The fully plastic moment of the tubular cross section for thin walled members ($D>>t$) is normally given by ([126]):

$$M_p = \sigma_y D^2 t$$

(4-22)

Because the use of empirical functions $\sigma = f(\varepsilon)$ in deriving the yielding bending moment it can normally produce very complex expressions or overestimate the structure capacity (Equation 4-22), the product of the pipe section modulus by the yield stress is adopted:

$$M_y = \left[\frac{\pi\left(D^4 - (D - 2t)^4\right)}{32D}\right] \sigma_y$$

(4-23)

Also for reasons of simplicity, the strain hardening effects are neglected. The approximated tangent moduli $E_t$ usually adopted in bi-linear force-deformation curves of steel materials is very low compared to the elastic modulus so that the resistance function shown in Figure 4-5 is widely used. The effects of the strain hardening have already been approached in the previous chapter for impact on steel tubular members.

The parameters needed to solve Equation 4-16 for cases 0-1C are defined in Table 4-3. The equivalent mass and stiffness values are obtained considering the cantilever equivalent system as defined in Figure 4-1, and are evaluated through Equations 4-6 to 4-14. The horizontal predicted displacements at the cantilever free end by the equivalent SDOF approach are compared to those obtained from the FE simulations in Figure 4-7. The contact loads for each of the cases obtained from the FE calculations are shown in Figure 4-6, from which the equivalent load to be used for the evaluation of the
Chapter 4

equivalent SDOF is derived by multiplying the curves in Figure 4-6 by the respective load factors given in Table 4-3.

Table 4-3 Data for deriving the equivalent SDOF system of cantilever tubes hit by different striker types

<table>
<thead>
<tr>
<th>Case</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\beta_3$</th>
<th>$\Phi_L$</th>
<th>$\Phi_K$</th>
<th>$\Phi_M$</th>
<th>$K_e$ [MN/m]</th>
<th>$M_e$ [kg*10^3]</th>
<th>$P_y$ [MN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-0.85</td>
<td>1.80</td>
<td>0.04</td>
<td>0.54</td>
<td>1.09</td>
<td>0.26</td>
<td>13.97</td>
<td>25.96</td>
<td>4.85</td>
</tr>
<tr>
<td>1A</td>
<td>-0.85</td>
<td>1.80</td>
<td>0.04</td>
<td>0.50</td>
<td>1.09</td>
<td>0.26</td>
<td>13.97</td>
<td>54.54</td>
<td>4.85</td>
</tr>
<tr>
<td>1B</td>
<td>-0.88</td>
<td>1.81</td>
<td>0.06</td>
<td>0.50</td>
<td>1.09</td>
<td>0.27</td>
<td>14.74</td>
<td>53.57</td>
<td>5.35</td>
</tr>
<tr>
<td>1C</td>
<td>-0.88</td>
<td>1.81</td>
<td>0.06</td>
<td>0.50</td>
<td>1.09</td>
<td>0.27</td>
<td>9.95</td>
<td>44.73</td>
<td>3.42</td>
</tr>
</tbody>
</table>

Figure 4-6 Force-time history of loading for cases 0-1C
Since no local plastic straining at the contact takes place for the situation in which the pipe is struck by the rigid sphere, i.e., cases 0 and 1A, the loading is characterized by oscillations due to the stiff nature of the contact and consequent rebounds from the vibrations. From the displacement time histories and the residual displacements shown in Figure 4-7, it can be concluded that the base of the tube yields in the two cases.

For cases 1B and 1C, in which the striker initiates plastic deformation during the contact, the loading naturally assumes a different configuration, having a lower loading amplitude and longer duration. As shown in Figure 4-7, the displacement responses predicted by the SDOF analysis agree reasonably well with the FE analysis for cases 0, 1A and 1B during the loading phase, whereas the SDOF analysis underestimates the displacement response during the loading phase in case 1C as compared with the finite element simulations. This is because of the lower stiffness of the tube with 60 mm wall thickness (longer period), in which the wave propagation over the tube is more prominent at the very beginning of the contact, which is not captured by the equivalent
SDOF analysis. To improve the prediction accuracy of the equivalent SDOF system, the estimation of the equivalent parameters of the SDOF system would have then to consider the dynamic deformation shape during the early contact to derive $K_e$ and $M_e$. The influence of any higher mode governing the platform response would therefore be accounted. This is further discussed in the next section.

The predicted displacement responses in the free vibration phase, i.e. after the action of impact force, by the SDOF system implies the SDOF system has a higher stiffness than the FE model since the response oscillates at a higher frequency. In fact, the free vibrations of the equivalent SDOF system and the FE model of the tubes are governed by the first mode of the tube defined by Equation 4-4. In FE analysis, the formation of the plastic hinges at the tube fixed end during the loading phase, especially in case 1C, results in a reduction in the tube stiffness, hence the vibration frequency. This, however, is not captured by the equivalent SDOF system. These results suggest that for the evaluation of the response of a structure subjected to dynamic loads the use of the elastic deformation shape or a particular vibration mode could not be fully adequate because structural yielding changes the deformation shape. Moreover, the primary structural deformation shape in the impact loading phase and in the free-vibration phase are not necessarily the same. All these therefore affect the accuracy of the subsequent simulations based on the original equivalent model.

4.2.4 Dynamic effects

As discussed in the previous section, the assumption of a deformation shape proportional to a cantilever static elastic deflection under an assumed transverse force can become inappropriate for impact cases, especially when the inertia effect is prominent. This is more likely to happen when the concentrated mass at the free end of the cantilever increases. Fixed steel offshore platforms can actually bear heavy deck weights, having different deck to supporting frame mass ratios according to the design conditions for operations and the frame configuration (depth, geometry, frame type, etc.).

For a cantilever beam with a very large mass concentrated at the free end being hit by a striker, the member response is expected to be governed by higher modes at the beginning owing to the large inertial resistance that resists the displacement of the free
end. Therefore the maximum displacement can occur in the span rather than at the beam ends, with the location also depending on where the load is applied. With the cantilever vibration, the deformation mode gradually shifts to the fundamental mode of the cantilever with the maximum transverse deformation occurring at the free end.

In order to try to estimate the structure response by including the inertia effects caused by the respective mass distribution, it is necessary to find an approximate deformation mode for the equivalent SDOF system during the loading phase, knowing that the cantilever’s first mode is not appropriate. Consideration must then be given to the boundary conditions at the free end. For instance, by assuming the beam to be fixed at one end and constrained against rotation on the other, with a single transverse load applied at the free end the elastic deformation is given as:

\[
\begin{align*}
\phi(z) &= -2 \left( \frac{z}{L} \right)^3 + 3 \left( \frac{z}{L} \right)^2 \\
\end{align*}
\]  \hspace{1cm} (4-25)

The same can also be obtained by replacing the support reaction by a concentrated moment of equal value, where the desired bending moment redistribution can be achieved. Equations 4-24 and 4-25 would be similar to a scenario where the rotation at the tube free end is null although the transverse displacement is allowed. It is perceptible that the coefficients $\beta_1$ and $\beta_2$ are greater in magnitude, leading to a higher equivalent stiffness based on the evaluation of the kinetic energy as described above. However, the response of a cantilever during the first stages of the forced vibration phase might be governed by modes that would require a curvature differing from that one described in Equation 4-25. Considering a cantilever beam subjected to the following actions (Figure 4-8):
where the moment $M_2$ can be related to the reaction for rotation at the end 1, $M_1$, as:

$$\frac{M_2}{M_1} = \mu$$ \hspace{1cm} (4-26)

The evaluation of the elastic deformation can thus be performed as follows:

$$EI \cdot u(z) = -\int_0^L \int_0^L \left( P \cdot z - \frac{P \cdot L}{1 + \mu} \right) dz \, dz$$ \hspace{1cm} (4-27)

$$u(z) = \frac{P}{EI} \left( -\frac{z^3}{6} + \frac{Lz^2}{2(1 + \mu)} \right)$$ \hspace{1cm} (4-28)

$$\phi = \frac{\mu + 1}{\mu - 2} \left( \frac{z}{L} \right)^3 - \frac{3}{\mu - 2} \left( \frac{z}{L} \right)^2$$ \hspace{1cm} (4-29)

and the equivalent flexural stiffness given in the form:

$$K = \frac{\xi EI}{L^3}$$ \hspace{1cm} (4-30)

with $\xi$ equated to the evaluation of the deformation as done in Equation 4-7.

Different deformation modes are illustrated in Figure 4-9 and the relationship between the parameters $\xi$ (evaluated according to Equations 4-7 and 4-10), $\beta_1$ and $\beta_2$ are listed in Table 4-4 for some $\mu$ values:
Table 4-4 Beam flexural stiffness variation for different rotation restraints at the transversely loaded end

<table>
<thead>
<tr>
<th>( \mu )</th>
<th>( \beta_1 )</th>
<th>( \beta_2 )</th>
<th>( \xi )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-0.5</td>
<td>1.5</td>
<td>3.00</td>
</tr>
<tr>
<td>0.2</td>
<td>-0.67</td>
<td>1.67</td>
<td>3.12</td>
</tr>
<tr>
<td>0.4</td>
<td>-0.88</td>
<td>1.88</td>
<td>3.58</td>
</tr>
<tr>
<td>0.6</td>
<td>-1.14</td>
<td>2.14</td>
<td>4.64</td>
</tr>
<tr>
<td>0.8</td>
<td>-1.5</td>
<td>2.5</td>
<td>7.00</td>
</tr>
<tr>
<td>1</td>
<td>-2</td>
<td>3</td>
<td>12.0</td>
</tr>
<tr>
<td>1.1</td>
<td>-2.33</td>
<td>3.33</td>
<td>16.4</td>
</tr>
<tr>
<td>1.2</td>
<td>-2.75</td>
<td>3.75</td>
<td>23.3</td>
</tr>
<tr>
<td>1.3</td>
<td>-3.29</td>
<td>4.29</td>
<td>34.1</td>
</tr>
<tr>
<td>1.4</td>
<td>-4</td>
<td>5</td>
<td>52.0</td>
</tr>
<tr>
<td>1.5</td>
<td>-5</td>
<td>6</td>
<td>84.0</td>
</tr>
<tr>
<td>1.6</td>
<td>-6.5</td>
<td>7.5</td>
<td>147</td>
</tr>
<tr>
<td>1.7</td>
<td>-9</td>
<td>10</td>
<td>292</td>
</tr>
<tr>
<td>1.8</td>
<td>-14</td>
<td>15</td>
<td>732</td>
</tr>
<tr>
<td>1.9</td>
<td>-29</td>
<td>30</td>
<td>3252</td>
</tr>
</tbody>
</table>

The possible different deformation shapes considered in Figure 4-9 and Table 4-4 indicate that the dynamic reaction caused by the inertial resistance of the heavy tip mass would gradually decrease with the curvature configuration change during the forced vibration. The definition of a deformation shape for this stage suggests that the chosen mode could give a good estimation of the structure response governed by a high mode.

The evaluation of the dynamic response of the cantilever beam with high tip mass under transverse loading through the SDOF approach is then tried to be separated into two different and independent steps that could nearly be related to the forcing vibration and
free vibration. This implies that for the second stage of the response of the SDOF approach the respective adopted mode must be given the necessary initial conditions from the end of the first stage at the transition time \( t_m \). Therefore:

\[
Y^{II}(t_m) = \frac{\int_0^t \overline{\omega}(z) \phi^{II}(z)u(t_m)dz}{M_c^{II}} \tag{4-31}
\]

\[
\dot{Y}^{II}(t_m) = \frac{\int_0^t \overline{\omega}(z) \phi^{II}(z)\dot{u}(t_m)dz}{M_c^{II}} \tag{4-32}
\]

where \( Y^{II} \) and \( \dot{Y}^{II} \) are the initial displacement and initial velocity of the second, i.e., free vibration phase, and \( \phi^{II} \) and \( M_c^{II} \) represent its respective normalized deformation shape and equivalent mass (calculated according to the steps described in section 4.2.2). Since the decks of steel fixed offshore platforms contain the majority of the total mass of such structures, it is normally true that \( Y^{II}(t_m) \approx u(t_m) \) and \( \dot{Y}^{II}(t_m) \approx \dot{u}(t_m) \). The load factor corresponding to the mode shape at the second phase may also be used to derive the equivalent loading function in case \( t_m < t_d \), where \( t_d \) is the impact load duration.

For the same numerical models the top mass of the cantilever is adjusted to investigate the influences of different mass ratios on the response characteristics. The limits defined in the API [1] for the pre-axial force of 50% and 100% of the maximum admissible compressive stress \( F_a \) are taken as the reference to define the mass ratio range in the study, although the mass ratios for a single tube considered in the study may lie above those usually known for steel framed platforms. For cases 1D to 1H, listed in Table 4-5, ship impacts against the tube at the same vertical position as in cases 1B and 1C are considered.

### Table 4-5 Data for ship impact against cantilever tubes with heavy top mass

<table>
<thead>
<tr>
<th>Case</th>
<th>Tube</th>
<th>Striker</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>t [mm]</td>
<td>D [m]</td>
</tr>
<tr>
<td>1D</td>
<td>60</td>
<td>1.7</td>
</tr>
<tr>
<td>1E</td>
<td>100</td>
<td>1.7</td>
</tr>
<tr>
<td>1F</td>
<td>100</td>
<td>1.7</td>
</tr>
<tr>
<td>1G</td>
<td>100</td>
<td>1.7</td>
</tr>
<tr>
<td>1H</td>
<td>45</td>
<td>1.7</td>
</tr>
</tbody>
</table>
Differing from the other cases listed in Table 4-5, it is expected that in case 1H the cantilever response would be highly affected by crumpling at the contact owing to the thinner wall thickness of the tube, thus requiring a different way of estimating $P_y$. It is however verified for the same case that, despite the plastic hinge initially formed at the contact point, the structure fails due to the collapse of the hinge formed at the fixed support. This is because the maximum capacity of the tube is not sufficient to resist the moment generated by the impact.

The contact loads plotted in Figure 4-10 for cases 1D-1G show there is no significant difference between the cases where no local denting of the tube is measured in the struck membrane of the tube and the impact energy is kept constant, i.e., cases 1E, 1F and 1G regardless of the top mass or the structure stiffness. The evaluation of the approximated deformation shapes of the tube with time during and right after the contact is shown in Figure 4-11 and the estimation of the respective equivalent stiffness can be seen in Figure 4-12, following Equations 4-7 and 4-29. The tube deformation is measured assuming the horizontal displacement of different selected nodes along the tube length. The relatively very high curvature rates experienced at the very beginning of the impact are easily explained due to some imprecision in evaluating the normalized deformed shape as the value of the transverse displacement measured at the top is infinitesimal, and with an order of magnitude smaller than the maximum horizontal deflection observed near the loaded area. As long as the curvature changes its configuration with time, closer to the elastically loaded cantilever, the equivalent stiffness decreases. Such decrease appears to be more or less linear after 0.2 s the contact has started and until it reaches $\zeta \approx 3$. For cases 1D-1G the deformation and the respective parameter $\zeta$ are measured along the simulation time, up to 10-12 s. It is found that, even for the free vibration period, despite relatively closer to the reference value of 3, the parameter $\zeta$ suffers some oscillation around this value. The period of the free vibration oscillations observed in Figure 4-13 points out that the equivalent stiffness $K_e$ considered can actually be smaller than the static stiffness of a cantilever transversely loaded at the free end. In fact, for cases 1D-1G the yield stress is reached at the base, with plastic hinges being formed in the compressed fibres, and in the form of local buckling at the back surface for case 1D ($D/t = 28.3$, lower than the value indicated in [123]). For this reason, and since the cross-section at the yielding zone does not recover
totally, i.e., the cross section is only partially effective, the reduction of the stiffness results in a longer period of the structure equivalent mode.

The procedure used to obtain the pair of values $u_0$ and $\dot{u}_0$, necessary for the evaluation of the response of the SDOF system at the second stage of the impact, consists of comparing the displacement and velocity curves measured from the numerical model at the free end to those estimated by assuming the different mode shapes observed during the forced vibration period. While very high $\zeta$ values may lead to a conservative solution, endowing the system with very high velocity and displacement values as the load factor is equally great the reverse can result in underestimation of the structure response. The data taken for the different cases is shown in Table 4-6.

![Figure 4-10 Force-time history of loading for cases 1D-1H](image-url)
Figure 4-11 Normalized deformation shapes at different stages for cases 1D-1G

Figure 4-12 Evaluation of system equivalent stiffness with time from deformation configuration
Table 4-6 Parameters related to the double step evaluation of SDOF response

<table>
<thead>
<tr>
<th>Case</th>
<th>$\xi$</th>
<th>$t_m$</th>
<th>$u_0$</th>
<th>$u_0$</th>
<th>$T$</th>
<th>$t_d$</th>
<th>$t_m/t_d$</th>
<th>$T/t_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1D</td>
<td>13.1</td>
<td>0.78</td>
<td>0.50</td>
<td>1.12</td>
<td>5.35</td>
<td>0.94</td>
<td>0.83</td>
<td>5.69</td>
</tr>
<tr>
<td>1E</td>
<td>3</td>
<td>-</td>
<td>0*</td>
<td>0*</td>
<td>5.28</td>
<td>0.84*</td>
<td>0*</td>
<td>6.29</td>
</tr>
<tr>
<td>1F</td>
<td>9.5</td>
<td>0.80</td>
<td>0.45</td>
<td>0.30</td>
<td>7.46</td>
<td>0.94</td>
<td>0.85</td>
<td>7.94</td>
</tr>
<tr>
<td>1G</td>
<td>6.4</td>
<td>0.96</td>
<td>0.16</td>
<td>0.62</td>
<td>5.28</td>
<td>1.18</td>
<td>0.81</td>
<td>4.47</td>
</tr>
</tbody>
</table>

*a single step calculation is used for this case

Figure 4-13 Displacement response time histories obtained by FE analysis and SDOF system (cases 1D-1G)

It is a common aspect in cases 1D-1G that the ratios between $t_m$ and the loading duration are close in value and that the transition between the two stages defined for the SDOF approach takes place close to the end of the contact or loading phase, during the descending trend of the loading curves. In other words, the first mode can be assumed to govern the structure response even before the free vibration period starts. For case 1E, however, the equivalent stiffness of the deformation mode found to provide an accurate
response prediction in the earlier contact stage is close to the first mode, so that a two-step calculation process is not required. Even though the deformation shapes in Figure 4-11 show significant changes with time, they converge after the first time steps and therefore the deformation shapes to be used will be normally similar to the configuration of the column observed during the last stages of the forced vibration phase (Table 4-6). The difference for case 1G, in which the same tube with the same loading conditions as case 1E is considered, is the impact velocity of the striker. Although for the cases in general the SDOF curves can satisfactorily match the numerical predictions in terms of the maximum and residual displacements, the SDOF approach using a two step-calculation shows an offset of the time-displacement curve relatively to the FE numerical results during the first displacement cycle, when the fundamental period of the structure is higher (particularly in case 1F). This issue, however, does not occur when the impact velocity changes (cases 1E and 1G), where only the deformation induced by the greater amount of energy imparted to the tube results in a higher value for $\xi$ and the plastic deformation is higher, hence resulting in a greater residual displacement.

As referred before, the formation of some plastic strain at the tube fixed support reduces the capacity of the member and its stiffness. An equivalent stiffness of 80% of that obtained for $\xi = 3$ (first mode of cantilever) is assumed for all the cases for the second step of the dynamic SDOF evaluation, having the comparisons between the two curves in Figure 4-13 shown good consistence in terms of the period and amplitude of the free-vibrations and the residual displacements (when combined with the first calculation step) of the total response.

The single column model shows the effects the inertia of the tip mass can have for quick impacts and this shall be accounted for multi-leg systems, as shown in the following section.

### 4.3 Case studies of typical steel offshore platform models

The evaluation of the response of a platform model involving a frame instead of a single column has its own particularities, if the principles described in section 4.2 are to be followed. First of all, the evaluation of the strain energy has to take into account the deformation mechanisms of, for example, bracings, which may include shear and axial stresses. The structure might still experience some twisting over its vertical axis,
depending on whether the impact direction is or not in line with the centroid of the platform in the horizontal plan. This is, in fact, a very realistic scenario observed in the majority of such impact cases. Moreover, the elastic perfectly plastic properties of the resistance curve that are set to calculate the response of the dynamic SDOF might also introduce errors. For instance, the determination of the maximum resistance is not as straightforward as analytically demonstrated earlier. The value for $P_y$ might be determined corresponding to the formation of the first plastic hinge, which could be related either to bending or local buckling of a leg or an adjacent brace. Depending on the redundancy of the frame, this might not always be associated with the global collapse of the structure. Hence, it is expected that for some cases the mechanisms formed will only reduce the structure overall stiffness. The use of elastic perfectly plastic materials might therefore underestimate the overall maximum capabilities of the structure to resist a leg transverse impact and overestimate the maximum and residual global deformations. Although the desired accuracy is difficult to achieve for complex steel frames, static pushover analyses performed on the structure can primarily give an idea of what to expect in terms of where the hinges are formed and the secant stiffness values that are observed throughout the deformation. However, care must be taken on how the approximate value of $P_y$ is defined. As discussed above, under impact, the deformation shape may not follow the fundamental deformation mode owing to the inertial resistance from the concentrated large mass at the deck level. This makes the determination of the distributed static load for pushover analysis difficult. If an equivalent dynamic SDOF system derived through the deck displacement accounts for a deformed shape that is different from that obtained through a simple static pushover analysis, the load, mass and stiffness factors that are based on the work done (Equation 4-6) will differ from one another, leading to inaccurate response predictions. On the other hand, if the static pushover curves are obtained for a single point load applied at the top of the structure instead of the impact point, the deformed shape of the struck leg can change and also other plastic hinges can be formed, resulting in different values of the maximum resistance for the equivalent SDOF system. Moreover, it should be noted that it is a precondition in deriving the equivalent SDOF system that the struck legs shall be locally stiff enough so that local response in the contact area is not so predominant, even some local deformations occur the overall response is predominantly global structural responses. Nonetheless, for jacket platforms, the possible impact locations are relatively limited and close to the deck. Then the selection of $P_y$ value for
the equivalent SDOF model can be based on static pushover calculations that could give an insight of the structure global deformation and the respective hinge mechanisms.

For the numerical analysis purpose of more realistically represented offshore structures, three different models are considered in the study. They are a three and a four-legged shallow water jackets and a jack-up. Different issues that associated to the structure local and overall response to large energy impacts vary from case to case. Phenomena like small local denting, brace failure due to buckling or significant overall twisting will be addressed in the following case studies.

4.3.1 Four-legged jacket (Model ‘P4’)

The tubular steel of which the structure is made of has the same properties as listed in Table 4-1. The thickness given to the tubular members is 20 mm for the braces and 60 mm for the legs in some cases. For others, the wall thickness of the legs is changed to 45 mm in order to allow some local denting. The ground and the third elevations constitute the exceptions where K-joints are used. The heights of the elevations from the first elevation to the top are, respectively, 20.7 m, 38.1 m, and 53.4 m. The joints are modelled as rigid. The deck is placed 58 m above the mud-line and its total mass is assumed to be 10000 tons. The vertical action from the top weight due to gravity is within the limits of axial stress [1]. The elements in the contact area and the adjacent members are discretised with shell elements of 100 mm of size whereas for the remaining structure the size of the elements is increased to 150 mm, the accuracy of the mesh size for this and other platforms analysed in this work was verified through convergence tests, as well as comparing the ultimate strain values with those given in [101], [102] (ranging between 0.38 and 0.52). For brevity reason the mesh convergence analysis is not presented. There are a total of 236223 elements for the steel frame of the 4-leg platform used in the FE simulations. The structural particulars of this and the other platform models can be seen more in detail in the Appendix. Five impact scenarios were defined for the four-legged platform (Figure 4-14). Cases 4LA to 4LE include both span and joint impacts, as well as the possibility of local denting of the thinner legs. The different cases also take into account the possibility of eccentric impacts and the impact orientation (Impact points II and III shown in Figure 4-14). The initial parameters defined for the numerical analysis to be carried out are given in Table 4-8 ($v_0$, $P_y$, $E_k$,
collision type, angle, leg thickness, etc.). The different contact load time histories are compared in Figure 4-15.

![Collision impact zones for 4-legged jacket](image)

**Figure 4-14 Collision impact zones for 4-legged jacket**

<table>
<thead>
<tr>
<th>Case</th>
<th>Impact Point</th>
<th>(v_0) [ms(^{-1})]</th>
<th>(E_{k0}) [MJ]</th>
<th>(l_{leg}) [mm]</th>
<th>(m_{deck}) [kg*10^6]</th>
<th>(m_{deck}/m_{frame})</th>
<th>(Z_{deck}) [m]</th>
<th>(Z_{Pbow}) [m]</th>
<th>(Z_{deck}/Z_{Pbow})</th>
</tr>
</thead>
<tbody>
<tr>
<td>4L1 (6)</td>
<td>II</td>
<td>6</td>
<td>54.6</td>
<td>60</td>
<td>9.887</td>
<td>8.93</td>
<td>58.7</td>
<td>41.4</td>
<td>0.71</td>
</tr>
<tr>
<td>4L2 (9)</td>
<td>I</td>
<td>6</td>
<td>54.6</td>
<td>60</td>
<td>9.887</td>
<td>8.93</td>
<td>58.7</td>
<td>47.4</td>
<td>0.81</td>
</tr>
<tr>
<td>4L3 (7)</td>
<td>II</td>
<td>6</td>
<td>54.6</td>
<td>45</td>
<td>9.887</td>
<td>10.55</td>
<td>58.7</td>
<td>41.4</td>
<td>0.71</td>
</tr>
<tr>
<td>4L4 (10)</td>
<td>I</td>
<td>6</td>
<td>54.6</td>
<td>45</td>
<td>9.887</td>
<td>10.55</td>
<td>58.7</td>
<td>47.4</td>
<td>0.81</td>
</tr>
<tr>
<td>4L5</td>
<td>III</td>
<td>6</td>
<td>54.6</td>
<td>45</td>
<td>9.887</td>
<td>10.55</td>
<td>58.7</td>
<td>39.0</td>
<td>0.67</td>
</tr>
</tbody>
</table>

( ) – related to the case number in chapter 5
Similarly to section 4.2.4, the deflection shape of the impacted leg is tracked during and after the impact instants and the different configurations can be seen in Figure 4-16. The following observations can be drawn:

- it is clear that not all the legs will follow exactly the same deformation configuration at the same time;
- the overall rotation experienced in eccentric impacts needs to be considered in deriving the equivalent system; for case with significant rotation, both translational and rotational degrees of freedom should be considered;
- since the flexibility of the frame is not uniform along its height and the elastic strain energy is not owed only to bending, the determination of $\Phi_k$ as from Equations 4-7, 4-10 and 4-13 can be erroneous. Due to rotation, different legs might show different deformation shapes and also deform in different directions in the horizontal plan. The stiffness of the equivalent system, i.e., lateral and torsional responses, can be estimated through the displacement measured at the top of the platform caused by a unit load in the direction of impact.
It is obvious in Figure 4-16 that the normalized deformation shapes of the struck legs are similar with time and for all the different cases, regardless of the impact direction. Some difference, however, is observed from the joint impact cases to the span impact cases, where for the first the coefficients $\beta_1$ and $\beta_2$ are lower in magnitude. According to Equation 4-7, this leads to a lower stiffness of the equivalent system. The comparison between the displacement responses obtained from the FE simulations and simplified approach is shown in Figure 4-17 and the corresponding input data are given in Table 4-8. From the comparison, it can be concluded that, unlike the case of the impact against a single column with tip mass, the response of the four-legged platform might satisfactorily be evaluated using a single-step calculation with the same deformation shape for the forced and free-vibration stages. This is because of the lower mass ratios ($m_{top}/m_{frame}$) and smaller structure period of the four-legged platform model than the
single tube model, which results in less inertia resistance effect than a single column structure. Equally negligible seems to be the influence of the denting generated on the leg walls by the impact. Dents of 20%, 14% and 7.5% of the cross section are measured from the FE simulations for cases 4L3, 4L4 and 4L5, corresponding to the estimated amounts of plastic strain energy of 2.62 MJ, 1.62 MJ and 0.61 MJ, respectively [85]. Such amounts can be significant when compared to the total strain energy taken by the whole platform (global + localized), suggesting that the calculation of the equivalent stiffness from the evaluation of the strain energy (Equation 4-7) should be affected, which is in fact observed from $\Phi_K$ values in Table 4-8. The reduction of $\Phi_K$ appears however to be very small for the measured dents and respective plastic strains. These results indicate that the equivalent SDOF analysis could lead to reasonable estimate of the global response of the structure even some local deformations at the contact area occur.

From the FE displacement-time curves, the small differences between the predicted peak values from the two approaches are observed, which can be attributed to the influences of local stiffness (joint/span and wall thickness) and vertical position of the impact.

Once the platform response appears to be governed by a single mode, some simplifications can be taken in order to empirically determine the stiffness, load, and mass factors:

- as the top mass represents no less than 90% of the total mass, and according to the deformation shapes, the influence of the frame distributed mass times the mass factor $\Phi_M$ would not significantly affect the value of the equivalent mass of the system. Such value would then not differ much from the deck mass (top concentrated mass multiplied by a factor of 1.0);
- the stiffness could therefore be estimated from the period of the free vibration and the stiffness factor $\Phi_K$ estimated through the ratio between the calculated stiffness and the equivalent stiffness from the static analysis.
- the load factor for which the contact force shall be multiplied can be taken from the deformation shape, assuming the work initially done through the struck platform leg.
### Table 4-8 Input parameters for impact with 4-legged jacket

<table>
<thead>
<tr>
<th>Case</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\Phi_L$</th>
<th>$\Phi_K$</th>
<th>$\Phi_M$</th>
<th>$K_e$</th>
<th>$M_e$</th>
<th>$I_e$</th>
<th>$P_y$</th>
<th>$M_y$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4L1</td>
<td>-3</td>
<td>4</td>
<td>0.93</td>
<td>1.08</td>
<td>0.49</td>
<td>159.0</td>
<td>-</td>
<td>10.43</td>
<td>-</td>
<td>29.1</td>
</tr>
<tr>
<td>4L2</td>
<td>-2.5</td>
<td>3.5</td>
<td>0.97</td>
<td>1.08</td>
<td>0.43</td>
<td>159.0</td>
<td>-</td>
<td>10.36</td>
<td>-</td>
<td>26.6</td>
</tr>
<tr>
<td>4L3</td>
<td>-3</td>
<td>4</td>
<td>0.93</td>
<td>1.04</td>
<td>0.49</td>
<td>131.7</td>
<td>-</td>
<td>10.35</td>
<td>-</td>
<td>28.8</td>
</tr>
<tr>
<td>4L4</td>
<td>-2.5</td>
<td>3.5</td>
<td>0.97</td>
<td>0.97</td>
<td>0.43</td>
<td>127.1</td>
<td>-</td>
<td>10.29</td>
<td>-</td>
<td>25.6</td>
</tr>
<tr>
<td>4L5</td>
<td>-2.5</td>
<td>3.5</td>
<td>0.57*</td>
<td>1.05</td>
<td>0.43</td>
<td>137.6</td>
<td>-</td>
<td>10.29</td>
<td>-</td>
<td>25.6</td>
</tr>
</tbody>
</table>

*Load factor multiplied by component factor of approximately $\sqrt{2}/2$*
For case 4LE in particular, where the ship collides the jacket leg in line with the deck diagonal (top view), the main contact takes place having the bulb aligned with the centre of the tube at the hit cross section. Since the leg is not vertical, the forecastle deck will act eccentrically on the platform and the ship slides to the right and smoothly yaws around the jacket leg that has already penetrated the bow. Despite the initial impact direction and the total contact force have the same direction of the initial impact, the maximum leg displacement mainly occurs in the direction approximately 45 degrees relatively to the initial surge motion. The load factor for this case is then adjusted not only by taking into account the lower vertical position of the centroid of the contact load, but also its component in the direction of the major displacement. In such situations, as already mentioned, some yawing may develop mainly during the contact. Design codes currently make use of constant coefficients to which marginal importance is given. Even for all the six degrees of freedom such constants can be assumed. It is known that the determination of such constants made either via energy or momentum equations provides different values. If the yaw is considered the added mass should be increased [127], but this would on the other hand over-predict the contact between ship and platform during the initial stages.

In general, the two curves for each of the cases illustrated in Figure 4-17 show good agreement. The maximum displacement occurs at the deck corner of the impact. Global deformations of other critical parts of the installation can be approximated from the evaluated point through the deformation shapes used for the struck leg, and through the centre of rotation.

4.3.2 Tripod jacket (Model ‘P3’)

The structure of the tripod with four elevations is very much alike the 4-legged jacket. Each of the three jacket legs is fixed at the base elevation. The pile foundation is not modelled in this study. The 3-legged fixed platform has four elevations. The first elevation is 29.3 m from the base elevation placed at the seabed. The second and third elevations are respectively 52.4 m and 70.9 m from the mud-line. All the elevations consist of T-bracings. Vertically, the structure is single braced, except between the seabed level and the first elevation, where X-bracing is used. Although thicker than average, and for reasons of simplicity and comparison with cases described in the previous sections, all the braces are given a wall thickness of 20 mm. The thickness of
the leg tubes is set to 60 mm. Comparing with the previous studied case, i.e., the 4-legged platform, the global stiffness of the current platform is lower not only due to the less number of legs, but also because of its vertical single bracing instead of X-joints. 45 mm tube thickness is not considered because it would result in local denting and a more localized response of the structure. Moreover, due to its small redundancy, a high energy impact caused by the ship, whether at one joint or mid span, would cause the structure global failure. The top deck, placed 86.2 m above the mud-line and modelled with rigid solid elements, is given a total mass of 8000 ton. The structure frame is modelled with shell elements of 80 mm size, making up a total of 516169 elements. According to the defined values and geometry, the structural response for a struck leg will be characterized by formation of plastic hinges at the adjacent braces. An illustration of the tripod can be seen in Figure 4-18 together with the location and orientation of the selected impact cases. The tripod structure is described more in detail in the Appendix.

![Figure 4-18 Collision impact zones for tripod jacket](image)

The parameters for the analysis of the collision cases 3J (joint) and 3M (beam) are presented in Table 4-9:
Table 4-9 Parameters used for the impact cases with tripod platform

<table>
<thead>
<tr>
<th>Case</th>
<th>Impact</th>
<th>$v_0$ [ms$^{-1}$]</th>
<th>$E_{k0}$ [MJ]</th>
<th>$t_{leg}$ [mm]</th>
<th>$m_{deck}$ [kg*10$^6$]</th>
<th>$m_{deck}/m_{frame}$</th>
<th>$Z_{deck}$ [m]</th>
<th>$Z_{Pbow}$ [m]</th>
<th>$Z_{deck}/Z_{Pbow}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>3LJ (3)</td>
<td>II</td>
<td>7.0</td>
<td>74.3</td>
<td>60</td>
<td>8.000</td>
<td>8.55</td>
<td>87.9</td>
<td>57.0</td>
<td>0.65</td>
</tr>
<tr>
<td>3LM (1)</td>
<td>I</td>
<td>7.0</td>
<td>74.3</td>
<td>60</td>
<td>8.000</td>
<td>8.55</td>
<td>87.9</td>
<td>62.4</td>
<td>0.71</td>
</tr>
</tbody>
</table>

( ) – refers to the case number in chapter 5

Again, some amount of rotation is expected for case 3LJ whereas for case 3LM the deck shall practically shift according to the impact motion. From the bow deformation curves (Figure 4-3) and non-linear preliminary static analysis of the tripod for each or the impact cases, it is also expected that, for the joint impact, the brace between the second and the third elevation of the platform may fail due to buckling. This is also expected to take place quickly and before the contact load reaches its peak. The equivalent stiffness for this case is then obtained for the platform with the respective member removed. Similar observations as done for the 4-legged jacket are shown for the tripod in Figures 4-19 to 4-21. The parameters for evaluating the structure response through the simplified system are given in Table 4-10.

Figure 4-19 Load time-history for tripod jacket impact cases
### Table 4-10 Input parameters for impact with tripod jacket platform

<table>
<thead>
<tr>
<th>Case</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\Phi_L$</th>
<th>$\Phi_K$</th>
<th>$\Phi_M$</th>
<th>$K_e$</th>
<th>$M_e$</th>
<th>$I_x$</th>
<th>$P_y$</th>
<th>$M_y$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[kg*10^6]</td>
<td>[MN/m]</td>
<td>[MN.m/rad]</td>
<td>[kg*10^6]</td>
<td>$[kg^\circ.m^2]$</td>
<td>[MN]</td>
<td>[MN.m]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3LM</td>
<td>-0.85</td>
<td>1.85</td>
<td>0.63</td>
<td>1.163</td>
<td>0.268</td>
<td>35.02</td>
<td>-</td>
<td>8.251</td>
<td>17.63</td>
<td>-</td>
</tr>
<tr>
<td>3LJ</td>
<td>-3</td>
<td>4</td>
<td>0.86</td>
<td>1*</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>3785</td>
<td>1270</td>
<td>220.4</td>
</tr>
</tbody>
</table>

*Applied to rotational stiffness

---

A single-step calculation looks again to be acceptable in predicting the system response even though the tripod has a relatively long period. For case 3LJ, despite the rotation...
component (Figure 4-22), the resultant displacement of the deck corner of the impact is in accordance with the displacement of the ship surge.

Figure 4-22 Variation of deck position (scaled) in the horizontal plane, top view, for case 3LJ

### 4.3.3 Jack-up model type

The jack-up rig model describes the traditional drilling unit of this type. Assuming that the model is developed for deeper waters, the three legs the structure is supported on are of the open-truss type, made of tubular steel sections that are crisscrossed. The legs have a height of 124 m and are 59 m apart from each other, and the deck-box, with a total mass of 16120 tons, is placed 75.2 m above the seabed level, 18 m above the assumed water level. For reasons of simplification, the racks are neglected in the analysis, but the local strength of the legs is compensated by increasing the wall thickness. Such members are mostly made of high-strength steel plates, of which similar mechanical properties are given in Table 1, although with higher yielding stress of the steel ($\sigma_y = 690$ MPa). The level of detail regarding the system of stabilization, either via mats or spud-cans, is assumed to be sufficiently represented for the current study by constraining the bottom nodes of the frame against all the translational and rotational degrees of freedom. Different from the jacket models, only the tubular members of the
impact zone of the hit leg are discretised using shell elements, with a total of 99219 shells. The remaining part of the struck leg and the two other legs are modelled by beam elements (4155 in total) in order to save not only processing time, but also modelling time. The number of joints in this structure model is significantly more than that for the jacked frames. Because the vertical distance between joints is smaller than the ship height, the ship impact is more like a leg impact involving more than one joint in the contact. Two different directions are considered for the impact in this study (Table 4-11 and Figure 4-23). The surge motion is set in order to be either in line with two legs or by crushing the leg at angles of ± 30 degrees relative to the direction in case 1 (Figure 4-23). Figure 4-24 shows the contact forces for the two cases.

![Figure 4-23 Collision impact cases for jack-up](image)

<table>
<thead>
<tr>
<th>Case</th>
<th>$v_0$ [ms$^{-1}$]</th>
<th>$E_{k0}$ [MJ]</th>
<th>$t_{leg}$ [mm]</th>
<th>$m_{top}$ [mg*10^6]</th>
<th>$m_{legs/mframe}$</th>
<th>$Z_{full}$ [m]</th>
<th>$Z_{Pbow}$ [m]</th>
<th>$Z_{Pbow}/Z_{top}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>JU0</td>
<td>5</td>
<td>37.5</td>
<td>100</td>
<td>16.120</td>
<td>2.86</td>
<td>95.4</td>
<td>75.2</td>
<td>123.8</td>
</tr>
<tr>
<td>JU30</td>
<td>5</td>
<td>37.5</td>
<td>100</td>
<td>16.120</td>
<td>2.86</td>
<td>95.4</td>
<td>75.2</td>
<td>123.8</td>
</tr>
</tbody>
</table>
The dynamic response of the jack-up platform has its unique characteristics as compared to the legged platforms. The kinetic energy to be converted into strain energy of the platform will be transferred initially to the struck leg and from there different portions will be transmitted to the ground and, via deck, to the other legs. Other aspects refer to the hull of the platform which is not necessarily positioned at the very top, and the legs are further from each other and only connected through the hull. Due also to the higher strength of the steel materials that these platforms are made of, and the most likely stiffer contact involving joints, the response is expected to be mainly elastic with no significant local deformations. It is also predictable that, for case JU30 the rotation of the deck needs to be considered as well, implying two DOF rather than a SDOF analysis to be more appropriate. For the rotation case, Equation 4-16 shall be re-written in the form:

\[ I_e \ddot{\theta} + R_e(\theta) = M_e(\theta) \]  

(4-33)

where \( \theta \) stands for the rotation angle around the centroid, \( I_e \) is the equivalent rotational inertia and \( M_e \) the equivalent moment, that is obtained from the product of the equivalent load by the horizontal distance between the load application point and the centre of gravity of the platform. For reasons of simplification the estimation of \( I_e \) is done using the equivalent masses as illustrated in Figure 4-25. The considered parameters for calculation of the structure response using a SDOF or a possible superposition between two DOF are listed in Table 4-12. For the situation in which
rotation has a significant influence on the installation global response, this is therefore obtained through superposition of the two SDOF evaluated separately, assuming the two SDOF are not coupled. The horizontal displacement of the contact point due to rotation is obtained from the relationship between the rotation angle from Equation 4-33 and the distance between the centre of rotation of the platform and the impact location. As already mentioned, the superposition is however only allowed since the response of these structures is linear, which is expected due to the higher local strength of the struck jack-up legs.

![Figure 4-25 Estimation of rotational inertia for jack-up](image)

**Table 4-12 Input parameters for impact JACK-UP**

<table>
<thead>
<tr>
<th>Case</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\Phi_L$</th>
<th>$\Phi_K$</th>
<th>$\Phi_M$</th>
<th>$K_x$ [MN/m]</th>
<th>$M_x$ [MN.m]</th>
<th>$I_e$ [kg.m$^2$]</th>
<th>$P_y$ [MN]</th>
<th>$M_y$ [MN.m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>JU0</td>
<td>-2.5</td>
<td>3.5</td>
<td>0.9</td>
<td>1.67</td>
<td>0.43</td>
<td>97.7</td>
<td>-</td>
<td>18.5</td>
<td>-</td>
<td>54.1+</td>
</tr>
<tr>
<td>JU30</td>
<td>-2.5</td>
<td>3.5</td>
<td>0.9</td>
<td>1.31*</td>
<td>0.43</td>
<td>97.7</td>
<td>154655</td>
<td>18.5</td>
<td>16.3</td>
<td>+</td>
</tr>
</tbody>
</table>

* $\Phi_K = 1$ for rotation DOF

+ elastic response for $P_y$ of 54 MN
The results shown in Figure 4-26 demonstrate the importance of accounting for both rotation and horizontal displacement in estimating the responses of the jack-up platform to ship impact (comparison between the oscillations for the two case studies). The differences for the curves in case JU30 are explained by the overestimated value of \( I_e \), leading to a different (shorter) period of the rotational degree. As shown, the real value lies about 90% of the estimated one, where the 2-DOF system demonstrates to be appropriate for assessing the elastic response of the jack-up platform to high energy ship impacts (Figure 4-27).

![Figure 4-26 Response of jack-up to ship impact](image1)

![Figure 4-27 Assessment of different DOF’s for case JU30](image2)
4.4 Discussion and recommendations

It has been shown that the inertia effects of the top mass of a cantilever structure can become determinant in evaluating the response to a transverse impact. This is especially prominent when the fundamental periods of the structure are long and the ratios of the top mass to the frame/column mass are large. In such situations it is sometimes necessary to include the contribution of the higher modes in deriving the equivalent SDOF system for reliable structural response analysis. It has been demonstrated that more accurate results could be obtained by performing two-stage response analysis. In the first stage up to about 80% of the loading duration $t_d$, deformation shape corresponding to a higher mode should be used to derive the equivalent SDOF system. After that in the second stage, the responses can be calculated by using the equivalent SDOF system derived with the fundamental mode shape as the deformation shape.

As for the more realistic steel platforms, since both the mass ratio (top mass/frame mass) and fundamental period are not as large as the considered example of single column supporting a concentrated mass, the inertial resistance effect from the concentrated mass at the deck level is less prominent, responses can thus be reliably predicted by using one step analysis with the equivalent SDOF system derived by assuming the fundamental mode shape as the deflection shape. A drawback of using equivalent SDOF analysis is that sometimes responses consist of two prominent modes, such as lateral and torsional response modes of the platform under eccentric impact. A SDOF system is not possible to capture the responses from the two distinctive modes. In such cases, two independent SDOF models, each representing a dominant response mode, need be developed for the analysis, and the total response is obtained by superimposing the results from two independent calculations. It should be noted that although the numerical results presented above indicate that this approach yields good predictions as compared to FE simulation, the approach is not valid when large nonlinear inelastic response exists because the total responses cannot be calculated by superposition of two independently obtained results. For the jack-up platform model considered in this study, the superposition of the results from two SDOF models yielded accurate predictions because of the primarily elastic response of structure owing to high strength/stiffness of the steel frames. For the jacket cases, despite the deck rotation, the analysis of a SDOF system without considering torsional responses appeared to be adequate as compared to the FE results, indicating relatively small torsional responses.
of the platform. The equivalent stiffness determined empirically showed not to differ much from those obtained directly from static pushover analysis for the 4-legged jacket (increase up to 10%). For the tripod case with no rotation of the deck, the equivalent stiffness of the SDOF estimated through the deformation of the struck leg can well match the value calculated from the period of the displacement curve of the top deck given by the FE calculations ($\Phi_K = 1.16$). For the case of same platform undergoing rotation, the deck displacement could be better predicted if using the rotational SDOF rather than the translational one. From the frame configuration, it is expected that the decks of fixed steel platforms with three legs are more vulnerable to rotation than those with 4 legs. The calculation of the structure overall response using the translational SDOF, with consideration the local deformation at the hit area, gives good response predictions of the majority of jacket frames as well as jack-ups when no significant rotation of the hull is expected. Based on the performed numerical simulations, the values listed in Table 4-13 can be used for estimation of ship impact through the evaluation of a SDOF system. It appears that the variation of the period for different platforms is the parameter that can be better related to the equivalent stiffness factor. The equivalent mass and load factors are also function of the deformation of the leg that must consider the vertical position of the ship impact and the total height of the platform. Therefore, parameters $\beta_1$ and $\beta_2$ must be provided in order to obtain reliable deflection shape $\Phi_L$ and $\Phi_M$, whereas $\Phi_K$ shall be empirically provided for including some rotational effects. For instance, in regards to the deformation of the struck leg of the jack-up, the deformed shape is also controlled by the position of the hull, which might not necessarily be at the top. As for the different ratios of $m_{deck}/m_{frame}$ considered in this study, no significant contribution to the platform response is found. In order to make the results listed in Table 4-13 more consistent, some more data from different cases is recommended. For any values lying between the given ones interpolation does not necessarily lead to accurate predictions.

<table>
<thead>
<tr>
<th>Type</th>
<th>$T$ [s]</th>
<th>$Z_{\text{deck}}/Z_{\text{bow}}$</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\Phi_M$</th>
<th>$\Phi_M$</th>
<th>$\Phi_L$</th>
<th>$\Phi_K$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jacket</td>
<td>1.5</td>
<td>0.7 - 0.8</td>
<td>-2.5 ; -3.0</td>
<td>3.5 ; 4</td>
<td>Eq. (4-6)</td>
<td>Eq. (4-8)</td>
<td>1.15</td>
<td></td>
</tr>
<tr>
<td>Jacket</td>
<td>3.5</td>
<td>0.7</td>
<td>-0.85</td>
<td>-1.85</td>
<td>Eq. (4-6)</td>
<td>Eq. (4-8)</td>
<td>1.05</td>
<td></td>
</tr>
<tr>
<td>Jack-up</td>
<td>4.5</td>
<td>0.8</td>
<td>-1</td>
<td>0.4</td>
<td>0.9</td>
<td>1.65</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 4-13 Estimation of platform response to ship impact through SDOF system
Equally important by means of simplification is the loading characterization of the ship surge impacts. This suggests that approximate and regular shapes for the impact load could be used for the evaluation of the simplified system response. From the selected velocities and ship displacement and according to Figure 4-3, any scenario in which the majority of the initial kinetic energy be converted into ship strain deformation is likely to be enough to provoke the maximum peak force. Since the platforms are not rigid, the resultant peak load of the loading curves is expected to be low. The load-time history curves in section 4.2.4 are different from those obtained for the real platform impacts. It is shown for the ship model used in this study that, for impact velocities of at least 5 m/s any contact load with a platform with not much localized damage can be approximated to a triangular shape. The triangles are defined by an ‘ascending’ edge which is steeper than the ‘descending’ edge along the time axis, meaning that the peak of the curve shall occur no later than the first third of the impact duration. The characteristics of the approximated load shape for each of the cases of section 4.3 considered in this study are listed in Table 4-14 and the results compared in Figure 4-28, from where it can be concluded that such approximations are adequate.

<table>
<thead>
<tr>
<th>Case</th>
<th>$t_p$ [s]</th>
<th>$t_d$ [s]</th>
<th>$P_{max}$ [MN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3LJ</td>
<td>0.4</td>
<td>2.02</td>
<td>24.0</td>
</tr>
<tr>
<td>3LM</td>
<td>0.36</td>
<td>2.00</td>
<td>20.9</td>
</tr>
<tr>
<td>4L1</td>
<td>0.44</td>
<td>1.58</td>
<td>22.5</td>
</tr>
<tr>
<td>4L2</td>
<td>0.46</td>
<td>1.54</td>
<td>23.8</td>
</tr>
<tr>
<td>4L3</td>
<td>0.48</td>
<td>1.60</td>
<td>21.3</td>
</tr>
<tr>
<td>4L4</td>
<td>0.5</td>
<td>1.68</td>
<td>22.3</td>
</tr>
<tr>
<td>4L5</td>
<td>0.36</td>
<td>2.04</td>
<td>22.5</td>
</tr>
<tr>
<td>JU0</td>
<td>0.48</td>
<td>1.30</td>
<td>21.2</td>
</tr>
<tr>
<td>JU30</td>
<td>0.46</td>
<td>1.34</td>
<td>20.3</td>
</tr>
</tbody>
</table>
Figure 4-28 SDOF response obtained with real and simplified impact load
Chapter 4

The variation of the time of the peak load, the load duration and the peak force are in general consistent with the variation of the initial impact velocity, so it is plausible to suggest the following values in Table 4-15:

Table 4-15 Estimated ship impact loads

<table>
<thead>
<tr>
<th>$v_0$ [ms$^{-1}$]</th>
<th>$t_p$ [s]</th>
<th>$t_d$ [s]</th>
<th>$P_{\text{max}}$ [MN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.0</td>
<td>0.45 – 0.50</td>
<td>1.30</td>
<td>20 – 21</td>
</tr>
<tr>
<td>6.0</td>
<td>0.45</td>
<td>1.60</td>
<td>22 – 23</td>
</tr>
<tr>
<td>7.0</td>
<td>0.35 – 0.40</td>
<td>2.00</td>
<td>24</td>
</tr>
</tbody>
</table>

The structure response can be still compared with equivalent static loads to the ship impact (Table 4-18). The dynamic amplification factors for different triangular loads for undamped elastic SDOF are plotted against $td/Tn$ and $tp/td$ in Figure 4-29 and listed in Table 4-17 considering the first mode of the different platform modes (the first modes for each platform are also listed in Table 4-16). The use of the dynamic factors from Figure 4-29 is somewhat conservative as the inelastic response of the SDOF system is neglected and the damping is not considered either.

![Figure 4-29 Dynamic Amplification factor for triangular pulse with zero damping](image-url)
Table 4-16 Natural periods/frequencies (first 6) of platforms

<table>
<thead>
<tr>
<th>Mode</th>
<th>Tripod</th>
<th>4-Leg</th>
<th>Jack-up</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_n$</td>
<td>$f_n$</td>
<td>$T_n$</td>
</tr>
<tr>
<td></td>
<td>[s]</td>
<td>[s$^{-1}$]</td>
<td>[s]</td>
</tr>
<tr>
<td>1</td>
<td>3.28</td>
<td>0.30</td>
<td>1.59</td>
</tr>
<tr>
<td>2</td>
<td>2.83</td>
<td>0.35</td>
<td>1.54</td>
</tr>
<tr>
<td>3</td>
<td>0.41</td>
<td>2.43</td>
<td>4.04</td>
</tr>
<tr>
<td>4</td>
<td>0.34</td>
<td>2.98</td>
<td>0.33</td>
</tr>
<tr>
<td>5</td>
<td>0.32</td>
<td>3.08</td>
<td>0.26</td>
</tr>
</tbody>
</table>

It is still possible to observe that for the jack-up case and based on the ratio between the loading duration and the fundamental period that the impulse can be used to get the equivalent static load.

Table 4-17 Equivalent static loading

<table>
<thead>
<tr>
<th>Case</th>
<th>$T_n$</th>
<th>$t_d$</th>
<th>$t_d/T_n$</th>
<th>$t_p/t_d$</th>
<th>DAF (approx.)</th>
<th>$P_{static}$ [MN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3LJ</td>
<td>3.28</td>
<td>2.02</td>
<td>0.62</td>
<td>0.20</td>
<td>1.4</td>
<td>504*</td>
</tr>
<tr>
<td>3LM</td>
<td>3.29</td>
<td>2.00</td>
<td>0.61</td>
<td>0.18</td>
<td>1.4</td>
<td>29.3</td>
</tr>
<tr>
<td>4L1</td>
<td>1.49</td>
<td>1.58</td>
<td>1.06</td>
<td>0.28</td>
<td>1.6</td>
<td>36.0</td>
</tr>
<tr>
<td>4L2</td>
<td>1.49</td>
<td>1.54</td>
<td>1.03</td>
<td>0.30</td>
<td>1.6</td>
<td>38.1</td>
</tr>
<tr>
<td>4L3</td>
<td>1.59</td>
<td>1.60</td>
<td>1.01</td>
<td>0.30</td>
<td>1.6</td>
<td>34.1</td>
</tr>
<tr>
<td>4L4</td>
<td>1.59</td>
<td>1.68</td>
<td>1.06</td>
<td>0.30</td>
<td>1.6</td>
<td>34.1</td>
</tr>
<tr>
<td>4L5</td>
<td>1.59</td>
<td>2.04</td>
<td>1.28</td>
<td>0.18</td>
<td>1.7</td>
<td>17.7</td>
</tr>
<tr>
<td>JU0</td>
<td>4.32</td>
<td>1.30</td>
<td>0.30</td>
<td>0.37</td>
<td>-</td>
<td>17.5</td>
</tr>
</tbody>
</table>

*Moment [MN.m]
Table 4-18 Comparison of platform deck response

<table>
<thead>
<tr>
<th>Case</th>
<th>$u_{\text{max}}$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FEM</td>
</tr>
<tr>
<td>3LJ</td>
<td>0.48</td>
</tr>
<tr>
<td>3LM</td>
<td>1.14</td>
</tr>
<tr>
<td>4L1</td>
<td>0.22</td>
</tr>
<tr>
<td>4L2</td>
<td>0.27</td>
</tr>
<tr>
<td>4L3</td>
<td>0.25</td>
</tr>
<tr>
<td>4L4</td>
<td>0.30</td>
</tr>
<tr>
<td>4L5</td>
<td>0.13</td>
</tr>
<tr>
<td>JU0</td>
<td>0.28</td>
</tr>
<tr>
<td>JU30</td>
<td>0.35</td>
</tr>
</tbody>
</table>

*From moment [MN.m] +2SDOF

Table 4-18 shows that the simplified systems using a reduced number of degrees of freedom have a good agreement with the FE models, whereas the error from equivalent static analysis can in some situations be significant (tripod case studies).

4.5 Summary

This chapter deals with the overall response of steel offshore platforms to large collision ship impacts. The dynamic aspects regarding the collisions are studied by means of detailed FE calculations. By filling some requirements in terms of local and global strength, steel offshore fixed platforms are shown to be able to resist high energy impacts from typical supply vessels at higher velocities than those normally considered in the current design practise.

Although it has been shown that the high concentration of mass on the top of these structures can have significant effects on the global response through the participation of higher modes, it is found that such influence is not pronounced for the real structure models having shorter periods than those of the single column models tested. The other factor that combined with the natural period of the structures can determine the
inclusion of the wave propagation effects is the impact velocity. Nevertheless, the impact speeds considered or in real ship impact scenarios seem to lie below such critical values. The response of such platforms to high energy impacts involving typical supply vessels is therefore of easier estimation and was also carried out through the integration of equivalent SDOF systems. The application of the methodologies described in the first part of this paper on equivalent systems appears to provide satisfactory accuracy in estimating the deck response of the platforms under real impact scenarios. Aspects such as local deformation or overall rotation can also be included to some extent in such estimations. There are however some parameters, such as the maximum resistance of the SDOF system, to which some particular attention must be drawn due to the complexity of the steel frames of real platforms. The suggested numerical approach represents an economic solution, but is recommended to be performed in conjunction with preliminary non-linear static calculations. It is also assumed that the results presented in this study can be made more consistent from the analyses of additional cases involving different and more complex platform configurations as, for instance, 8-legged jackets and/or other steel frames used in deeper waters.

In chapter 5 the collision problem is analysed for significant plastic deformations of the offshore platforms by means of energy dissipation and relative ship/facility strength.
Chapter 5

Energy Dissipation in High-Energy Ship-Offshore Platform Collisions

5.1 Introduction

The ship collision evaluation often includes impact events between the offshore installation and nearby vessels subject of off-loading strike. A risk assessment usually needs to be carried out for potential collision events that are screened according to their risk for the structure and also to their likelihood. For the assessment process, the failure shall be considered for members individually or by means of the overall performance of the facility, i.e. keeping the structure functional after, for instance, rupturing of a brace or denting of a leg. During the impact, the kinetic energy of the striking ship is converted into strain energy of the vessel and the facility (that can be either fixed or floating). Some of the energy might also remain associated with the motion of the structures after the impact (rebound). It is therefore important to account for the plastic deformation and failure of structural members that are affected by the collision since they will generally be associated with the primary structural effects. Fixed platforms are typically lower in redundancy than the floating ones and also constitute the most representative offshore structures [7, 122]. Likewise, the acceptance criteria defined for each type is also different. As for the energy amounts specified for the collision assessment, these are derived from both vessel size and impact speed. Although collision events involving supply vessels are currently predicted for ship sizes up to 5000 ton [3], these have significant variations in size according to the region they
As for the impact velocity, it might range from 0.5 m/s for low-energy collision to 2 m/s for drifting supply vessels. The combination between these two factors can actually result in large amounts of energy especially if the incidents involving passing vessels are considered, as well as a plausible increase in the number and average size of the world’s fleet in the coming years.

For the evaluation of the structural damage via energy balance, the internal energy consists of contributions from both the vessel and installation strain energy. Such contributions might vary upon the relative strength between the two structures. The methods used to estimate the strain energy in the current design practice can be very conservative because of neglecting the ship-platform interaction through the assumption of the ship to be rigid and the entire strain energy from the installation deformation, or less conservative, by analysing ship and platform being collided by a rigid body separately. For the latter case, the strain energy from the vessel, as well as the associated damage to it is usually underestimated same as the correspondent applied load. To improve the prediction accuracy, the high fidelity FEA provides a mean to perform the coupled analyses by simultaneously considering deformations of both the facility and ship, and including their interaction. This approach gains significance, in particular for cases where greater energy amounts than those currently predicted by the design practice, since a better accuracy could allow for a less conservative solution.

**5.2 Energy absorption**

Even though the elastic stiffness of the structures involved in the collision can affect the energy dissipation process, for high energy collisions the plastic deformations will absorb most of the initial kinetic energy, considering that the ship rebound will not be significant. Besides the global elastic vibrations of the installation, different plastic mechanisms can be formed locally on both ship and offshore facility depending on the collision scenario. The contribution of each of such modes is normally determined upon simplified hand calculation methods that can be found throughout the literature (Ref. [129], for instance).

**5.2.1 Local denting and beam bending**

For tubular members subjected to transverse loads, there are two mechanisms, i.e., beam bending and tube wall denting, which can interact with each other, as discussed in
chapter 3 and illustrated below in Figure 5-1. While global bending might govern the
deformation of braces, legs are usually designed against local denting.

![Bending deformation](image)

**Figure 5-1 Tube deformation under lateral transverse loading**

In a brief review of what has been discussed earlier, the plastic force-deformation
relationships for local denting of tubes are normally modelled by an equivalent spring
determined according to the mechanical and geometrical properties of the tube. The
energy absorbed due to local denting is then evaluated through integration of the force-
deformation curve of the spring. Commonly referred relationships for the prediction of
the tube lateral response formulated by Furnes and Amdahl [84] and Ellinas and Walker
[85] yield significantly different energy and impact force values depending on the
different $D/t$ ratios used, having the variations on the striker shape been considered in
[130]. Significant work has also been conducted in [131] on tube indenting and
crushing. The extension of the dent has been accounted in the integrated expression of
the denting deformation and energy absorption described in the DnV code [3] (Equation
3-25). For large deformations the global bending response is estimated using the three-
hinge mechanism with the load being modelled as a concentrated load, where the effects
of axial flexibility and the strength of the connections can be included by introducing
additional elements with the equivalent axial strength of the adjacent structural
members (Figures 3-3 and 3-4).

Despite the combined occurrence of local denting and beam bending, it has been
demonstrated that the contribution of the two different modes in terms of energy
absorption is not of easy estimation. Procedures based on idealized cross-section
deformation upon penetration of regular shape indenters [106] have been proved to be
inaccurate for denting values lower than 30% of the diameter of the intact cross section.
in chapter 3, especially when the contact areas with the striker are irregular. However, in the same section it is shown that the amount of strain energy dissipated by the tubes can be predicted based on the total displacement of the membrane in contact with the striker, regardless of whether bending or denting govern the tube deformation. It can be assumed that if the energy of the collision is written in its non-dimensional form considering only the strain energy of the tube, the same predictions can be performed regardless of the striker deformation. Equation 3-22 might therefore be taken for reference in the current chapter.

5.2.2 Axial crushing and buckling

The study of crushing mechanics of thin-walled structures has had its application in various fields such as design of energy absorption devices or car or ship crushing. The basic folding mechanism model constructed and presented in [58] constitutes the basis for the calculation of the crushing strength of boxes or square tubes and the quasi-static cross section methods used for determination of ship bow response. Modelling involving axial crushing of circular tubes can be found in [109] and [132]. The plastic energy can be evaluated from the axial deformation of the crushed tube in the form of a 'concertina', being equivalent to the work required to crush the element through a distance $2H$, where $H$ is the distance between two plastic hinges of a convolution (Figure 5-2).

Figure 5-2 Axial crushing mechanisms
In Ref [133] the collapse modes are studied for convolutions formed internally. The dissipated plastic strain energy can be calculated [133] as:

\[ E_b = 4\pi M_0 \left( \frac{\pi D}{2} + H \right) \]  

(5-1)

or from [134], as:

\[ E_b = 2\pi \sigma_0 \delta H^2 \left( 1 + \frac{2}{3} \frac{H}{D} \right) \]  

(5-2)

The above equations for the estimation of plastic energy absorption in the installation frame are applicable to braces adjacent to the struck legs or to the case with joint impacts in which the brace axial strength is lower than the leg transverse strength. However, for eccentric axial loads or long tube length, buckling of the platform tubes are more likely to exhibit modes similar to Euler buckling modes or even higher dynamic modes (Figure 5-3) that lead to less amounts of energy absorption than those obtained for progressive buckling in Equations 5-1 and 5-2. Studies regarding transition from buckling to bending can be found in [114-117] and [135]. The energy absorbed during the development of an axisymmetric fold in, for instance, a brace can be estimated by [117]:

\[ E_b = 2\pi \sigma_0 \delta \cdot \left\{ \left( \frac{\pi D}{2} + \bar{\ell} \right) \sqrt{1 + \bar{\ell}^2} \right\} \]  

(5-3)

where

\[ \bar{\ell} = \left( \frac{\pi D \delta}{2 \sqrt{3}} \right)^{1/2} \]  

(5-4)

Equation 5-3 is nevertheless only recommended for circular tubes with ratios \( D/t \leq 40-45 \) for being believed to underestimate the energy absorption for thicker tubes. This owes to the neglecting of the strain hardening effects and the absorption of some energy in axial compression at the development of each fold.

5-5
A parameter that also appears to influence the tube deformation under dynamic axial actions, besides its geometry, is the impact velocity of a striker [117]. From both experimental and numerical studies it is shown that the global bending mode develops quicker than progressive collapse for lower impact velocities, which can result in global bending of the tube, whereas higher impact velocities cause local folds (Figure 4-3) to develop more rapidly than a global bending mode. It is possible that both the global bending and buckling modes are combined during the tube deformation under compressive loading. The expressions currently known are thus not able to provide more than a general idea of the relative amounts of energy absorption by braces under compression in comparison with other deformation modes that the offshore facility is subjected to during the ship impact.

### 5.2.3 Overall frame deformation

Similar to the estimation of the denting energy, the overall elastic strain energy can be obtained if the structure deformation is simplified into a single-degree-of-freedom system with an equivalent stiffness. Depending mainly on the impact direction, the structure can either rotate or suffer a horizontal translation or even a combination of them. The relationships for global deformation would be easily defined as:
for translation, or:

\[ P = K_u u \quad (5-5) \]

\[ E_u = \int K_u u du = \frac{1}{2} K_u u^2 \quad (5-6) \]

for rotation. The spring stiffness \( K \) can simply be defined by measuring either the displacement \( u \) of the desired node or the rotation \( \theta \), upon applying a unitary single point load at the impact area. It is frequent to assume the global response of platforms to transverse loading to be simplified into a cantilever with an equivalent stiffness \( 3EI/L^3 \) at the free end for transverse loading (mentioned in chapter 4). As seen before, for dynamic impact however, in the early forced vibration period the platform response might also be governed by higher modes due to the inertia effects caused by the huge mass of the deck that is ‘concentrated’ at the free end of the so-called cantilever. The span curvature for such situation would then assume different and varying configurations with time, meaning higher equivalent flexural stiffness amounts.

### 5.3 Ship loading

#### 5.3.1 Ship types

According to [136] the number of world’s cargo carrying fleet of propelled sea-going merchant ships of no less than 100 gross ton was estimated (in January 2013) to be 86942, consisting of general cargo ships, tankers, bulk carriers, passenger ships, container ships and others. Among them, general cargo ships have the highest portion, and the oil tankers rank the second. Not only the sizes and layouts vary in these ships, but also steels of different strengths can be employed in shipbuilding. Therefore it is important to select more representative ship structures in the study as it is not possible to model all the ships. The International Association of Classification Societies (IACS) consisting of thirteen classification societies keep data of the majority of the world’s cargo carrying ships’ tonnage. From the number distributions of the principal general cargo ships, bulk cargo carriers, container ships, and oil tankers, analysed by the Systems Laboratory of Port and Harbour Research Institute (PHRI), approximate
relationships between the deadweight tonnage $DWT$ and general dimensions (length – $L_{pp}$) can also be established, in addition to the relationship between the vessel displacement $DT$ and deadweight tonnage $DWT$ mentioned in Chapter 3 [104]:

\[
\begin{align*}
\text{Cargo ships (<10000DWT)}: \log(L_{pp}) &= 0.867 + 0.310 \log(DWT) \\
\text{Cargo ships (≥10000DWT)}: \log(L_{pp}) &= 0.964 + 0.285 \log(DWT) \\
\text{Container ships}: \log(L_{pp}) &= 0.516 + 0.401 \log(DWT) \\
\text{Oil tankers}: \log(L_{pp}) &= 0.793 + 0.322 \log(DWT)
\end{align*}
\]

With regards to collision assessment of ships with offshore installations, the DnV design against accidental loads [3] prescribes deformation relationships for bows against jacket legs for supply vessels between 2000-5000 tons and energy impacts up to approximately 60 MJ, meaning, for instance, velocities of approximately 6 m/s for a vessel of ~3000 ton. Nonetheless, the trend observed in recent years is that heavier vessels have been being built [96]. The supply vessel can also differ according to where it operates, ranging, for instance, from 1000 tons in the Gulf of Mexico to 8000 tons in the North Sea [1, 128]. The nature of the collision events can also give an overview on how severe the facility damage can be. Different impact velocities are expected from regularly visiting supply vessels, passing vessels and off-loading shuttle tankers. Reports have shown [6] that the transfer of cargo followed by vessels that approach the installation and unloading operations seem to be the most common type of activities that lead to collisions with the offshore installation. Despite less frequent than transfer of cargo or approaching vessels, incidents involving passing vessels have also been reported [112]. These incidents involving passing vessels most likely involve high energies from the impact caused by higher velocities, and the highest energy collisions are likely to involve head-on impacts.

Some models representing ships of different sizes have therefore been considered in this study. The fleet comprises the following models:

1) Ship ‘S2’ – for supply vessels with displacements between 2000 and 5000 tons when fully loaded (already used for simulations in Chapters 3 and 4);
2) Ship ‘S10’ – for heavier supply vessels. Stronger scantlings than usual are assumed for this model;
3) Ship ‘S20’ – greater than model ‘S10’, displacement up to about 25000 ton;
The scantlings of these models are given in detail in the Appendix, whereas the particulars concerning the FEM are described in the next section.

### 5.3.2 FE models

Since the ship models have been chosen with the purpose of evaluating collisions that involve high amounts of energy, i.e., head on collisions, particular attention has been given to the ship bows. It is true that for lateral collision the hydrodynamic mass coefficients are greater than for bow impacts. However the velocities of head on collisions are higher, therefore leading to higher impact energy. The modelling of ship striking bows in literature diverges from rigid and approximate shapes [137] to detailed models with the deformable structure represented [138, 139]. Bows can be divided into conventional and bulbous. For the second case, the bow is expected to provoke higher stresses on the obstacle due to the increased stiffness resulting from the bulb contribution. In the current work the three ships are modelled with bulbous bows. Decks, frames and girders are included in all the models. The influence of longitudinal stiffeners is also checked by comparing the simulation results obtained with or without including the stiffeners in the model. Two other parameters which influence the bow deformation response are also considered. They are the strength of the scantlings and steel types. For the first, the scantlings are increased by increasing the thickness of the plates, while for the second different stress-strain curves are assumed. Some attention is also put on the impact velocity and the width of the obstacle, i.e., the contact area between the struck object and the striking bow.

The stress-strain curves of the ship steels used in the simulations meet the IACS criteria and are at this time considered from tensile tests [140]. The steels for scantling purposes have usually minimum yield stresses ranging from 235 MPa to 390 MPa (oil tankers and bulk carriers). The DnV rules, in turn, prescribe a tensile strength in the range of $\sigma_u = 400–490$ MPa. In this chapter yield stresses of 285 MPa and 365 MPa are considered for mild steel and high strength steel, and named as ‘M’ and ‘HT’ respectively. The curves of the piecewise linear plasticity material models adopted in the simulations and remaining mechanical properties are shown in Figure 5-4 and Table 5-1. Strain rate values are defined according to [50] and [108] for the different steel types and the
Cowper-Symonds (standard strain rate formulation) model is used [103]. The FE setup follows the same procedure as previously used for model ‘S2’ in chapters 4 and 5:

- The critical fracture strain values are set according to the element size of the adopted mesh size and the nominal values normally taken for steel materials.
- Shell elements of size between 80 mm and 100 mm are used for the steel plates of first sections of the ship. The element size is determined after performing mesh convergence tests [101, 102].
- For the rear part of the ships, where no significant deformations are expected to occur, rigid solid elements are employed and rigidly connected to the shells of the front portion. The total mass of the ships is obtained by adjusting the density of the rigid blocks, and the ships real shape is kept in order to capture the inertia effects in the model. The total mass of the models also includes the hydrodynamic added mass of 5-10% assumed for surge.

![Figure 5-4 Stress-strain curves of ship steels](image)

**Table 5-1 Steel properties of the ship models**

<table>
<thead>
<tr>
<th>Steel type</th>
<th>E [GPa]</th>
<th>(\sigma_y) [MPa]</th>
<th>(\sigma_u) [MPa]</th>
<th>(\epsilon_u)</th>
<th>(\nu)</th>
<th>(\rho) [Kg/m³]</th>
<th>C [s⁻¹]</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>‘M’</td>
<td>200</td>
<td>285</td>
<td>550</td>
<td>0.17-0.37*</td>
<td>0.3</td>
<td>7800</td>
<td>500</td>
<td>4</td>
</tr>
<tr>
<td>‘HT’</td>
<td>200</td>
<td>365</td>
<td>620</td>
<td>0.17-0.37*</td>
<td>0.3</td>
<td>7800</td>
<td>3200</td>
<td>5</td>
</tr>
</tbody>
</table>

*dependent on the shell thickness

### 5.3.3 Bow deformation

The estimation of the deformation and loading exerted by striking bows has been addressed, since Minorsky [24], either via mathematical models [50, 81, 141], or model
Chapter 5

tests [142-144]. The mathematical models have been developed from the basic folding mechanisms [83] described in Section 5-2 (Figure 5-2) and continuously improved. In general, for such quasi-static analysis it is required that the ship layout/structure susceptible of deformation is well known as the deformations are evaluated based upon the number and nature of the plate intersections in each cross section along the ship length. This might turn such procedures too complex as part of the full collision analysis of vessels against offshore facilities.

Because a very high level of detail could be, to some extent, very time consuming, not only in terms of the model complexity, but also by means of FE calculation time, the concept of equivalent plate thickness [145] could be plausible in order to account for the effects of relatively small stiffeners. In this case, the stiffened bow is replaced by an unstiffened structure in which the outer shell is calculated, following the smearing out technique as:

\[ t_{eq} = t + k \cdot \frac{A_s}{\delta} \]  

(5-10)

where \( t_{eq} \) is the equivalent thickness, \( t \) is the thickness of the outer shell, \( k \) is an empirical constant usually taken as 1.0, \( A_s \) stands for the sectional area of the stiffeners and \( \delta \) represents the spacing between the stiffeners. Experiments by Paik and Pedersen [146] have shown that longitudinally stiffened structures could be reasonably replaced by equivalent unstiffened structures by using the smearing out technique (Figure 5-5). This method was applied by Yamada and Pedersen [147] in combination with Yang and Caldwell’s method [50] to different kinds of bulbous sections. In fact, the mean crushing forces calculated for the equivalent unstiffened structure are lower, although they could be more accurate when correlated with experimental data. The neglecting of lateral buckling of the stiffeners actually results in a lower value of the plastic bending moment \( M_0 \). The adopted technique is considered for the assumed striking bows with moderate deformations since for a struck ship, the fracture initiation could take place at a different location than the ship front.
Simple empirical formulas have been derived by other authors, making use of the databases of previous collision cases and other statistical data. Parameters such as the ship total mass, ship size, impact velocity, strain rate effects are accounted in different expressions. Saul and Svensson [148] give the maximum impact force based on the deadweight of the vessel with a scatter of 50% based on the bow shape and structure type:

$$P_{bow} = 0.88\sqrt{DWT} \pm 50\% \tag{5-11}$$

The US-Guide Specifications [149] estimate the maximum crushing force with the inclusion of the ship initial velocity $v_0$ by:

$$P_{bow} = 0.12v_0\sqrt{DWT} \tag{5-12}$$

while Pedersen’s expression (Equation 3-4) was empirically derived, for ice-strengthened bows, based on a series of analysed collision cases using Amdahl’s [81] and Caldwell and Yang’s [50] procedures, for the calculation of the maximum bow collision load.
It is however common to all these expressions that the width of the struck obstacle is neglected, and thus the bow response for impacts against obstacles of limited width would likely overestimate the deformed steel volume and underestimate the penetration depth of the obstacle. The eccentricity for impacts against obstacles of limited width is another parameter that must be taken into account, not only due to the different strength of the contact areas between ship and obstacle, but also because of the hydrodynamic effects.

To account for all the above issues, in this study numerical simulations are carried out first by subjecting the three bow models against rigid obstacles.

Despite the impact velocity is considered in some of the above formulas, it is shown in Figure 5-6, from the impact forces calculated according to Equation 3-4, that its influence is not noticeable on the bow deformation caused by a fixed rigid object, in which the force-deformation curves for different velocities are obtained for Ship ‘S10’ using steel ‘M’ and larger scantlings against a rigid cylinder of \( D = 2 \text{m} \). For initial velocities of 3, 5 and 7 m/s the contact force can be directly related to the crushing distance \( s \) since it is mainly a function of the amount of kinetic energy. This can be assumed since the duration of the loading, which is not shown here, is much greater when compared to the period of the ship motion.

![Figure 5-6 Bow deformation for different impact velocities (ship ‘S10’)](image)

The comparison between the deformation of the same ship model using the stiffened bow and unstiffened bow with smeared out thickness is also made for an impact speed of 7 m/s (Figure 5-7), where the reduction of the bow force with the crushing distance can be about 10% – 20% for penetration depths greater than 2 m. The same bow is
considered with the same impact speed against a rigid wall, which shows an increase of
the peak force from 81.7 MN to 96.9 MN (19% increase) and a decrease of the bow
crushing distance from 6.4 m to 4.7 m (27% decrease) relatively to the impact with the
rigid cylinder (bow deformation illustrated in Figure 5-8). This proves the importance of
accounting for the width of the struck obstacles, especially when evaluation of the
energy dissipation is based upon the stiffness of the contact areas.

Figure 5-7 Stiffened bow section vs. unstiffened bow section with smeared out
thickness (ship ‘S10’ vs. rigid wall)

Figure 5-8 Bow deformation, stiffened model ‘S10’ (left – rigid wall; right – rigid
cylinder)

The influences of other parameters such as different scantlings and steel materials are
examined for the three bow models against a rigid cylinder of 2.0 m diameter (Figure 5-
9) and are compared with the predictions from other authors in Figure 5-10. The
classification of the ships, according to their deadweight tonnage, is made according to statistical data from [104], as well as considering Equation 5-11 in order to estimate the bow forces according to [48]. The penetration depth $s$ is obtained/measured from the forecastle deck end that can be more or less forward in comparison with the bulb, depending on the different bow configurations.

Figure 5-9 Bow deformation (FEA)
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The difference between the various predictions confirms the difficulty in generalizing the deformation response of ship bows, especially if accounting for special ship categories or by looking at both upper and lower bounds of Equation 5-11. However, regardless of the variations of the rigid obstacle width or the bow layout (stiffened or unstiffened with smeared out thickness), the FEA seem to show good agreement with the curve from the US Guide (Equation 5-12), and is in between the upper and lower bounds from Saul & Svensson's curve (Equation 5-11). Equation 3-4 clearly indicates greater bow forces and the respective spread of them.

5.3.4 Simplified equivalent system

Knowing that explicit finite-element techniques may not be ideal for practical engineering problems, as the explicit integration of the models normally requires supercomputing resources and long calculation time, simplified systems are developed for efficient bow crushing response analysis, based on the FEA. By assuming that in a
frontal collision the energy absorbed is mainly function of the deformation in the
direction of the initial ship motion (surge), the deformed bow can be replaced by a
spring (SDOF) or a set of springs in parallel. In [150], where the bow deformation is
studied for ship-ship collisions, the calculations of the collision damage consider the
vertical variation in the stiffness of the striking bow. The bow stiffness is then
represented as a set of non-linear springs modelled according to [151] with associated
slacks representing the geometrical bow form. From observing the energy curves in
Figure 5-6, it can be drawn that their growth follows a quadratic trend (at least) until the
peak force is reached. After that point, the increase is nearly linear with the null growth
of the contact force. The energy curves for the ascending trend of the force can therefore
be approximated as:

\[ E_{bow} = A_I \cdot s^2 + B \cdot s \]  \hspace{1cm} (5-13)

and the force obtained from:

\[ P_{bow} = \frac{\partial E_{bow}}{\partial s} = \kappa \cdot s + B \]  \hspace{1cm} (5-14)

where \( A_I \) and \( B \) are constants that characterize the force and energy curves and \( \kappa = 2A_I \). The spring law is then described by a rigid behaviour until the force reaches a
magnitude of \( B \). From this point until reaching the peak force, the spring deforms
linearly with a stiffness value equal to \( \kappa \). Such parameters might also provide a practical
tool in evaluating the energy dissipation based on the relative stiffness approach when
the struck obstacles are deformable too.

When it comes to collisions in which the bow is only partially involved, a simplified
system using two or more springs in parallel may reproduce the ship response more
accurately. This is the case, for instance, when the obstacle is an inclined platform leg or
even a brace. The curves in Figure 5-6 are plotted assuming two parts of typical bulbous
bows of which loading on a struck obstacle could be split into two main and distinct
highly concentrated stress areas from bulb and stem (Figure 5-11).
The bows under analysis have higher strength when compared to conventional bows due to the inclusion of the bulb in the structure, which is also the stiffer part of the bow. Based on the FEA, the trends shown in Figure 5-12 are assumed for estimation of $\kappa$ and $B$ from the ship dimensions, including factors such as the strength of the scantlings and the steel grades. Since the evaluation is carried out considering central deformations, for eccentric impacts both loading and $\kappa$ and $B$ would be lower if the contact parts did not involve the central girders. For eccentric impacts the ship would likely rotate over the obstacle, therefore changing the direction of the penetration and the hydrodynamic mass factors would equally have to be readjusted due to the ship sway. Another consideration to be done with regards to the stiffness of the springs used in the equivalent systems is that after the maximum contact load the bow force seems either to reach a plateau or decrease. Thus, in case of a very stiff obstacle penetrating the ship bow much further than the limits considered in the analysis (higher energy impacts) the spring stiffness shall rather be considered as null after the penetration depth corresponding to the maximum force has been reached. Values for $\kappa$ and $B$ are also given if only the bulb is accounted/involved during the collision ($\kappa_b$ and $b_b$). The same can be estimated for the remaining part separately from the idealization of a system of parallel springs ($\kappa_a$ and $b_a$).
The curve fit for $\kappa$ in Figure 5-12 is done for the three different bow models considering the respective variations in terms of plate thickness or steel strength. For model 'S20' the shell thickness of the unstiffened bow is smeared out, making $\kappa$ to lie below the projection. It has already been shown for model 'S10' that for equal crushing depths the difference between the peak force using stiffened and unstiffened bows with increased shell thickness varies between 10-20%. The respective equivalent spring stiffness is therefore affected. The use of larger scantlings that can be more representative of special ship classes could also indicate the projections to be conservative for the majority of the cases as these ICE class ships or similar are less common. As for parameter $B$, it appears that the reference value of the force for which bow deformations start to become significant should be taken as constant regardless of the ship size (if within the range of the vessel sizes considered). Yet, the extrapolation of these values for ships of possible higher impact energies is not recommended and does not seem to be of significant interest as the respective strengths and energy amounts for collisions.
involving such vessels would likely go much beyond the order of magnitude that offshore steel fixed platforms could bear.

5.4 Collision simulations

5.4.1 Platform models

For the purpose of examining the energy dissipations on different strength obstacles, three platform models are considered: a tripod, ‘P3’, and a four-legged jacket, ‘P4’, with the total heights from the seabed to the deck of 71.0 m and 53.6 m, respectively; and an eight-legged jacket model, ‘P8’, for deeper waters with a height of 201.2 m (Figure 5-13). The steel frames of the three platforms are constituted by tubular members, with the material properties described in Table 5-2. The API RP-2A [1] requires that structural steel pipes must conform to the American Society for Testing and Materials (ASTM) A36, with yield strength of around 250 MPa. However, the medium grade structural steels of which offshore platforms are conventionally constructed have yield strengths in the range of 350 MPa with some platforms installed in the North Sea having been constructed with 400-450 MPa steel, according to [152]. The joints are modelled as welded in this study. For different collision cases the same platform model is given different thickness values so that not only global but mainly local strength and different ship/platform stiffness ratios could be considered in the evaluation of energy dissipation. Impacted legs have lengths ranging between ~15 and ~25 m and diameters of 1.25-1.70 m, whereas the struck braces range in length from ~14.0 m to 20.0 m and have diameters of 0.5m and 0.8m. The steel frames of models 'P3' and 'P4' are discretised with shell elements, while for model 'P8' only the two top storeys (see Appendix) are modelled using shells. Due to the dimensions and complexity of the model ‘P8’ and in order to save computation resources, the rest of the frame where no big plastic deformations are expected from the impact, beam elements are used. The fracture strain for the defined steel materials is adjusted according to the mesh size of the shell elements (100 mm) and thickness. The mesh size is determined after a mesh convergence test. The top decks of the platforms are rigidly connected to the frames and modelled using rigid materials. The vertical action of gravity is added to the deck and the design weight values adopted are checked according to the axial preload design values [1]. The mass values are 8000, 10000 and 12500 ton for models 'P3', 'P4' and 'P8' respectively. The structural particulars of the platform models can be seen more in detail in the Appendix.
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Figure 5-13 Offshore platform models

Table 5-2 Platforms steel material

<table>
<thead>
<tr>
<th>Part description</th>
<th>E [GPa]</th>
<th>$\sigma_y$ [MPa]</th>
<th>$\sigma_u$ [MPa]</th>
<th>$\varepsilon_u$</th>
<th>$\nu$</th>
<th>$\rho$ [Kg/m$^3$]</th>
<th>C [s$^{-1}$]</th>
<th>P [10$^3$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube</td>
<td>200</td>
<td>345</td>
<td>450</td>
<td>0.20</td>
<td>0.3</td>
<td>7800</td>
<td>40.4 [103]</td>
<td>5 [103]</td>
</tr>
</tbody>
</table>

5.4.2 Collision cases

The numerical models for ships and platforms described above are used to perform ship collision simulations. The case studies under analysis refer to different possibilities of ship-platform collision and different deformation scenarios. As mentioned earlier, not only different ship and platform models are used, but also the strength of some of the tubular steel members is adjusted by changing the thickness. It is expected that by
adjusting the relative thickness of ship and installation the energy may be dissipated through the different structures in different proportions and also through the different plastic mechanisms formed. The collision scenarios are illustrated in Figure 5-14 and the initial data reporting to each collision case are shown in Table 5-3. The scenarios considered include collisions on leg, joint and brace as well as different impact angles. The position of the arrows in Figure 5-14 indicates the approximate position of the highest concentrated stresses caused by the contact. This should normally correspond to the bulb position. The only exception is case 19, in which the upper part of the bow strikes the brace. In case 20 the contact also involves the struck brace of case 19 although the ship position is lowered. For reasons of simplification and analysis, the same thickness values indicated in Table 5-3 are used for all the braces or legs.
### Table 5-3 Description of collision cases

<table>
<thead>
<tr>
<th>Case</th>
<th>Model</th>
<th>Scantlings</th>
<th>Steel</th>
<th>$v_0$ [m/s]</th>
<th>$E_{ik0}$ [MJ]</th>
<th>Installation Model</th>
<th>$t_{leg}$ [mm]</th>
<th>$t_{brace}$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>‘S2’</td>
<td>Normal</td>
<td>‘M’</td>
<td>7.0</td>
<td>74.3</td>
<td>‘P3’</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>2</td>
<td>‘S2’</td>
<td>Normal</td>
<td>‘M’</td>
<td>7.0</td>
<td>74.3</td>
<td>‘P3’</td>
<td>45</td>
<td>20</td>
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<tr>
<td>3</td>
<td>‘S2’</td>
<td>Normal</td>
<td>‘M’</td>
<td>7.0</td>
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<td>‘P3’</td>
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<td>Normal</td>
<td>‘M’</td>
<td>7.0</td>
<td>74.3</td>
<td>‘P3’</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>5</td>
<td>‘S2’</td>
<td>Normal</td>
<td>‘M’</td>
<td>6.0</td>
<td>54.6</td>
<td>‘P4’</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>6</td>
<td>‘S2’</td>
<td>Normal</td>
<td>‘M’</td>
<td>6.0</td>
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<td>‘P4’</td>
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<td>‘S2’</td>
<td>Normal</td>
<td>‘M’</td>
<td>6.0</td>
<td>54.6</td>
<td>‘P4’</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>8</td>
<td>‘S10’</td>
<td>Large</td>
<td>‘M’</td>
<td>3.0</td>
<td>59.0</td>
<td>‘P4’</td>
<td>60</td>
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</tr>
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### 5.4.3 Model response

Unless the offshore steel frames have very stiff elements as well as stiff connections and stronger bracings, the energies involved in any of the scenarios should be enough to provoke the individual failure of some members. The need of considering higher redundant platforms such as model ‘P8’ owes to a better assessment of the contribution from different plastic mechanisms in the whole system (vessel + installation) response. For instance, in cases 2 and 5, the deck of platform ‘P3’ falls (global collapse of the deck) after the failure of the hit leg, regardless of the impact type – beam or joint. The amounts of internal energy of the installation are in both cases similar (see Table 5-4), but the failure mechanisms are highly influenced by the gravity action on the deck after the global collapse has occurred. In fact, the platform has absorbed respectively 11.2 MJ...
and 12.3 MJ energy when the leg failure occurs. In contrast, for platform ‘P8’, larger
dents on the leg wall and eventual leg fracture are not sufficient to lead to an overall
structural collapse. The energy dissipation in each of the numerical cases is shown in
Table 5-4 and in Table 5-5 the platform response is described more in detail,
considering the different deformation mechanisms.

Table 5-4 Energy balance

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<th>Case</th>
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<td>$E_{inst}$</td>
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<tr>
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<td>[MJ]</td>
<td>[MJ]</td>
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*deck fall
*brace failure

5-24
Table 5-5 Installation response

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<th>Case</th>
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<th>E</th>
<th>[%]</th>
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<th>Adjacent members$^x$</th>
<th>Total</th>
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<td>[MJ]</td>
<td></td>
<td>u [m] E [%]</td>
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*Structure global collapse – only total internal energy of installation provided

$^x$brace failure

$^x$Braces with significant local contribution to the total internal energy of the installation

Attention must be paid to any connection that can be made between the displacement of the membrane of the hit zone of the tube platform and the energy dissipated. For situations in which the full leg is able to bend without any local denting, the values $u$ (displacement of the struck membrane of the platform due to beam bending, local denting or both) and $\Delta$ (displacement of the rigid deck) can be directly correlated with the configuration of the structure shape due to the impact action and respective wave propagation through the frame. This is actually regarded in cases 4 and 13, where the strain energy is assumed as part of the overall strain energy. In general, the values of the
strain energy can also be well estimated using Equation 3.25 if \( u \) as given in Table 5.5 is relatively small as compared to the tube diameter.

The contribution of adjacent braces near the contact point is visible especially when the platforms are single or K-braced (‘P3’ and ‘P8’). The strain energy absorbed by each of the braces in which the plastic deformations are significant is shown in Table 5-6 (\( u_b \) stands for the maximum deformation measured perpendicular to the tube axis due to axial compression) and Figure 5-15. The relationship between the deformation configuration and the amount of strain energy of the braces due to axial compression is not of easy estimation since the development of folding can be combined with global bending. The maximum energy amounts on each of the braces subjected to axial actions are generally close and seem to vary more with the change of wall thickness rather than the diameter. As a reference, according to Equation 5-3 a variation of tube wall thickness from 15 to 20 mm for the range of diameters considered (0.5 to 0.8 m) would result in a factor of approximately 1.6 times more energy to develop an axisymmetric folding. In cases 17 and 18 it is still possible to figure out that the rupture of two braces occurs at the joint also due to the influence of shear. Nonetheless, the tubes under axial compression always end up failing before energy values reaching around 3 MJ for \( t = 15 \) mm and below 4 MJ for \( t = 20 \) mm. The only exception is found for case 18, where the failure occurs at an energy value of 4.7 MJ, possibly due to a stiffer brace with short length, which makes the transition from progressive buckling to global bending more difficult and therefore increases the capability of absorbing energy of the tube.

<table>
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<tr>
<th>Case</th>
<th>( L ) [m]</th>
<th>( D ) [m]</th>
<th>( t ) [mm]</th>
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<th>( E ) [MJ]</th>
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<td>0.8</td>
<td>20</td>
<td>0.9</td>
<td>3.6</td>
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The configuration of the braces is equally important to define the overall strength of the installation. From the cases involving platform ‘P4’, there is no yielding in any other members than the hit leg. Despite in some of the cases that some small local denting is noticed for the $t = 45$ mm legs, the majority of the kinetic energy is converted into strain energy via ship deformation, whereas among the small portion of strain energy converted to the platform, the majority is taken by the frame deformation.

The frame deformation (global elastic deflection of the entire jacket) capacity associated with large global displacement of the frame is very important because it can absorb large amount of impact energy, in particular for bigger installations such as model ‘P8’. The frame global energy is not only dependent on the stiffness of the structure in the impact direction, but also proportional to the squared displacement $\Delta$ (by following the SDOF analogy). The comparison between cases 15 and 18 can be illustrative of it if both values of $\Delta$ and $E$ are accounted. Any of these ‘global’ energy values measured for model ‘P8’ from cases 14 to 18 presented in Table 5-5 are above those currently considered by any design code. However, in any of the collision scenarios both braces and legs can be subjected to local deformations, so that the local strength must always be considered. Still in regards to the measured $\Delta$ displacements, the model ‘P3’ undergoes considerable rotation over one of its legs (case 3) so that in such situations
the rotational stiffness plays an important role in the way how the energy is transferred through the facility frame.

The interaction between ship and installation for each of the cases can be seen in Figure 5-16. The displacements $u$ and $s$ are measured from the main contact point. Although these graphics can give a general idea of the relative strength of each of the structures and the energy taken by each of them, the energy absorbed by adjacent braces is not directly reflected on the global deformation curves (cases 3 and 16 to 18, for instance, when compared to the amounts given by Tables 5-4 and 5-5). Comparatively to the values shown in Table 5-5, the maximum $u$ plotted in Figure 5-16 is sometimes higher as they represent the peak values and not the final/residual displacements of Table 5-5 that account for the system unloading.

Another note concerns the way how the bow crushing distance must be pointed as in some cases, as mentioned earlier, the eccentricity of the collisions has to be considered and may influence the deformation configuration of the bow. Since it has been tried for all the situations that the ship bulb is aligned with the axis of the struck member and the global crushing distance $s$ is measured from where the first deformations occur, these not always coincide with the centre of the ship at the forecastle deck (see Figure 5-17). The approximated stiffness values estimated previously might therefore overestimate the ship action on the platform if simplified hand calculations are to be used.
Figure 5-16a Collision strain energy balance
Figure 5-16b Collision strain energy balance
Figure 5-16c Collision strain energy balance
The predictions in terms of the total platform internal energy from the deck displacement/rotation (Table 5-7) and based on Equations 5-5 – 5-8 are compared against the FEM results for six of the 18 case studies in Table 5-8. Local damage is also verified against hand calculations (Sections 5.2.1 – 5.2.2)

Table 5-7 Rigid deck measured displacement/rotation

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</thead>
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<td>1.04</td>
<td>0.13</td>
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<td>1.28</td>
<td>0.27</td>
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Chapter 5

Table 5-8 Platform internal energy prediction

<table>
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<th>Case</th>
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<th>Predicted E [MJ]</th>
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<td>8.1</td>
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<td>-</td>
<td>4.2</td>
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</table>

*reference value taken from Equation 5-3 (for axial crushing)

As it has been demonstrated in Chapter 4, the estimation of the elastic strain energy during the collision time using an equivalent SDOF system based on the translation of a cantilever tip mass can be complex, as during the forced vibration period the effect of higher modes influences the overall response. In other words, the equivalent stiffness associated to the deformation decreases in value diminishing the increasing rate of elastic strain energy. Nonetheless, at later stages the values of strain energy are expected to match one another, as for the later contact and free vibration stage of the structure the first mode of the cantilever should give a good estimation of the platform internal energy. Collision scenario 1, where no rotation of the platform occurs, can be used to illustrate it. On the other hand, for the other five cases, the prediction of the absorbed energy by the platform is better made if the rotational stiffness is considered rather than translational, as Figure 5-18 shows.

Cases 6 and 9 differ from one another in terms of where the contact occurs i.e. joint and mid-span, upper and lower point, respectively. The maximum energy estimated by the SDOF system based on the rotation of the ‘rigid’ deck is in good agreement with the numerical results, where greater rotation is generated by the mid-span impact case (softer point). The residual rotation of the deck for both cases appears to be similar, whereas some very small residual horizontal displacement in the impact direction of a shell node of the hit leg right below the deck is measurable for case 9 and nearly null for case 6 (Figure 5-19).

As for case 7 the comparison can be made with case 6 where the difference for the joint impact relies on the different thickness values adopted for the tube walls. It is expected that for a leg with the given values of 1.7 m diameter and 45mm thickness at least the
zone of higher concentrated stresses, corresponding to the contact with the bulb, would
generate some local denting of the tube wall and the consequent additional strain
energy. Despite the very small maximum denting of 0.13 m measured in the leg at the
joint level, the global response of the structure is still similar to that in which the leg
cross-sections at the contact zone remain intact, although the maximum angle of twist
that is measured for both cases is slightly greater for case 7 (Figure 5-19). In terms of
the maximum strain energy, the sum of those from denting and overall rotation is in
agreement with that measured from the FE analysis.

In regards to case 8, the dent area has a larger extension than in case 7 and a maximum
depth penetration of approximately 0.3 m. It can be observed that, comparative to case
7, the frame elastic energy amounts of the platform are close (4.7 and 5.4 MJ),
regardless of the localized damage.

It is a common fact to all these six case studies that even for situations where the
maximum energy can be estimated from the sum of localized and global mechanisms or
just the second, either the residual rotation angle of the deck or its residual horizontal
displacement underestimate the amounts of strain energy that remain in the platform
after the impact (Figure 5-18). It is believed that some stresses tend to concentrate in
some of the elements of the joints in which plastic strain can occur. Although the
accumulation of energy in the joints takes place, this is not always reflected or visible
from the global displacement of the platform. It shall also be mentioned that part of
these zones undergoing plastic strain mainly grow/develop after the first peak of the
global rotation or displacement have been reached. The total energy stored by the
installation is not therefore much inferior to that predicted by hand calculations based
on maximum rotation/horizontal displacement.
Figure 5-18 Strain energy of installation – comparison between numerical FE and theoretical predictions
5.5 Discussion

The set of numerical simulations described in the previous section provides meaningful information regarding the response of both ship and platform systems (and considering their interaction) within a range of diverse scenarios. The degree of complexity of these two structures demands the use of simplified equivalent systems for quick hand calculations. In both structures the formation of plastic mechanisms can involve various members. The development of numerical models for the estimation of the ship bow response is well documented and the bows can be assimilated to a single or a small number of multiple degree of freedom systems. According to the calculations performed in section 5.2, such models can be characterized mainly by springs initially rigid and then deformable according to an approximate constant plastic stiffness after the loading has reached a minimum value $B$. For the installation, although the deformation
mechanisms of the members of the platform are normally assessed through simple hand calculations as well, adjacent members that are not directly involved in the collision can also experience significant deformations. This in fact makes the evaluation of the overall response more complex when such deformation mechanisms in different parts are combined. The separation between the different absorption mechanisms can become a complex process when taking into account the coupled response of both local and global effects. Whereas for beam impacts the mechanisms observed involve the flexural response of the struck member and the frame elastic response, provided that the connections are strong enough (for which the single bracings of case 3 is the only exception among the studied cases), when the ship action is directly exerted on a joint the adjacent braces are more likely to contribute to the formation of different mechanisms to the energy absorption process. Deformations caused by shear or axial crushing can be found for braces that are perpendicular or more aligned with the direction of the impact loading. Equations 5-1 to 5-4, however, cannot predict approximate energy amounts based on the deformations in braces, where these always differ from the illustrations of Figures 5-2 and 5-3. This can make the estimation of the total response of the platform less accurate for joint impacts when the connections have a lower strength. For cases 14 and 16 to 18, of which deformed braces are shown in Figure 5-15, different areas of concentrated stresses are noticeable for both ends and middle span. From the contribution/weight that the buckling of braces can have on the total response as shown in Table 5-5, the number of braces with plastic strain is shown to not necessarily reflect the total amount of energy taken by these members. Their contribution appears to reach up to approximately 25% of the total energy taken by the platform (collision 18), even though the separation between elastic and plastic strain energy is not made for the deformed tubes.

For the remaining cases where the flexural response of the struck leg or brace is responsible for the plastic strain energy of the installation, the analysis is easier to be carried out and the local mechanisms analyzed in previous studies [84, 85 and 95] can be employed effectively. As for the global energy, the frame elastic energy values given Table 5-5 follow the measured displacement $\mathcal{D}$ suggesting, for instance, the use of simple SDOF analogies, provided that the global stiffness of the facility is not significantly affected by the local damage of the collision point (chapter 4).
Chapter 5

With respect to the energy dissipation and respective design principles, a study by Storheim and Amdahl [153] takes into account the compactness factor $R_c$ (Equations 3-25 and 3-29) of the struck beam to predict the energy dissipation. The majority of the energy will be dissipated by the ship if $R_c$ is larger than the maximum collision force, otherwise the platform member will deform according to a three-hinge-mechanism as described in [95]. In [153] values of $R_c > 1.7$ MN are assumed to be sufficient to crush the bulbous bows without significant denting ($R_c > 1.3$ MN for broadside impacts), according to the tube characteristics, the leg of 45 to 60 mm thickness, depending on the diameter. These values are obtained assuming a concentrated load, i.e. parameter $b$ which is used in [3] for the tube resistance against local denting and in Equations 3-25 to 3-29 is equal to 0. If distributed loads from the contact area are considered then $R_c$ could in theory assume lower values than those presented for $b = 0$.

For a consistent ship/installation stiffness ratio that can be practical in estimating the energy dissipation from collisions involving significant amounts of kinetic energy, it has been shown that different mechanisms and their interaction need to be addressed. For the ship structure, statistical data that account for parameters such as the plate thickness, ship dimensions or number of plate intersections are taken in order to derive SDOF or MDOF consisting of parallel springs, from which the spring stiffness can be derived. For the platform, however, the frame deformation is shown to dissipate high percentages of the total energy besides the local mechanisms occurred in or nearby the collision point. Expressions from [84] and [85] have a degree of $\frac{1}{2}$, contrary to the quadratic equations derived for the deformation of the bow models. As for Equation 3-25, the complexity is higher since it depends on the relationship between the length of the dent and the diameter of the cross section. In turn, the stiffness assessment through beam bending or even the global displacement of the facility would mainly consider the linear response of the platform. For steel facilities like those considered in this study, i.e. made of tubular thin-walled members, the wall thickness plays an important role especially in how the local (and also global) strength is considered for purposes of comparison with the bow strength. The energy dissipation (strain energy, $E$) with respect to the relative strength is described in Figure 5-20 for three different ratios/parameters. The three parameters taken into consideration for the installation are the stiffness obtained by the displacement measured on the impact point upon the application of a unit load $\kappa_i$, the compactness factor $R_c$ from Equation 3-25 and
mentioned in [153] the plastic moment of the tube wall $m_p$ that is given, according to [84], as:

$$m_p = \frac{t^2 \sigma_y}{4} \quad (5-15)$$

whereas for the ship, a value $\kappa_\text{s}$ is assumed from the deformation behaviour as defined in section 5.3. For $\kappa_\text{s}$ only the deformed bow parts are considered. Thus, for brace impact in cases 19 and 20 $\kappa_\text{s} = \kappa_\text{b}$ (Figure 5-11) of the respective ship model. The differentiation between the cases is made for the influence of the buckling of braces near the impact point and for the collision type (joint or span impact).
Each pair of points corresponding to a different case is plotted symmetrically according to the respective parameter. While the energy dissipation is characterised in terms of the local response of the platform for $\frac{\kappa_s}{m_p}$ and $R_c$, for the ratio $\frac{\kappa_s}{\kappa_i}$ the structure global behaviour assumes higher preponderance. Although in each pair of graphics some
trends can be speculated, the dispersion that can be observed relative to these trends increases the degree of uncertainty if any of the tried assumptions were to be considered. The use of $R_c$ confirms the results from [153] that (for model 'S2') a value of $R_c > 1.6\sim1.7$ MN should be sufficient to cause the bulb to crush without significant denting of the tube. If the ship is assumed to be stronger, such as model 'S10', then a slight increase of $R_c$ would become more appropriate, even considering the stress distribution from the respective contact areas. The dissipation can however occur in other ways when the compactness requirement is fulfilled. In case 15, for instance, the installation is able to dissipate $\sim45\%$ of the total energy, owing to the platform dimensions and overall elastic response of the frame. Nevertheless, it must be noted that such case corresponds to an impact scenario where the joint is directly hit by the bulb. In turn, the buckling of adjacent braces does not seem to affect the trend for when $R_c$ is used.

By considering the ratio $\kappa_s/m_p$ some attention is drawn to the variation of the ship stiffness. The influence of the tube thickness is also emphasized, rather than its diameter or even length. In fact, if Equations 3-25 to 3-29 are taken into account, it is clear that more energy is dissipated by a tube when its thickness increases rather than with the decrease of the $D/t$ ratio through the decrease of the diameter. The respective graphics show some similarity with the qualitative curves of energy dissipation vs. the relative strength [3], being that for $30 < \kappa_s/m_p < 60$ the platform can dissipate from $\sim0$ to $\sim50\%$ of the energy. Again, this arises from the joint impacts where the frame global deformation plays an important role. Some lack of data is found for $65 < \kappa_s/m_p < 145$, mainly due to the tested thickness values on the platforms that range from 45 mm to 70 mm for the legs and 15 mm to 20 mm for braces. If for the ship case the length $H$ of the folds that also contribute for the energy absorption (Equations 5-1 and 5-2) is influenced by size and stiffening of the plates, for the installation case, the plate thickness $t$ almost determines whether the tube wall gets dented as well as it contributes more than any other geometrical factor to the platform local stiffness. This affirmation is of course valid for the ratios $L/D$ and $D/t$ and their range of values for the cases being analysed ($8 < L/D < 28$ and $17 < D/t < 40$), even though these values can often be adopted in the design of the steel frames of offshore facilities. For $\kappa_s/\kappa_i$, where more emphasis is given to the platform elastic stiffness at the collision point, the plot assumes approximately a linear trend, although with some dispersion due to either local denting or brace buckling.
in the smaller platform models. Therefore, $\kappa_s/\kappa_i$ is affected by the platform dimensions, directly related to the frame energy.

It must also be noted that the results plotted in Figure 5-20 only indicate the relative strain energy. The absolute values of internal energy necessary to collapse a member of an offshore platform or provoke certain damage on a ship hull shall be provided by an independent assessment of the respective collision scenario that is beyond the scope of this study.

The above observations lead to the conclusion that the adoption of a coupled analysis is very limited, i.e., other factors such as the platform size or the impact location should be taken into account prior to defining the right parameter or stiffness ratio for the estimation of the energy dissipation. Accounting for the influence of brace buckling, impact location or global frame deformation, the dimensionless energy of the platform are also analysed based on Equation 3-22. The numerical cases are presented and compared in Figure 5-21 by means of the relationship between the dimensionless energy $\lambda$ (Equation 3-22) of the platform and the dimensionless displacement $u/t$ of the platform struck zone. For comparison, the testing data from chapter 3 for isolated tubular members within a range of different dimensions, boundary conditions and axial loading are also included in the Figure. The numerical simulation cases performed in the current study are grouped according to the platform type, impact type (span or joint) and the contribution (or not) of any adjacent braces (leg impact) with plastic deformation to the platform energy dissipation.
For the calculation of $\lambda$ in scenarios of collision with a joint, the total length $L$ is taken as the adjacent leg span. This assumption is made since for the joint impact cases the ship bulb not only hits the joint, its upper part is also usually in contact with the span. Parameter $P_u$, as referred in Equation 3-22, stands for the plastic limit load applied at the mid-length of a clamped tube. Such value varies according to, for instance, the tube boundary conditions, where for pinned ends it is reduced to half. The length considered for the estimation of the dimensionless energy (Equation 3-22) of the platform is described as $L_1$, according to Figure 5-22. For the full platform models, the application of Equation 3-22 that establishes the relationship between the displacement of the contact area considers, besides the strain energy of the struck tube, any additional energy amounts dissipated over the remaining structure through the different mechanisms (brace buckling, frame vibrations, etc.). This includes the following particularities: firstly, the tube length associated with flexural behaviour might extend to
the lower adjacent span of the leg ($L_2$ in Figure 5-22). Despite not so significant, the contribution of the strain energy due to bending might not be restricted to only one leg span; the second point is the fact that brace buckling due to axial forces also contributes to the total energy absorbed by the platform and included in Equation 3-22, which mainly takes into account the effects due to local denting and beam bending; the same is observed for the frame overall displacement, that contributes to the total displacement measured at the contact point and with energy amounts that can become very significant depending on the platform type, as seen above. The contribution of the bent brace due to local axial forces is not significantly reflected in the linear relationship between $\lambda$ and $u/t$, contrary to the platform size that can be closely associated to the frame elastic energy. This reinforces the idea that performing a single analysis that accounts for both local and global mechanisms, aiming to estimate the energy dissipation of the complete system (ship + platform), does not appear possible. On the other hand, the evaluation of the platform response seems to be possible with acceptable accuracy if preliminary assessment of the platform local strength at the impact point for the respective impact action is carried out. The uncoupled analyses of SDOF for global frame response and/or hand calculations for the tube flexural deformation are then performed according to the predicted/pre-established conditions.

![Figure 5-22 Substructures considered for definition of $P_u$](image-url)
Finally, regarding the adoption of SDOF systems with equivalent stiffness for evaluating the energy absorption of the loaded structures besides the local response, the considered case studies demonstrate that the SDOF approach can well predict the energy absorption during the loading stage, where the maximum rotation/displacement is reached as well as the total energy absorption. This is valid as long as the localized plastic mechanisms are not very significant. However, when making use of such hand calculations, the residual values of displacement/rotation obtained after the impact do not reflect the energy that remains in the structure. It is believed that during the forced vibration period some regions across the installation, especially near the joints, will yield, thus storing part of the strain energy from elastic vibrations. Yet, such deformations do not seem to affect much the final displacement of the platform, at least if the final position of the deck is considered. The total energy taken by the structure would therefore be better predicted if an estimated reduction of the maximum strain energy achieved during the loading would be taken into consideration.

5.6 Summary

In this chapter the results of a series of detailed FEM simulations of the impact between vessels and fixed offshore steel platforms are discussed with respect to the energy dissipation. Predicting the size increase of supply vessels in the near future, as well as possible ship impacts involving higher energy amounts than those usually considered by the current design practice, both loading curves and simplified equivalent systems representative for ships up to 25000 DWT were derived from the FEA. Aspects such as the steel grades or the scantling size variation were considered in the analyses for the purpose of broadening the scope of ship types/categories.

Because the platform response to strong impacts might involve yielding of other members besides those directly affected by the contact with the ship, different scenarios were defined involving different plastic mechanisms and possible combinations among these mechanisms. It has been revealed that the plastic energy absorbed by the platform can be evaluated and will mostly depend on the impact area, but also include contributions from deformations that can take place in adjacent members. The thickness of the installation tubes, in particular of the struck tubes, has been shown to have some degree of connection to the platform internal energy after-impact and therefore the plastic moment of the tube walls can be used to describe the platform relative strength,
depending on the platform dimensions as well, since elastic strain energy gains importance for big steel platforms and for joint impacts. The interaction between the vessel and the platform appears to be important, particularly in cases in which the energy share is closer to 50/50 corresponding to significant energy amounts and significant plastic deformation/damage in both structures. For the ship range considered, these are observed mainly when the tube thickness is in $45 \text{mm} < t < 70 \text{mm}$. However, it should be noted that the influences of tube cross section shape are not considered since the membrane denting depends upon the load distribution and the respective concentrated stress areas, which is not reported in this paper.

In conclusion, if plastic analysis is to be considered in the design practice against extreme accidental loads from higher energy ship impacts, the present results might provide some good indications especially regarding the global integrity of the structure and the ship-installation interaction. However, this can be better complemented with additional investigation on the individual failure of members based on energy absorption since that is only addressed generally in the current study.

The soil-structure interaction may also have some influence on the entire process of energy absorption. This issue is briefly addressed in the following chapter.
Chapter 6
Dynamic Analysis of Ship Impact on Fixed Offshore Platforms Considering Soil-Structure Interaction

6.1 Introduction
All the numerical simulations performed in this study have been carried out by pinning the platforms to the seabed. The sensitivity of foundation modelling of platform under ship impact should therefore also be evaluated. By explicitly modelling the foundations, the platform models have their first natural frequency slightly lowered, meaning that their dynamic response can be more significant when compared to the case when the foundation flexibilities are neglected. The differences might also extend to the energy dissipation process, since different stiffness ratios between vessel/installation should also result in different shared amounts of strain energy, including any local mechanisms as well, such as denting.

The way how the foundation modelling can affect the structural response has been investigated in [154] for light weight platforms (commonly referred to as minimum structures), in which four models, namely a braced caisson, a 4-pile jacket, a monotower and a vierendeel (see Figure 6-1), and several collision cases are considered. It was found that in general a higher degree of damage and plastic energy dissipation was predicted in the impacted members when pinning the platforms to the seabed. The ship used in the numerical simulations had a mass of 3500 tons and an initial speed of 2 ms\(^{-1}\).
In this chapter, similar analysis is carried out for conventional jackets, higher in redundancy, using the numerical models developed in the previous sections.

6.2 Pile and SSI modelling

Pile foundations will be subjected to shear forces, during and right after the impact, that are transmitted through the frame. These can be the governing design constraint for single piles and pile groups supporting the different offshore platform types. The piles are assumed to be vertical and flexible with circular cross section. These are represented by beam elements. The conventional above water system (Figure 2-2) is assumed. The pile properties are given in Table 6-1.
Table 6-1 Piles properties

<table>
<thead>
<tr>
<th>Platform</th>
<th>$E$ [GPa]</th>
<th>$\sigma_y$ [MPa]</th>
<th>$E_t$ [MPa]</th>
<th>Penetration [m]</th>
<th>$D$ [m]</th>
<th>$t$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>P4</td>
<td>200</td>
<td>345</td>
<td>4830</td>
<td>90</td>
<td>1.55</td>
<td>60</td>
</tr>
<tr>
<td>P8</td>
<td>200</td>
<td>345</td>
<td>4830</td>
<td>90</td>
<td>1.15</td>
<td>60</td>
</tr>
</tbody>
</table>

Modelling the load-deformation behaviour of soil requires soil properties to be determined. Therefore, the soil profile described in Figure 6-2 is assumed for the considered cases. The soil is divided into $n$ layers with different soil properties assigned to each layer according to the soil profile considered. Based on the Winkler assumption [156] that the soil-pile interaction resistance at any depth is related to the pile-shaft displacement at that depth only, independent of the interaction resistances above and below, a set of discrete elements is connected to the pile elements with vertical spacing of 2m. At each interval of 2m, the pile nodes are connected to two horizontal springs with non-linear p-y behaviour (lateral resistance) and one vertical nonlinear t-z spring that accounts for the effects of skin friction. At the bottom of each pile a vertical q-z spring is applied to model the bearing resistance. The springs are modelled using the LS-DYNA non-linear spring material *MAT_SPRING_-GENERAL_NONLINEAR which allows specification of separate loading and unloading curves describing the force versus displacement relationship for the spring. Due to their smaller thickness, and for reasons of simplification, the sand layers are neglected and the soil properties of the upper and lower neighbour clay layers are given instead.
The nonlinear p-y behaviour is modelled using the element described in [157], which accounts for gapping and radiation damping. The p-y parameters for the soft clay as well as $P_{\text{ult}}$ and $\gamma_{50}$ are based on Matlock’s [158, 159] equations, which are as follows:

$$P_{\text{ult}} = C_u D N_p$$  \hspace{1cm} (6-1)

$$N_p = \left(3 + \frac{\gamma' z'}{C_u} + \frac{J z'}{D}\right) \leq 9$$ \hspace{1cm} (6-2)

$$\gamma_{50} = 2.5 D \varepsilon_{50}$$ \hspace{1cm} (6-3)

where $D$ is the pile diameter, $N_p$ the lateral bearing capacity factor, $\gamma'$ the average buoyant unit weight, $z'$ the depth, $C_u$ the undrained shear strength and $\varepsilon_{50}$ the strain corresponding to a stress of 50% of the ultimate stress in a laboratory stress-strain curve, and $J$ is taken according to the recommendations for soft clay in [158, 159]. For stiff clay the ultimate bearing capacity would vary between $8C_u$ and $12C_u$. The p-y load-deflection curves are based on the API [1] recommendations. The ultimate skin friction resistance of the t-z elements in the clay was calculated using the method proposed in

Figure 6-2 Soil layers and properties
the API [1], with the shaft friction being represented with $f = \Omega C_u$ where $\Omega$ is equal to unity. The q-z elements are also modelled using the API recommendations, with the ultimate bearing capacity being given by $q = 9C_u A$, where $A$ is the area of the pile-cross section at the tip.

The discretised models of platforms ‘P4’ and ‘P8’ with the foundations included, used for analysis, are shown in Figure 6-3.

![Model P4 and Model P8](image)

Figure 6-3 Illustration of platforms FEM with piles included (Note: only one set of springs per part and per pile is enabled in the picture)
6.3 Numerical analysis and discussion

The developing of the FEM becomes very time consuming, especially when the structures are modelled with a very high level of detail. Furthermore, the comparison between the cases using pinned and piled installations requires the simulations to be done for a longer time in order to also capture the free vibration phase. Therefore, for each of platform models ‘P4’ and ‘P8’ only one case study as defined in Chapter 5 for each platform model is selected, i.e., cases 6 and 13, respectively. The two selected collision cases have in common the fact that the response of the platforms is mainly governed by the frame elastic deformations i.e., neither significant denting of the stuck legs nor plastic deformation and buckling of adjacent bracings is observed. This means that higher load actions are transmitted through the steel frame to the soil. Another important parameter that is taken into consideration in selecting the study cases is the different height of the installations. Since platforms ‘P4’ and ‘P8’ operate at different water depths, the impact points in the two situations would be at different heights relative to the sea bed, with model ‘P4’ expected to be less compliant than ‘P8’. The comparison between the pinned and piled models for the two cases is made by means of the contact force, deformation and energy dissipation (Figures 6-4 to 6-8). In Figures 6-5 and 6-6 the displacement of the rigid deck is compared assuming its absolute and relative displacement towards the seabed.

![Figure 6-4 Contact force for pinned and piled platform models](image)
Figure 6-5 Deck total displacement for pinned and piled platform models

Figure 6-6 Deck relative displacement with respect to the seabed for pinned and piled platform models
Figure 6-7 Strain energy dissipation for collision with pinned and piled platforms
As can be noticed above, the primary influence of including foundation in the model is on the absolute displacement of the topside decks, where small increase of the oscillation period can also be observed. Despite not so significant, the maximum deck shifts relative to the seabed are also higher for piled platforms. Such differences appear to decrease the bigger the platforms become, as a result of their higher compliance. The increase of the absolute deck displacement is 260% for case ‘P4’ and 70% for ‘P8’, whereas the relative displacement increase rates are 70% and 30% for ‘P4’ and ‘P8’ respectively. The influence of local plastic deformations cannot be analysed from the selected case studies, although it is believed that the extension of dents on the tube wall and plastic energy dissipation occurred in the impacted members would be smaller for
piled platforms and such difference would be even greater for shallow water jackets such as model ‘P4’.

The immediate system response to the collision does not seem to be affected by the inclusion of the foundations in the models as shown in Figure 6-4, where the predicted impact forces with or without considering SSI are also most the same. As for the energy balance, the installation ends up taking more energy when the piling system is modelled due to the wave propagation that later reaches the soil. This is visible in Figure 6-8, in which the platform displacement at the contact point is still rising after collision. The integration of the added segment of the platform P-u curve roughly represents the additional amount of internal energy of the facility, subtracted from the P-s curve in Figure 6-8. The internal energy of the platform elements and the piles is, for cases ‘P4’ and ‘P8’, 2.5 MJ and 2.3 MJ. Such amounts are nearly the same and represent, for case ‘P8’, the difference between the internal energy amounts for pinned and piled platform cases. For the ship impact with model ‘P4’, such difference is higher than 2.5 MJ (Figure 6-7), but so is the difference observed between the maximum $u$ values, which might be related to the elastic strain energy of the frame.

In conclusion, the neglecting of pile foundation might underestimate the displacement of topside structures, while the energy absorbed by the installations and local damage might be overestimated, providing a conservative predictions of platform responses to ship impact. In order to make the additional note given in this chapter more widely applicable, more numerical simulations should also be carried out to include collision cases where local deformations are more significant as well as the cases with different soil profiles.
Chapter 7

Concluding Remarks

7.1 Main findings

The ongoing growth and development of the offshore industry has been demanding innovative solutions to face the diversity of offshore environments where industry operations can be performed. Consequently, despite they can be grouped according to their type, offshore facilities can still diverge in terms of their dimension and the dimension of the structural members individually. Other factors that also differ are the mechanical properties of the platform components, the way they are connected, and their interaction with the seabed. Similarly, supply vessels and other passing ship types might have very different sizes, internal structures and material strengths. Combining together, these factors result in a reasonable number of possibilities that affect the dynamic responses and damages of both the platform and ship structures upon collisions, which make the derivations of general solutions difficult, especially for cases with high amounts of energy involved such that large plastic deformations are to take place. For practical engineering problems, simple hand calculations provided by design codes mainly cover accidents with small supply vessels at low approaching speeds, which collision with offshore platforms has a higher likelihood. The major contributions and findings made in this research, however, are related to heads on impacts of large ships on offshore platforms, in view of the world's fleet increase, as well as the intensification of industrial activities offshore, and the resulting possibilities of
collisions between offshore facilities and passing vessels and bigger supply vessels or others.

In Chapter 3 the response of steel tubular members of jacket platforms is analyzed by means of FEM for penetration of a deformable ship bow. Special emphasis is given to the ship-platform local interaction accounting the plastic deformation mechanisms formed in both structures. The numerical results from LS-DYNA explicit solver provide new force-deformation relationships for bulbous bows of typical supply-vessels (2000-5000 tons) as alternative to the more conservative DnV curves [3]. The denting of steel tubes is also shown to be of easier estimation if the energy-indentation relationship from Equation 3-25 is used. Energy-deformation relationships can account for the ship-structure interaction. Equation 3-25 also considers effects such as the axial stress on the member and the length of the dent, although it is limited for irregular dent shapes provoked by irregular shape strikes such as ship bows. A linear relationship is derived, from extensive parametric studies, for the strain energy and the lateral displacement of the tube wall regardless of the deformation behaviour (denting and beam bending) or other parameters such as the strain rate effects, impact velocity, boundary conditions, cross-section properties or axial preloading.

In Chapter 4 the dynamic analysis is carried out for the entire platform models against high energy ship impacts, where the influence of the topside deck inertia effects are observed and simplified hand calculations are derived from the FEM. It is shown that the use of the proposed equivalent systems with a reduced number of DOF’s can provide accurate results at significantly less computational efforts as compared to the FE simulations. The derivation of some parameters of the equivalent dynamic elastic-plastic SDOF/2SDOF systems however needs to consider the complexity of the analysed steel frames and perform preliminary non-linear static analyses. Therefore further studies of different impact scenarios on platforms with different configurations are recommended to augment the results presented.

The results from various ship and platform models in different collision scenarios are compared in terms of of the strain energy dissipation with respect to the different ship/installation strength ratios in Chapter 5. From the FEA simplified approaches are also derived in terms of the relative stiffness of the two structures for assessing the responses and energy absorptions of the two structures. The conclusions drawn can be
applied to a broader range of collision assessment of fixed offshore steel platforms subjected to high-energy ship impacts.

Chapter 6 constitutes an additional note with respect to the influence of the foundation modelling. From the selected collision cases, it has been shown that neglecting the piling system and the soil-pile interaction effects for high energy ship impact may underestimate the response of the topside deck structures in terms of their maximum displacement, especially for stiffer and shallow water jacket frames.

7.2 Recommendations for future work

Although a wide range of scenarios and several aspects have been covered in Chapters 2 to 5, the current study has focused essentially on the response of fixed steel platforms as the overwhelming majority of the world's offshore installation types. The conclusions from the addressed topics can always be made more general if further collision cases using additional ship/installation models and respective parametric studies are considered. Such procedure will of course be very demanding in terms of time and CPU costs.

For collision scenarios which have not been explicitly addressed in the present work, the current results can be extrapolated to some extent:

1. For lateral or stern impacts, regardless of the impact speeds that are lower than those considered in this work, the prediction of the ship deformation can be covered by the current plastic deformation methodologies that make use of plate intersections mentioned in Chapters 2 and 3. For the penetration of the ship hull by the structure, the DnV requirements [3] quantify local concentrated collision forces evenly distributed over regular areas, as presented and discussed in Chapter 3 for bow impacts. For avoiding local denting on tubular members the DnV requirements are given with the assumption of concentrated load applications, i.e., the denting extension is approximately zero-valued, which is always conservative for any impact scenario. On the other hand, the vessels used to derive the DnV formulas are from the 1990s and used to have raked bows with no bulb. It is also important to refer that the expected increase in the requirement for the kinetic energy from accidents will place heavy demands on the energy dissipation, which becomes particularly relevant for jackets for which there is a limited potential for energy
dissipation in their members. The current study suggests that the guidelines must therefore be revised, also in view of significant increases in the supply vessel sizes and strength (ice reinforced supply vessels) and a wide variety of ship configurations;

2. If other platform types are considered for high-energy ship impact, such as floating units, the principles of energy dissipation shall always take into account the relative stiffness between the two structures. Differences can be found with respect to the width of the contact areas. If larger contact areas are involved, results obtained in Chapters 3 and 5 for impacts against rigid walls can be taken as reference provided that the ‘strength design’ principle (‘rigid’ platform) is adequate. This should be verified according to the thickness of the installation plates (if steel) or the material strength (steel or concrete). For local denting on larger diameter tubes/columns or wide steel plates further investigation may be required. In the particular case of floating platforms (TLP’s or similar), particular attention must be also given to the outer mechanics problem, where the ship initial energy must also account for the correct hydrodynamic added mass factors. Yet, assumptions may be possible to be taken from ship-ship collision problems in the current literature. When the struck object is a floating platform anchored to the seabed, the equations of motion expressed in terms of the translations and the rotation of the geometrical centre of the platform are analogous to the equations of motion for the ship. The main difference is that for collisions that result in large amplitude motions it may be necessary to take into account the restoring forces from anchoring system (Figure 7-1);

3. In regards to the foundation modelling, the analysis carried out in Chapter 6 can be enriched if additional case studies considering the effects of localized plastic deformations, as well as different soil profiles, are performed.
Another aspect which has not been addressed and that might deserve some investigation is the use of protecting devices in order to mitigate the effects of ship impact. Similar studies in the literature can be found for offshore wind farms [160] or bridge piers [161, 162] concerning the energy dissipation through the use of such devices. For the bridge case, energy-dissipating crashworthy devices consisting of hundreds of steel-wire-rope coil (SWRC) (Figures 7-2 and 7-3) connected in parallel and series are studied for a Chinese standard steel bulk carrier with a length of 182 m and an impact speed of 4 m/s. Such devices are shown to be responsible for nearly 72% of the energy dissipation while the ship only takes 28%. Another finding is that there is no great difference between the ship kinetic energy before and after the collision since 81% of the kinetic energy is carried off by the turned-away ship after the actual end of the collision. As for the offshore wind-farm case (Figure 7-4), a 'rigid' 3000 ton ship class model at a speed of 2 m/s and a typical 3MW offshore wind turbine are considered. The used device, a sphere shell of 10 m diameter and a thickness of 10 mm, has good performance both for isotropy and defending impact loads. The device is shown to be able to absorb most of the impact energy due to its structural deformation, and shows also good performance for ship side impact due to its isotropy character. The capability of the slippery outer surface of the sphere to reduce the structural damage is also demonstrated.
Figure 7-2 Numerical model of a part of the steel structure of SWRC crashworthy device [161]

Figure 7-3 Schematics of ship-bridge collision using a crashworthy device
With respect to the platform modelling, it must be mentioned that the study of tubular connections is becoming of great importance in recent years. Individual full scale tubular connections have been modelled and analysed by means of FE in [163] and [164] to investigate the effect of joint flexibility on overall behaviour of offshore installations. The effect of joint flexibilities becomes more apparent when a structure undergoes deformation beyond elastic region and shows nonlinear behaviour. For typical jacket frames the primary period of vibration can also increase more than 10% if non rigid joints are modelled. Nonlinear dynamic analyses also indicate the fact that platforms with flexible connections show higher displacements and inter-story drifts.
and lower base shear due to less stiffness and strength of the jacket structure. Such aspects also deserve some attention under the ship impact problem.

To conclude, in future studies of this novel panel, the detail mechanics of the dynamic deformation would be interesting and useful. Also, ductile tearing of such structure may be important and may be studied.
References

References


[19] Vasco Costa F, The berthing ship, the effect on impact on the design of fenders and other structures, the Dock and Harbour Authority, 1964.


[57] Consolazio, GR, Cook, RA, Lehr GB, Barge impact testing of the St George Island causeway bridge phase I: Feasibility study, Univ. of Florida, Gainesville, Fla, 2002.


References


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Appendix

A.1 Structural details of vessels and offshore installations analysed

*Ship ‘S20’*

![Figure A-1 Bow ‘S20’](image)

**General Dimensions**

- Length – 174 m
- Breadth – 29.5 m
- Height – 17 m
- Mass (+5% of surge added mass included) – 28200 ton

**Scantlings of unstiffened bow** (with smeared out thickness of equivalent stiffened bow with large scantlings)

- 1st, 3rd and 4th decks – plate thickness 15 mm
- 2nd and 5th decks – plate thickness 19 mm
- Lower and upper side shell – plate thickness 42 mm
Appendix

Mid side shell – plate thickness 21 mm
Girders – 11 mm
Frames – spacing 0.7 m, 12 mm
CL girder (bottom) -16 mm

Ship ‘S10’

Figure A-2 Bow ‘S10’

General Dimensions
Length – 130 m
Breadth – 19 m
Height – 15 m
Mass (+5% of surge added mass included) – 13300 ton
Radius of inertia for yaw – 37.0 m

Normal scantlings
Longitudinals – spacing 0.8m, L500x100x9/11 mm
1st, 2nd and 4th decks – plate thickness 11 mm
3rd and 5th decks – plate thickness 13 mm
Lower and upper side shell – plate thickness 13mm
Mid side shell – plate thickness 11 mm
CL girder – L1400x250x11x13 mm

Larger scantlings
Longitudinals – spacing 0.8m, L500x100x11/11 mm
**Appendix**

1\(^\text{st}\), 2\(^\text{nd}\) and 4\(^\text{th}\) decks – plate thickness 11 mm  
3\(^\text{rd}\) and 5\(^\text{th}\) decks – plate thickness 13 mm  
Lower and upper side shell – plate thickness 18 mm  
Mid side shell – plate thickness 13 mm  
CL girder – L1400x250x14x18 mm

Scantlings of unstiffened bow (with smeared out thickness of equivalent stiffened bow with larger scantlings)  
1\(^\text{st}\), 2\(^\text{nd}\) and 4\(^\text{th}\) decks – plate thickness 16 mm  
3\(^\text{rd}\) and 5\(^\text{th}\) decks – plate thickness 18 mm  
Lower and upper side shell – plate thickness 26 mm  
Mid side shell – plate thickness 21 mm  
CL girder – L1400x250x14x18 mm

**Ship ‘S2’**

![Figure A-3 Bow ‘S2’](image)

**General Dimensions**

Length – 63.5 m  
Breadth – 13.3 m  
Height – 6.3 m  
Mass (+5% of surge added mass included) – 3000 ton

**Normal scantlings**

Longitudinals – spacing 0.8m  
Plate thickness – 9 mm  
Shell thickness – 11 mm
Larger scantlings
Longitudinals – spacing 0.8m
Plate thickness – 9 mm
Shell thickness – 14 mm

Platform ‘P3’

Figure A-4 Platform ‘P3’ (Dimensions in meters)
Platform ‘P4’

Side views

Section A-A
D 0.85

Section B-B
D 0.71

Section C-C
15.9
D 0.65

Section D-D
D 0.50

Figure A-5 Platform ‘P4’ (Dimensions in meters)
Platform ‘P8’

Figure A-6a Platform ‘P8’ (Dimensions in meters)
Figure A-6b Platform ‘P8’ (Dimensions in meters)
NOTE: The thickness of the tubular members for all the platforms is variable according to the collision cases.