Analytical formulations for ship-offshore wind turbine collisions

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Where I have consulted the published work of others, this is always clearly attributed.

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I have acknowledged all main sources of help.

Where the thesis is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself.

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ABSTRACT

Owing to the changing of climate, environmental pollutions and the energy supply are getting short of demand, renewable energy has been gradually emphasized by many of the countries around the world nowadays. The wind energy is one of the highly developed renewable energy which has certain efficiency and potential. Among onshore and offshore wind turbine, offshore is more efficient but also more challenging. One of the most common offshore wind turbine, jacket type, is currently facing some challenges which are the collisions between its foundation and an OSV (Offshore Supply Vessel) or passing ships. Therefore, the crashworthiness of the foundation should be assessed somehow with a certain accuracy.

In order to avoid expensive and time-consuming analysis in preliminary design stage, a simplified analytical method has already been developed which is based on the concept of super-element. The oblique impacted leg (or brace) damage is studied by analyzing two super-elements corresponding to horizontal and vertical cylinders. The absorbed energy and impact load of the impacted cylinder in ship-jacket collision issue are able to be predicted rapidly and accurately in preliminary design stage by using this method. Nevertheless, not only the impacted member which absorbs nearly 60% of the total energy is involved but also some of the other members of the jacket foundation. The rear legs are especially on influence as they may be punched by the displacement of the impacted leg through the braces.

The aim of the thesis is to improve the simplified analytical method by developing a new super-element in order to account for the punching phenomena in the super-element method. Based on the concept of virtual work, the external energy is transferred to the internal energy associated with the deformation of the cylinders. The crushing resistance of a punched cylinder is then derived by choosing a realistic displacement field (related to the punching phenomenon) and by calculating the corresponding internal energy.

In addition, numerical solutions from non-linear finite element simulations performed using the finite element code LS-DYNA are used for validating the analytical formulations. The validations are first starting with simple tubular joint and then a real jacket structure is considered. In both cases, good agreement is found between super-element and LS-DYNA simulations.

In conclusion, discrepancy of total absorbed energy between numerical and analytical calculations is still appreciable. In addition to the energy dissipated by crushing of the impacted and the punched members, some part of the striking ship kinetic energy is also dissipated by buckling of the braces and shearing of the legs near the mud line on occasion. Therefore, these tasks are proposed to be studied further for a more accurate assessment of the total absorbed energy of the jacket.
摘要

由於氣候變遷、環境汙染和能源短缺的因素，現今再生能源逐漸被各國重視。風能為眾多高度發展的再生能源之一，其具有相當的效率及潛能。風能又可分為陸上風電與離岸風電，相較於陸上風電，離岸風電效率較高但也較具挑戰性。現今，jacket基座，一種常見的離岸風電基座，正面臨與OSV(Offshore Supply Vessel)或行經基座鄰近海域之船隻的碰撞問題。因此，有效準確的評估其防撞性為最重要的課題之一。

為避免在初步設計階段耗費過多成本及時間，一種依據super-element概念的簡化解析分析方法已被開發。運用super-element的概念將被船隻撞擊的主支柱，為一傾斜的圓柱，分為垂直及水平兩種圓柱來分析再將其結果疊合。運用此方法使得被碰撞支柱所吸收的能量及碰撞力得以迅速地被預測，並具有一定的準確性。然而，被碰撞的支柱僅吸收了約60%船的動量，其餘的能量則被其他基座之結構所吸收。鄰近的支柱(rear leg)則是首當其衝，被撞擊的支柱可能透過支架(brace)衝擊(punched)鄰近的支柱。

此論文的目的為開發出一個新的super-element，考慮鄰近支柱被支柱衝擊所吸收的能量，以提升現有之簡化解析法的準確度。基於虛功原理，外能即船的動能，被轉換為以使圓柱產生變形之內能。正確的選擇變形域(displacement field)可準確的預估基座於在不同時間點所吸收的能量。衝擊圓柱之破壞力可以從計算對應之內能及選擇符合實際情況之變形域求得。除此之外，利用非線性有限元素模擬軟體LS-DYNA來驗證此解析公式。首先，驗證簡易的管狀接合結構，確認解析公式的正確性後，接著考慮jacket基座。在此兩種不同被驗證的實例中，解析公式和LS-DYNA模擬結果皆相當一致。

最後，比較由解析公式及數值模擬所求得的總能量，兩者仍有一定程度的差異。除被撞擊及被衝擊支柱可吸收船隻動能外；在某些情形下，支架的挫折及基座根部的剪切也吸收船隻部分動能。因此，為能夠更準確評估基座吸收之總能量，未來應對支架挫折及基座根部的剪切做深入的研究。
1 INTRODUCTION

1.1 Context of the Study

Human beings are faced with the climate changing, depletion of non-renewable energy and increasing of fuel prices, so more and more renewable energy have been developing nowadays. The European Union 2020 Goal indicates that the EU must get 20% of its energy from renewables, have reduced its greenhouse emissions by 20% and its energy consumption by 20% through increase in energy efficiency.

The wind energy is one of the well-developed resources, which is second only to the hydropower. By European Wind Energy Association (EWEA), the 2009 EU renewable Energy Directive aims to increase the share of renewable energy in the EU from 8.6% in 2005 to 20% in 2020 (EWEA 2009)[1]. The wind energy capacity target for 2020, 2030 and final target in 2050 are depicted in Fig. 1-1.

<table>
<thead>
<tr>
<th>Year</th>
<th>Onshore Wind (GW)</th>
<th>Offshore Wind (GW)</th>
<th>Total Wind Energy Capacity (GW)</th>
<th>Average Capacity Factor Onshore</th>
<th>Average Capacity Factor Offshore</th>
<th>TWh Onshore</th>
<th>TWh Offshore</th>
<th>TWh Total</th>
<th>EU-27 Gross Electricity Consumption*</th>
<th>Wind Power's Share of Electricity Demand*</th>
</tr>
</thead>
<tbody>
<tr>
<td>2020</td>
<td>210</td>
<td>55</td>
<td>265</td>
<td>26.0%</td>
<td>42.3%</td>
<td>479</td>
<td>204</td>
<td>683</td>
<td>3,494</td>
<td>20%</td>
</tr>
<tr>
<td>2030</td>
<td>250</td>
<td>150</td>
<td>400</td>
<td>27.0%</td>
<td>42.8%</td>
<td>592</td>
<td>563</td>
<td>1,155</td>
<td>3,368</td>
<td>34%</td>
</tr>
<tr>
<td>2050</td>
<td>250</td>
<td>350</td>
<td>600</td>
<td>29.0%</td>
<td>45.0%</td>
<td>635</td>
<td>1,380</td>
<td>2,015</td>
<td>4,000</td>
<td>50%</td>
</tr>
</tbody>
</table>

Figure 1-1 Wind energy capacity target in 2020, 2030 and 2050 [1]

The wind energy can be classified into two main groups: onshore and offshore. The proportion of onshore and offshore are predicted by EWEA as shown in Fig.1-2. As can be seen the onshore wind energy is becoming saturation toward 2030 and offshore wind energy is in exponential growth. Therefore, the development of offshore wind energy is expectable even beyond 2030. Among these two sort of wind energy, offshore wind turbine can possess higher capacity than onshore wind turbine, less restriction of space and also less limitation of the noise. Nevertheless, there are also more challenges such as the installation and the ability to withstand the stronger wind.
Offshore wind turbine is classified by the type of the foundation, which are fixed foundation and floating foundation. The main effect factor is the depth of the sea bed, the different types of offshore wind foundations are shown in Fig.1-3.

For the fixed foundation, the most common type is monopile which is less costs and easier for installation and then following by tripod and jacket. Several different types of fixed foundation for offshore wind turbine are shown in Fig.1-4. Jacket foundation possesses 6% of fixed foundation which combine high global stiffness with low structural mass. The jacket foundation is focused in this project, the detail sketch is depicted in Fig.1-5.
Ship collision is one of the major hazards of the jacket foundation and constitutes a considerable threat to the environment. Either by an offshore supply ship or a passing ship such as bulk carrier, tanker and etc. The jacket structure is very stiff globally, but its member substructures are rather fragile. The phenomena can be seen in the research of collision of ship with OWT by (Biehl and Lehmann, 2006 [4]), during a collision of the considered double hull tanker with the jacket, it destroys the substructure by first tearing it off its bearings and then penetrates the jacket about 10 to 14m. In order to assess the crashworthiness of the jacket structure rapidly in preliminary design stage, it is worth to develop a simplified tool which can achieve the goal efficiently and with a certain extent of accuracy at the same time.
Many researches have been done and this thesis is involved in an ongoing project called CHARGEOL, which has already developed main portion of the simplified tool in C++ programming for predicting the crashworthiness rapidly, the framework of the simplified tool in C++ is shown in Fig.1-6. The target of CHARGEOL project is to study the behavior of the offshore wind turbine foundation while sustained ship collision impact, strong wave impact. The partners of CHARGEOL project are shown as below:

- STX FRANCE SOLUTION
- BUREAU VERITAS
- Hydrocean
- GEM Laboratory (ECN)
- IFFSTAR Laboratory
- ICAM

Figure 1-6 The framework of the simplified analytical program  
(CHARGEOL project presentation by Prof. Le SOURNE)
1.2 Objective of the Study

Some numerical results of jacket crashworthiness obtained by (Le Sourne, Barrera and Maliakel, 2014 [5]) have shown that the impacted leg dissipates approximately 60% of the total energy. Nevertheless, the plastic strain appears not only on the impacted leg but also on the rear leg and all other legs, which dissipate around 15% and 25% respectively (Fig.1-7). The punching problem has been found on the rear leg, as can be seen in Fig.1-8. The rear legs are punched through the connected braces and the dissipation of punching is around 15% of total energy, which is non-negligible. Therefore, it is necessary to take into account this phenomena while developing the simplified tool. In current phase, one additional super-element will be developed and implemented in the simplified tool COLEOL for improving the assessment of the dissipated energy. The sketch of difference between previous stage and current stage is shown in Fig.1-9.

![Internal energy and crushing force](image1.png)

**Figure 1-7**
Internal energy and crushing force [5]

![Punching scenario](image2.png)

**Figure 1-8** Punching scenario [5]

![Progress of the project](image3.png)

**Figure 1-9** Progress of the project (Buldgen et al. 20 [6])
2 THEORETICAL AND TECHNICAL BACKGROUND

2.1 Virtual Work

According to the principle of virtual work, the equilibrium of the energy rate is expressed as below:

$$\dot{E}_{ext} = \dot{E}_{int}$$  \hspace{1cm} (2-1)

For the external energy rate, only the lateral load applying on the cylinder is considered in our case. Therefore, the external energy is expressed by

$$\dot{E}_{ext} = P \delta$$  \hspace{1cm} (2-2)

The concept of external energy can be illustrated by calculating the area under the curve of the obtained load function as the simple sketch in Fig. 2-1.

![Figure 2-1 Concept of external energy](image)

On the other hand, the internal energy is associated with the deformation of the cylinder and can be expressed by

$$\dot{E}_{int} = \iiint_V \sigma_{ij} \cdot \dot{\varepsilon}_{ij} \cdot dV$$  \hspace{1cm} (2-3)

where $V$ is the volume of the solid body,

$\sigma_{ij}$ is the stress tensor, and $\dot{\varepsilon}_{ij}$ is the strain tensor.
2.2 Super-element

The principle of the super-element method consists in dividing the whole structure system into several sub-structures and then determining the relationship at the interface between sub-structures. As mentioned in (Le Sourne et al. [7]), for the ship collision, the total energy dissipated by internal mechanics is obtained by summation of all crushed super-elements. Several recent studies relating to ship or offshore structure collision based on the super-element method are introduced here. The analytical damage prediction of a ship impacted perpendicularly by rigid bow of another ship is proposed by (Lützen et al., 2000 [8]). The struck ship is divided into four super-elements which are:

- The 4 edges simply supported plate which is subjected to out-of-plane deformation – for outer side plating and longitudinal bulkhead, the sketch is shown in Fig.2-2 (a).
- Free edge of the plate which is subjected to in-plane load (3 edges simply supported, 1 free edge)- for the decks, transverse bulkheads, web girders, frames, bottom and inner bottom, the sketch is shown in Fig.2-2 (b).
- A beam which sustains a perpendicular transverse load- for small stiffeners, as shown in Fig. 2-2 (c).
- X-T-L form intersection- for the junctions, as the sketch in Fig. 2-2 (d).

![Super element diagrams](image_url)
Later on, super-elements for an oblique ship collision and inclined ship side are proposed by (Buldgen et al. 2012 [9] & 2013 [10]). For the oblique ship collision, the ship is divided into six super-elements and the crushing force is obtained by following the same approach as Lützen. A new super element is also developed for inclined ship side impacted by another ship, the scenario is shown in the Fig. 2-3.

A simplified tool based on super-element method for assessing oblique cylinder damage in the case of ship collision against a jacket is proposed by (Buldgen et al. 2014 [6]), the impact scenario is shown in Fig. 2-4. The jacket structure is divided into two types of super-elements, which are vertical and horizontal cylinders impacted by a ship respectively. The crushing force of an oblique cylinder impacted by the bow or by the bulb of a striking ship is obtained by linear interpolation of the force calculated on vertical and horizontal cylinders. This thesis is strongly related to reference above-mentioned and it is completed by using the same concept and modifying the analytical solutions in order to treat the punching phenomenon.

### 2.3 Upper-Bound Method

Following the principle of solid mechanics, three conditions should be obeyed, which are equilibrium equations, conditions of compatibility between strains and displacements, and constitutive law (stress-strain relations). There are two approximate solutions, lower-bound and upper-bound, which are very useful in engineering. An approximation solutions for large deformation of rigid-plastic solid in plane strain, the yield condition and boundary condition also need to be satisfied (MIT OpenCourseWare Mechanical Engineering 2004 [11]).

In the reference (Norman Jones 2012 [12]), the statement of the two approximate solutions are explained as below:
Analytical formulations for ship-offshore wind turbine collisions

- Lower bound: “If any system of generalized stresses can be found throughout a structure which is in equilibrium with the applied loads and which nowhere violates the yield condition, then the structure will not collapse, or is at the point of collapse.”

- Upper bound: “If the work rate of a system of applied loads during any kinematically admissible collapse of a structure is equated to the corresponding internal energy dissipation rate, then that system of loads will cause collapse, or incipient collapse, of the structure.”

Lower bound and upper bound solutions are obtained while the conditions are satisfied respectively [11].

- Lower bound: equilibrium, yield and boundary conditions for an arbitrarily assumed bending moment diagram.

- Upper bound: compatibility, incompressibility and boundary condition for an arbitrarily assumed displacement field.

In the PhD thesis of (Buldgen [13]), analytical formulations were developed to assess the crushing force of plane and miter lock gates collided by ships. To do so, an arbitrary velocity field in reality was chosen and calculation of the corresponding deformation energy rate provided an upper bound solution to the crushing force. Upper bound solution concerns about the strain increment and the associated strain rate but not stress equilibrium. The distribution of generalized particle velocity is kinematically compatible within itself and with the external imposing velocities at boundary (A.N. Bramley and F.H. Osman, 1992 [14]). This method leads to an overestimated solution which is proofed in reference [12] and the accuracy of the approximate solution depends on the right choice of the velocity field. The optimal solution is derived by minimizing the total energy rate with the unknown variables in the velocity field inside the deforming region. (A. Alfozan and J. S. Gunasekera [15]).

2.4 Rigid Plastic Material

(T.Wierzbicki and M.S. Suh 1988, [16]) mentioned that “a simple calculation performed in (M.S. Suh 1987 [17]) shows that the strains in the dented region are one or two orders of magnitude higher than the maximum elastic strain which the metal tube can tolerate”. By the research here above, idealizing by using the rigid-plastic material which neglects the elastic part of deformation in the dented region is acceptable. The sketch of stress-strain curve for rigid-plastic material in upper-bound method is shown in Fig.2-5, where the flow stress $\sigma_0$ is
taken conservatively equal to the yield stress $\sigma_y$, although some of researchers (Zhang [18], Lützen [19] et al.) indicated that doing so is too conservative and suggested taking $\frac{\sigma_y + \sigma_u}{2}$.

Nevertheless, as the upper bound method already leads to an overestimate of the crushing energy, it is more suitable to take the flow stress $\sigma_0 = \sigma_y$ in the facing problem.

![Figure 2-5 Stress-Strain curve of rigid plastic material [13]](image)

### 2.5 Finite Element Tool LS-DYNA

LS-DYNA combines the LS-DYNA explicit finite element program with the powerful pre- and post-processing capacities of the LS-PRE-POST. The explicit method of solution used by LS-DYNA provides solution for short time, large deformation dynamics, quasi-static problems with large deformations and multiple nonlinearities, and complex contact/impact problems (ANSYS, Inc. 2009 [20]). LS-DYNA is well-known for the simulation in automotive crash and explosion, and it becomes popular for non-automotive crashworthiness applications nowadays.

#### 2.5.1 Shell Element

In order to validate the results obtained by applying the super-element method, LS-DYNA is used for all the simulations in this thesis which are in the case of large deformation and nonlinear material behavior.

In all the simulations, the reduced integrated Belytschko-Lin-Tsay shell element is used, which is very fast and recommended for most applications (this is the default shell element in LS-DYNA). By the theoretical background of LS-DYNA (LSTC, 2014 [21]), this type of shell element is based on a combined co-rotational and velocity-strain formulation. The efficiency of the element is obtained from the mathematical simplifications that result from these two kinematical assumptions: 1. The co-rotational portion of the formulation avoids the complexities of nonlinear mechanics by embedding a coordinate system in the element. 2. The choice of velocity strain, or rate of deformation, in the formulation facilitates the constitutive evaluation, since the conjugate stress is the more familiar Cauchy stress.
2.5.2 Time Step Calculation

In order to integrate in time the dynamic general equations, a direct explicit integration is used in LS-DYNA. Knowing the solution at time $n$, displacement, strain and stress fields are calculated at time $n+1$ with help of a centered difference step by step integration scheme. For the stability of the explicit method, it is necessary to respect the Courant-Friedrich-Levy condition which specifies the maximum time step for convergence, the integration time step for shell element is given by the formula below:

$$\Delta t_{\text{min}} = \frac{l_c}{c}$$  \hspace{1cm} (2-4)

where, $l_c$ is the characteristic length

$c$ is the speed of sound, $c = \sqrt{\frac{E}{\rho(1-v^2)}}$

For the plate/shell elements in LS-DYNA, $l_c$ is defined as Fig.2-6 below. In addition, some warped elements are used when meshing the model. Hence, for the warped element, the $l_c$ is defined as eq.(2-5) and Fig.2-7.

$$l_c = \frac{(1 + \beta)A}{l_{\text{max}}} \quad \text{or} \quad l_c = \frac{(1 + \beta)A}{d_{\text{max}}}$$  \hspace{1cm} (2-5)

Where, $\beta=0$ for quadrilateral element

$\beta=1$ for quadrilateral element
3 ANALYTICAL SOLUTION

3.1 Deformation Mechanism and Displacement field

3.1.1 Deformation Mechanism

By observation, the following assumption is made while implementing punching in COLEOL program, the punching phenomena is taken into account only if one or several junctions of braces and legs are collided by the stem or bulb of the striking ship. In this study, the braces are assumed to transmit rigidly the global displacement of the junction to the corresponding rear leg. In addition, for simplicity, the indentation transmitted from the colliding leg is assumed to occur only in the x direction of each rear leg coordinate. Therefore, the global indentation of punched leg will be in the same magnitude as the impacted leg projecting at the x axis of rear leg coordinate, the sketch is shown in Fig.3-1. This assumption has been checked by post-processing junction displacements obtained from jacket collision finite element simulations and seems to be acceptable.

![Figure 3-1 Deformation mechanism of rear leg punching](image)

The deformed sections of leg ring and generator are depicted respectively in Fig. 3-2(a) & (b), which are the observation of a simple cylinder punching from LS-LYNA simulations. The deformation mechanism and displacement field recommended by (Buldgen 2014 [6]) is introduced. Nevertheless, it is obvious that the deformation mechanism is not entirely the same as [6]. As the impacted ring (section) is concerned, the deformation mechanism is assumed remaining the same as [6] which is shown in Fig.3-2(c). The deformation mechanism of one
generator of the cylinder is modified as shown in Fig.3-2(d) and the formulas for calculating the energy dissipated by the deformation will be developed based on these shapes.

In reality, the interface between a brace and a punched leg is oval, but for simplifying the problem and being conservative, the region is taken as the red rectangular depicted in Fig.3-2 (e) and the displacement is assumed to be the same for whole interface. ($\alpha$ is the angle between the brace and the leg)

Figure 3-2 Deformation mechanism
The deformation mechanism can be described by several equations. The angle $\psi$ depicted in Fig.3-2(c) defines the position of two plastic hinges which develop through the deforming process of the cylinder cross section and is related to three variables $R_1, R_2$ and $\delta$.

First, during the punching process, the indentation $\delta$ can be expressed by eq.(3-1) which is suggested by (T. Wierzbicki and M.S. Suh, 1988 [16]).

$$\delta = 2R - [R_1(1 + \cos \psi) + R_2(1 - \cos \psi)] \quad (3-1)$$

Second, since the ring is assumed to be inextensible, the circumference remains constant during the crushing process. As the deformation of the ring is symmetric with respect to x axis, following the mathematical expression given by Wierzbicki in [16], the half circumference can be expressed as below.

$$R_1(\pi - \psi) + R_2\psi + \overline{AB} = \pi R \quad (3-2)$$

Using eq. (3-1) and (3-2), $R_1$ and $R_2$ can be derived as functions of $R$ and $\psi$.

$$R_1 = R + \frac{\delta(\psi - \sin \psi)}{\pi(1 - \cos \psi) - 2(\psi - \sin \psi)} \quad (3-3)$$

$$R_2 = R - \frac{\delta(\pi - \psi + \sin \psi)}{\pi(1 - \cos \psi) - 2(\psi - \sin \psi)} \quad (3-4)$$

$$\overline{AB} = (R_1 - R_2)\sin \psi \quad (3-5)$$

$R$ is the radius of the intact ring and $\psi$ is the angle which defines the position of the plastic hinges as aforementioned. Currently, the angle $\psi$ is unknown. Therefore for determining $\psi$, (T. Wierzbicki and M.S. Suh, 1988 [16]) suggested the power dependence given by eq.(3-6). (The plastic hinge in this reference is defined by $\phi$, which is different from $\psi$ but denotes the same position of the plastic hinge. The relation between $\phi$ and $\psi$ is $\psi = \pi - \phi, \psi_0 = \pi - \phi_0$).

$$\frac{R_2}{R} = \left(\frac{\phi}{\phi_0}\right)^n \quad (3-6)$$

It is worth to discuss about $\phi_0$ here, which is the initial value of $\phi$. Wierzbicki and Suh generalized the relation between load-displacement curve and different values of $\phi_0$. As shown in Fig.3-3, which gives the non-dimensional crushing force versus the dented depth, different values of $\phi_0$ lead to different load-displacement curves.
Later on, (M. Zeinoddini et al., 1999 [22]) proposed a linear relationship between the change in the value of \( \delta \) and \( \phi \). While \( \delta \) changes from 0 to \( 2R \), \( \phi \) changes from \( \phi_0 \) to 0, as shown by eq. (3-7). Zeinoddini generalized the relation between load-displacement curves and different values of \( \phi_0 \) as well, the curves are shown in Fig. 3-4.

\[
\frac{\delta}{2R} = 1 - \frac{\phi}{\phi_0}
\]  

(3-7)

To be consistent, the plastic hinge will be denoted by \( \psi \) hereafter and eq. (3-7) can be written as eq. (3-8), which is also used by (Buldgen et al., 2014 [6]).

\[
\psi = \psi_0 + (\pi - \psi_0) \frac{\delta}{2R}
\]  

(3-8)
It implies that the ring is completely crushed at $\psi=\pi$. In the case of jacket leg dented at one point, the energy dissipation is proved to be well-predicted by taking $\psi_0 = \frac{3}{4}\pi$.

Nevertheless, in the case of jacket leg punching, the behavior should be considered differently. This contention can be corroborated by the comparison between jacket leg impact and the study of the strength of tubular joint by (M.M.K. Lee and A. Llewelyn-Parry, 1999 [23]), as shown in Fig.3-5 and Fig.3-6 respectively. The LS-DYNA simulations for the tubes in jacket-like dimension sustaining punching load also reveal the difference (Fig.3-7).
As can be seen in Fig.3-6 and 3-7, the load reaches a local peak at the very beginning, and the peak value varies with the ratio $\beta$ between brace and leg diameters. It implies that the value of $\psi_0$ which leads to a higher resistance at the beginning of load-displacement curve should be selected. Therefore, $\psi_0 = \frac{1}{3} \pi$ is suggested in this study for punching problem. In addition, affecting by the brace, it is observed in the finite element simulations that the ring is completely crushed at $\psi = \frac{1}{2} \pi$. To sum up, the following formula for $\psi$ is suggested:

$$\psi = \psi_0 + \left(\frac{1}{2} \pi - \psi_0\right) \frac{\delta}{2R} \tag{3-9}$$

where, $\psi_0 = \frac{1}{3} \pi$.

3.1.2 Displacement Field

As aforementioned, the deformation field of ring is symmetric with respect to $x$-axis and the ring is inextensible. Therefore, the distances $\overline{GH}$ and $\overline{AH}$ which belong to un-deformed ring $C$ and deformed ring $\overline{AF}$ respectively in Fig.3-8 are the same. The angle $\theta$ in this figure defines the different points on the intact ring. The displacement of each point on the ring for different values of $\delta$ and $\theta$ can be expressed by eq.(3-10).

$$w(\theta, \delta) = \sqrt{(x_H - x_G)^2 + (z_H - z_G)^2} \tag{3-10}$$
The ring can be split up into three regions, $C_1$, $C_2$ and the flat part $\overline{AB}$, the definition of each region are as below:

\[
0 \leq \theta \leq \frac{(R_1 - R_2) \sin \psi}{R} \quad \text{for the flat region}
\]

\[
\frac{(R_1 - R_2) \sin \psi}{R} \leq \theta \leq \frac{(R_1 - R_2) \sin \psi}{R} + \frac{R_2 \psi}{R} \quad \text{for } C_2
\]

\[
\frac{(R_1 - R_2) \sin \psi}{R} + \frac{R_2 \psi}{R} \leq \theta \leq \pi \quad \text{for } C_1
\]

The displacement in each region is derived by following the approach proposed by (T. Wierzbicki and M.S. Suh, 1988. [16]), and proposed formulas are detailed in appendix A. In addition, the velocity field $\dot{w}(\theta, \delta)$ is derived by taking time derivatives.

On the other hand, the deformation increases gradually with the indentation. Introducing the principle of reference [16], the displacement and velocity fields of generator $W(\theta, \delta, y)$, $\dot{W}(\theta, \delta, y)$ are defined in eq.(3-12) and (3-13). The displacement and velocity fields are constant in the region $-\alpha \leq y \leq \alpha$ and are assumed to be linear along the tube in the regions $-\xi_2 \leq y \leq -\alpha$ and $\alpha \leq y \leq \xi_1$. The rest of regions remain un-deformed.
Displacement field of generator:
\[
\begin{align*}
\dot{W}(\theta, \delta, y) &= 0 & \text{for } y \in [\xi_1 ; L_1] \\
W(\theta, \delta, y) &= w(\theta, \delta) \left( 1 - \frac{y-a}{\xi_1-a} \right) & \text{for } y \in [a ; \xi_1] \\
W(\theta, \delta, y) &= w(\theta, \delta) & \text{for } y \in [-a ; a] \\
W(\theta, \delta, y) &= w(\theta, \delta) \left( 1 - \frac{y+a}{\xi_2-a} \right) & \text{for } y \in [-\xi_2 ; -a] \\
\dot{W}(\theta, \delta, y) &= 0 & \text{for } y \in [-L_2 ; -\xi_2]
\end{align*}
\] (3-12)

where, \( a \leq \xi_1 \leq L_1, \ a \leq \xi_2 \leq L_2 \) and \( L_1, L_2 \) are known from the position of the brace.

Velocity field of generator:
\[
\begin{align*}
\dot{W}(\theta, \delta, y) &= 0 & \text{for } y \in [\xi_1 ; L_1] \\
\dot{W}(\theta, \delta, y) &= \dot{w}(\theta, \delta) \left( 1 - \frac{y-a}{\xi_1-a} \right) & \text{for } y \in [a ; \xi_1] \\
\dot{W}(\theta, \delta, y) &= \dot{w}(\theta, \delta) & \text{for } y \in [-a ; a] \\
\dot{W}(\theta, \delta, y) &= \dot{w}(\theta, \delta) \left( 1 - \frac{y+a}{\xi_2-a} \right) & \text{for } y \in [-\xi_2 ; -a] \\
\dot{W}(\theta, \delta, y) &= 0 & \text{for } y \in [-L_2 ; -\xi_2]
\end{align*}
\] (3-13)
3.2 Derivation of the Crushing Resistance

3.2.1 Local Crushing Resistance

The denting process depends on the stretching, flexure and shear of the punched cylinder. For simplicity, as suggested by (T.Wierzbicki and M.S. Suh, 1988 [16]), the shear effect is neglected and the cylinder is considered as the composition of a series of unconnected rings and a bundle of generators, so rings and generators can slide on each other without the resistance of shear effect. Because of the loose connection of rings and generators, the total energy rate can be expressed as eq.(3-14) which is the sum of rings and generators internal energy rates.

\[ \dot{E} = \dot{E}_r + \dot{E}_g \]  

As mentioned before, the internal energy rates can be calculated by applying the principle of virtual work. Regarding the internal energy, (T.Wierzbicki and M.S. Suh, 1988 [16]) stated that “In the case of rigid-plastic shells, the rate of internal work \( \dot{E}_{\text{int}} \) is given by the sum of contributions due to continuous deformations and discontinuous velocity fields in the stationary or moving plastic hinge lines”. The internal work rate can be expressed by eq.(3-15):

\[ \dot{E}_{\text{int}} = \int_S (M \kappa + N \dot{\varepsilon}) dS + \sum_i \int_{\Gamma^{(i)}} M_0^{(i)} [\Omega]^{(i)} d\Gamma \]  

where, \( M \) is the bending moment and \( \kappa \) is the curvature rate

\( N \) is the membrane force and \( \dot{\varepsilon} \) is the extension rate

\( M_0 \) is the fully plastic bending moment

\( \Omega \) is the relative rotation rate on both side of the hinge

\( dS \) and \( d\Gamma \) are the current deformed surface element and the hinge line respectively

- Energy Rate of Ring

As aforementioned, the rings are inextensible, which means \( \dot{\varepsilon} = 0 \). Proposed by (T.Wierzbicki and M.S. Suh, 1988 [16]), the energy absorbed by a ring is due to the circumferential bending in deformation field and the moving plastic hinge lines in discontinuous velocity field. Also, for simplifying the problem, the circumferential bending moment is assumed to be independent of axial force \( N \) and is taken to be equal to the fully plastic bending moment \( M_0 \). The equation of energy rate per unit width of the ring becomes:
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\[ \hat{E}_b = \int_{L} (M_0 \kappa) ds + \sum_{i} M_0^{(i)} [\Omega]^{(i)} \]  

(3-16)

$L$ is length of circumference and $i$ is the number of plastic hinge. The curvature and the rate of curvature are denoted as below:

\[ \kappa_1 = \frac{1}{R_1}, \quad \dot{\kappa}_1 = \frac{\dot{R}_1}{R_1^2} \]

\[ \kappa_2 = \frac{1}{R_2}, \quad \dot{\kappa}_2 = \frac{\dot{R}_2}{R_2^2} \]

(3-17)

As the deformation of the ring is considered to be symmetric, only half circumference and two plastic hinges will be discussed hereafter. For the first term of eq.(3-16), the integration of the curvature rate along $C_1$ and $C_2$ can be expressed as eq.(3-18) below:

\[ \int_{C_1}^{B} \kappa_1 dS = R_1 (\pi - \psi) \frac{\dot{R}_1}{R_1^2} = \frac{\pi - \psi}{R_1} \frac{\partial R_1}{\partial \delta} \dot{\delta} \]

(3-18)

\[ \int_{B}^{C} \kappa_2 dS = R_2 \psi \frac{\dot{R}_2}{R_2^2} = \frac{\psi}{R_2} \frac{\partial R_2}{\partial \delta} \dot{\delta} \]

(3-19)

Regarding the second term of eq.(3-16), the summation of the rate of rotation can be denoted as $\Omega_C + \Omega_B$ and the rate of rotation is defined as the change of the curvature $[\kappa]$ on both side of plastic hinges travelling with velocity $V$.

\[ [\Omega] = V[\kappa] \]

(3-20)

Hence, the rotation rate of the two plastic hinges $B$ and $C$ can be defined as below:

\[ \Omega_C = V_C \left( \frac{1}{R_2} - \frac{1}{R_1} \right), \quad \Omega_B = V_B \left( \frac{1}{R_2} \right) \]

(3-21)

where the $V_B$ and $V_C$ are the tangential velocities in the current deformed configuration, which is defined as $\frac{ds}{dt}$. So $V_B$ and $V_C$ can be defined as below

\[ V_B = \frac{C_1 + C_2}{dt} = \frac{R_1 (\pi - \psi) + R_2 \psi}{dt} = (R_1 - R_2) \psi - (\pi - \psi) \dot{R}_1 - \psi \dot{R}_2 \]

\[ = \left[ (R_1 - R_2) \frac{\partial \psi}{\partial \delta} - (\pi - \psi) \frac{\partial R_1}{\partial \delta} - \psi \frac{\partial R_2}{\partial \delta} \right] \dot{\delta} \]

(3-22)

\[ V_C = \frac{C_1}{dt} = \frac{R_1 (\pi - \psi)}{dt} = R_1 \psi - (\pi - \psi) \dot{R}_1 = \left[ R_1 \frac{\partial \psi}{\partial \delta} - (\pi - \psi) \frac{\partial R_1}{\partial \delta} \right] \dot{\delta} \]

(3-23)

Substituting eq.(3-18), (3-19), and (3-20) into eq.(3-16), the dissipated bending energy of the full circumference can be expressed as below:
\[ \hat{E}_b = 2M_0 \left[ \frac{V_B}{R_2} + \left( \frac{1}{R_2} - \frac{1}{R_1} \right) V_C + \int_C^D \kappa_1 dl + \int_C^C \kappa_2 dl \right] \] (3-24)

where, the fully plastic bending moment \( M_0 = \sigma_0 t^2_p / 4 \).

In order to obtain the energy rate of all the rings, each section of the deforming part is integrated. As aforementioned, for the purpose of conservation, the flat region of the indentation on the ring will be taken as an inscribed rectangular (as depicted in Fig.3-2(e)). The equation of the crushing energy rate of ring is then obtained by:

\[
\hat{E}_r = \int_{-\xi_2}^{a} \hat{E}_b \left( 1 + \frac{y + a}{\xi_2 - a} \right) dy + \int_{-a}^{a} \hat{E}_b dy + \int_{a}^{\xi_1} \hat{E}_b \left( 1 - \frac{y - a}{\xi_1 - a} \right) dy
\]

\[= \hat{E}_b \left\{ \left[ y + \frac{1}{2}y^2 + a \right]_{-\xi_2}^{a} + \left[ y - \frac{1}{2}y^2 - a \right]_{a}^{\xi_1} \right\} \] (3-25)

\[= \hat{E}_b \left( a + \frac{\xi_1 + \xi_2}{2} \right) \]

**Energy Rate of Generator**

The length and the curvature are changing while the indentation increases, so both membrane and bending effects are involved which is a complex issue. The change in the circumferential curvature is appreciable in comparison with the change in the longitudinal curvature of generator, so the terms with \( \kappa \) are supposed to be small enough to be neglected. Therefore equation (3-15) becomes:

\[ \hat{E}_{gen} = \int_S (N\hat{e}) dS = \int_C^{\xi_1} \hat{E}_m dy dl \] (3-26)

The extension rate \( \hat{e}_m \) is defined as eq.(3-27)

\[ \hat{e}_m(\theta, \delta, y) = \frac{\partial W}{\partial y} \frac{\partial \hat{W}}{\partial y} \] (3-27)

where,

\[ \frac{\partial W}{\partial y} = \frac{\partial W}{\partial y} = w(\theta, \delta) \left( \frac{1}{\xi_1 - a} \right) \quad \text{for} \quad y \in [a ; \xi_1] \] (3-28)
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First, dealing with the energy rate of each generator and introducing into the approach described in (Buldgen et al. 2014 [6]), the energy rate of a single generator at a specific position $\theta$ for cylinder punching problem is derived as:

$$\dot{E}_m(\theta, \delta) = n_0 \int_{-\xi_2}^{\xi_1} \dot{e}_m(\theta, \delta, y) dy$$

$$= n_0 w(\theta, \delta)\dot{w}(\theta, \delta) \left\{ \left( -\frac{1}{\xi_1 - a} \right) \left[ -\frac{y}{\xi_1 - a} \right]_{a}^{\xi_1} \right\} + \left[ w(\theta, \delta) \cdot 0 \right]_{-\xi_2}^{\xi_1}$$

$$+ \left( \frac{1}{\xi_2 - a} \right) \left[ \left( \frac{y}{\xi_2 - a} \right) \right]_{-\xi_2}^{\xi_1}$$

$$= n_0 w(\theta, \delta) \frac{\partial w}{\partial \delta} \delta \left[ \frac{1}{\xi_1 - a} + \frac{1}{\xi_2 - a} \right]$$

(3-29)

where, $n_0 = \sigma_0 \epsilon_p$ is the fully plastic membrane force of the generator per unit width.

Next, the crushing energy rate of all generators is derived by integrating each generator:

$$\dot{E}_g = \int_c \dot{E}_m dl = 2Rn_0 \delta \left[ \frac{1}{\xi_1 - a} + \frac{1}{\xi_2 - a} \right] \int_0^\pi w(\theta, \delta) \frac{\partial w}{\partial \delta} d\theta$$

$$= \left[ \frac{1}{\xi_1 - a} + \frac{1}{\xi_2 - a} \right] \dot{E}_m$$

(3-30)

Total energy rate

As aforementioned, the total energy rate can be easily expressed by the combination of rings and generators. Therefore, the total energy is the combination of eq.(3-25) and (3-30):

$$\dot{E} = \dot{E}_r + \dot{E}_g = \dot{E}_b \left( a + \frac{\xi_1 + \xi_2}{2} \right) + \dot{E}_m' \left( \frac{1}{\xi_1 - a} + \frac{1}{\xi_2 - a} \right)$$

(3-31)

Evaluation of $\xi_1$ and $\xi_2$

By observation, the indentation of the plastically deforming zone gradually decreases within certain length from the dent region (interface of leg and brace where the maximum indentation
occurs. The extent of the local damaged zone is assumed to be finite and to keep increasing during the loading process, which is shown in Fig.3-2 (d). As can be seen in this figure, the deforming zone is characterized by $\xi_1$ and $\xi_2$, which are two variables depending on the indentation $\delta$ in each time step.

As aforementioned, respecting the principle of virtual work, the energy dissipation is expressed by eq.(3-32).

$$P_l(\delta) \cdot \delta = E_b \left( a + \frac{\xi_1 + \xi_2}{2} \right) + E_m' \left( \frac{1}{\xi_1 - a} + \frac{1}{\xi_2 - a} \right)$$  \hspace{1cm} (3-32)

Therefore, the local resistance can be found as eq.(3-33):

$$P_l(\delta) = \left\{ E_b \left( a + \frac{\xi_1 + \xi_2}{2} \right) + E_m' \left( \frac{1}{\xi_1 - a} + \frac{1}{\xi_2 - a} \right) \right\} / \delta$$  \hspace{1cm} (3-33)

where $\xi_1$ and $\xi_2$ are assessed by minimizing the crushing force given by the equation above. This minimum can be found by the 1st differentiation with respect to $\xi_1$ and $\xi_2$.

$$\frac{\partial P_l}{\partial \xi_1} = 0 \Rightarrow \xi_1 = \min \left( a + \sqrt{\frac{2E_m'}{E_b}} ; L_1 \right)$$

$$\frac{\partial P_l}{\partial \xi_2} = 0 \Rightarrow \xi_2 = \min \left( a + \sqrt{\frac{2E_m'}{E_b}} ; L_2 \right)$$  \hspace{1cm} (3-34)

### 3.2.2 Global crushing resistance

By observing the finite element simulation of a simple tube punching, one can tell that the global crushing phenomena follow the deformation pattern considered by (Buldgen et al., 2014 [6]) which is described as follows. At the beginning of the crushing process, the crushing resistance dominates by local indentation and later on the global plastic bending process gradually happens. Nevertheless, for solving this problem analytically, a threshold is set in order to trigger the global mode. The threshold is defined as the minimal load in [6] that can activate the global mode and makes the entire cylinder bent. The crushing resistance during the punching process is depicted by the green curve in Fig.3-9, and the crushing force of leg/brace impact process is depicted by the red curve. ($\delta$ is total indentation and $\delta_t$ is transition point of indentation).

At the beginning ($\delta < \delta_t$), only local resistance is considered. While the local resistance achieves the threshold ($\delta = \delta_t$), which can activate the global mode, the mode is switched and only global crushing resistance is considered for the subsequent process ($\delta > \delta_t$).

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The global mode can be considered as typical plastic bending of beam, and a three hinges plastic mechanism is considered in the case of one point denting. Nevertheless, for the case of punching, instead of three plastic hinges, a four plastic hinges mechanism is considered, as shown in Fig. 3-10.

The global deformation can be expressed by \( \delta - \delta_i \) which is depicted in Fig.3-10. Following a classical plastic analysis, the energy dissipated in the plastic hinges and the work done by the displacement of the load are described by eq.(3-35) and eq.(3-36) respectively:
\[ W_{\text{hinge}} = M_0 \theta + M_0 \varphi + \xi \xi M_0 \theta + \xi \xi M_0 \varphi \] (3-35)

\[ W_p = \frac{P(\delta \xi)}{2} L_1' \theta + \frac{P(\delta \xi)}{2} L_2' \varphi \] (3-36)

where, \( L_1' = L_1 - a \), \( L_2' = L_2 - a \).

\( M_0 \) is the plastic bending moment which can be expressed as \( M_0 = 4R^2 \sigma_0 t_p \). However, this expression holds for the intact circular cross section. However, once the ring is dented, fully bending moment \( M_0 \) will not be reached. As can be seen in Fig.3-10 and eq.(3-35), \( M_0 \) is replaced by \( \zeta(\delta) \cdot M_0 \) at the dented point, which takes into account the deformed cross section by using a reduction factor \( \zeta(\delta) \). The formula of reduction factor derived by (Buldgen et al., 2014 [6]) is introduced which follows the suggestion of (de Oliveira et al., 1982 [24]). Under the assumption of the cross section as shown in Fig. 3-11, the arc \( C_1 \) is extended and the arc \( C_2 \) is neglected, the formula is obtained as shown in the following equation, eq.(3-37).

\[ \xi(\delta) = \frac{1}{2} \left( \left( \frac{\delta}{2R} \right)^2 - 1 \right) \left( \frac{\delta}{2R} - 2 \right) \] (3-37)

![Figure 3-11 Damaged cross section [6]](image)

The energy dissipated in the plastic hinges and the work done by external load must be equal when the plastic hinges appear, which means that eq.(3-35) and (3-36) should be equal, as written in the following eq.(3-38).

\[ M_0 \theta + M_0 \varphi + \xi \xi M_0 \theta + \xi \xi M_0 \varphi = \frac{P(\delta \xi)}{2} L_1' \theta + \frac{P(\delta \xi)}{2} L_2' \varphi \] (3-38)
and the global indentation can be expressed by eq.(3-38).

\[ L'_1 \delta = \delta - \delta_t = L'_2 \varphi \]  

(3-39)

Accordingly, the load which can activate the global mode is obtained by substituting eq.(3-39) into eq.(3-38) and eliminating \( \theta \). This load is considered as starting point of the global load, as shown in eq.(3-40).

\[ P_g(\delta_t) = \frac{L'_1 + L'_2}{L'_1 L'_2} \left( 1 + \xi(\delta_t) \right) M_0 \]  

(3-40)

As aforementioned, the global mode is triggered while the local load reaches this threshold. Therefore, at the transition point, the local load is equal to the global load as following equation.

\[ P_l(\delta_t) = P_g(\delta_t) = \frac{L'_1 + L'_2}{L'_1 L'_2} \left( 1 + \xi(\delta_t) \right) M_0 \]  

(3-41)

Nevertheless, taking this threshold value does not lead to a correct result in the case of punching. As mentioned before, the peak of the load-displacement curve is influenced by the \( \beta \) ratio, but the influence of \( \beta \) by using eq.(3-41) is unapparent, since the magnitude is governed by \( L'_1 \) and \( L'_2 \) which are just slightly influenced by \( \beta \). As shown in Fig. 3-12, the threshold varies along with different \( \beta \) ratios, but for the sake of long span, the influence of \( \beta \) to \( \frac{L'_1 + L'_2}{L'_1 L'_2} \) is relatively small. In comparison to the reality, the load-displacement curve can not be described correctly by using eq.(3-41). As can be seen in Fig. 3-12, the curve obtained by applying eq.(3-41) may under or over estimates the resistance, and the absorbed energy is affected directly by this curve.

![Comparison between analytical formulation and reality](image-url)
To conclude, the formula should be modified in order to account correctly for the influence of the diameter ratio $\beta$. Referring to several references such as (Department of Energy, 1990 [25], Moffat et al., 1996[26], and Standards Norway, 2004 [27]), it can be generalized that the peak is governed by $\beta$ (first order) and by the geometry factor $Q_\beta$ (function of $\beta$). The relation is described by a regression curve obtained from more than 50 T/Y joint experiments, which is done by the Department of Energy, as can be seen in Fig. 3-13(a) & (b). In Fig.3-13(a), the relationship is approximately linear for $\beta \leq 0.6$, but it increases faster while $\beta$ is over 0.6. Therefore, the geometry factor $Q_\beta$ is introduced and the linear relationship is obtained by taking $\sqrt{Q_\beta}$ in order to account for the nonlinear relationship in the non-dimensional strength.

![Diagram](image)

(a)

![Diagram](image)

(b)

Figure 3-13 T/Y joints – comparison non-dimensional strength against $\beta$ ratio [25]
In order to account for the influence of brace diameter, the empirical parameter $\beta$ and $Q_\beta$ are introduced into eq.(3-40). A new threshold is proposed as shown in eq.(3-42), which is considered as a critical force that marks load curve behavior off. It means that before this critical value, the punched leg behavior is governed by local crushing mechanism even though the global mode is activated, and the punched leg behavior is governed by global crushing mechanism until the critical force is reached.

$$P_t(\delta_t) = P_c(\delta_t) = \frac{L_1' + L_2'}{L_1' L_2'} \left( 1 + \beta \times \sqrt{Q_\beta \times \xi(\delta_t)} \right) M_0 \tag{3-42}$$

where $Q_\beta = 1$ for $\beta \leq 0.6$

$$Q_\beta = \frac{0.3}{\beta(1-0.833\beta)} \quad \text{for} \quad \beta > 0.6$$

The reason of taking $\beta$ and $Q_\beta$ only in the second term is that the central cross-section is the only part affected by the diameter of the brace. The plastic bending moment at the extremities should remain the same since it is irrelevant to the brace diameter.

When the punching process gets into the global mode ($\delta > \delta_t$), the local indentation is stopped and the cylinder is bent globally. The global crushing resistance is given by both bending moment and membrane force. These two factors are dependent and the relationship between bending moment and membrane force is proposed by (De Oliveira et al., 1982 [24]), as shown in eq.(3-43) and (3-44) which are valid for the extremities and the dented cross-section respectively. (eq.(3-34) is modified by multiplying $\beta$ and $\sqrt{Q_\beta}$)

$$M = M_0 \left( 1 - \frac{N^2}{N_0^2} \right) \tag{3-43}$$

$$M = \beta \times \sqrt{Q_\beta \times \xi_t} M_0 \left( 1 - \frac{N^2}{N_0^2} \right) \tag{3-44}$$

where $N_0 = 2\pi R\sigma_0 t_p$, is the plastic tensile capacity of the cylinder.

Following the suggestion of (Buldgen et al., 2014 [6]), applying the relation above mentioned and the beam theory, the global resistance can be expressed as eq.(3-45).

$$P_g(\delta) = \frac{L_1' + L_2'}{L_1' L_2'} \left[ \left( 1 + \beta \times \sqrt{Q_\beta \times \xi_t} \right) M_0 \left( 1 - \frac{N(\delta)^2}{N_0^2} \right) + N(\delta)(\delta - \delta_t) \right] \tag{3-45}$$

where $N(\delta) = \min \left( \frac{N_0^2(\delta-\delta_t)}{2(1+\beta\sqrt{Q_\beta\times\xi_t}M_0)}; N_0 \right)$
4 NUMERICAL VALIDATION-SIMPLE CYLINDER

4.1 Simple Cylinder Punching Simulation Introduction

There are many different tubular joints in different industries, and in general, T/Y joint and K joint are often used for manufacturing jacket foundations. Referring to the NORSOK Standard-Design of steel structures which is suggested by (Standards Norway, 2004 [27]), the validity range for application of the simple tubular joints is as following:

\[
0.2 \leq \beta \leq 1.0 \\
10 \leq \gamma \leq 50 \\
30^\circ \leq \theta \leq 90^\circ \\
\frac{g}{D} \geq -0.6
\]  

(4-1)

Some simple cylinder punching simulations are carried out by referring to a real jacket leg and brace dimensions provided by STX Europe in the framework of CHARGEOL project and listed in table 4-1. This simple cylinder, which represents the punched leg, is considered as a benchmark. The different influential factors are studied by varying the dimensions within the range mentioned above and based on the range of the dimensions suggested by STX.

![Diagram of geometrical parameters of K joint](image)

Figure 4-1 Definition of geometrical parameters of K joint

<table>
<thead>
<tr>
<th>DL</th>
<th>1.25</th>
<th>m</th>
<th>t_P</th>
<th>40</th>
<th>mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>DB</td>
<td>0.6</td>
<td>m</td>
<td>( \sigma_0 )</td>
<td>240</td>
<td>N/mm^2</td>
</tr>
<tr>
<td>L</td>
<td>17</td>
<td>m</td>
<td>a</td>
<td>45</td>
<td>degrees</td>
</tr>
<tr>
<td>L_1</td>
<td>10.5</td>
<td>m</td>
<td>B.C.</td>
<td>clamped</td>
<td>--</td>
</tr>
<tr>
<td>L_2</td>
<td>6.5</td>
<td>m</td>
<td>( \beta )</td>
<td>0.48</td>
<td>--</td>
</tr>
</tbody>
</table>

Table 4-1 Dimension of the jacket-like cylinder
4.1.1 Model Description

The model presented in this chapter is simply one cylinder joining with another one or two cylinders. Several models with different dimensions are built to study different influential factors, and the general information model is described as following.

- Mild steel material

For simulating the plastic deformation of the tube which sustains the brace axial load using LS-DYNA, the MAT_PIECEWISE_LINEAR_PLASTICITY behavior law is used. This law allows to describe an elasto-plastic behavior with an arbitrary Cauchy stress versus natural logarithmic plastic strain curve and plastic strain rate dependency. It is worth to mention that the strain values defined in the table may be given as the natural logarithm of the strain, since the tables are internally discretized to equally space the points, natural logarithms are necessary (LSTC, 2013 [28]).

The stress-strain curve of mild steel is defined as the curve depicted in Fig.4-2. The elastic part in the curve is accounted for specifying the Young’s modulus (σ₀=255 MPa) and the Poison’s ratio (ν=0.3) in the material description card. A piecewise linear curve is used to describe plastic behavior including the stress hardening.

![Stress-Strain Curve](image)

Figure 4-2 Cauchy Stress-natural logarithmic plastic strain curve of mild steel

- Prescribed displacement curve

The punching phenomenon is simulated by imposing a given displacement at the interface between the two cylinders (leg and brace) in current phase. In order to obtain numerical results comparable to analytical ones, the effect of inertia forces is intentionally eliminated by imposing gradually the displacement from 0 to 1m within 12 seconds. The displacement is
defined by using BOUNDARY_PRESCRIBED_MOTION_SET_ID card in LS-DYNA on a set of nodes which are located at the interface of two cylinders. The curve referenced in this card is depicted in Fig.4-3.

![Figure 4-3 Displacement curve used in LS-DYNA](image)

- **Boundary conditions**

The boundary condition at two extremities of the cylinder is assumed to be clamped. The constraint is defined by using BOUNDARY_SPC_SET_BIRTH_DEATH card in LS-DYNA and all the nodes on the extremities are grouped in a set and fixed for 6 degrees of freedom.

- **Converge study of the finite element mesh**

Under-integrated Belytchko-Lin-Tsay shell elements have been used to mesh the STX jacket. The element size of the mesh provided by STX was initially around 100mm. In order to check if the mesh size remains acceptable while focusing on the punching phenomenon, more refined meshes are built for executing explicit finite element impact simulation, and corresponding results are compared. Table 4-2 presents the different mesh sizes as well as discrepancy obtained by comparing more refined mesh results with the 100mm mesh size one and the sketches of the model in different mesh sizes are shown in Fig. 4-4. To illustrate this, the comparison of internal energy and crushing resistance between different mesh sizes are shown in Fig.4-5 and 4-6.

<table>
<thead>
<tr>
<th>Smallest mesh size</th>
<th>Number of element</th>
<th>Discrepancy of energy</th>
<th>Discrepancy of force</th>
<th>Simulation time</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 mm</td>
<td>7040</td>
<td>--</td>
<td>--</td>
<td>67 s</td>
</tr>
<tr>
<td>50 mm</td>
<td>7630</td>
<td>≈3.5%</td>
<td>≈8%</td>
<td>111 s</td>
</tr>
<tr>
<td>25 mm</td>
<td>16320</td>
<td>≈6%</td>
<td>≈12%</td>
<td>298 s</td>
</tr>
</tbody>
</table>
Figure 4-4 Sketch of simple cylinder in different mesh sizes

Figure 4-5 Internal energy obtained by different mesh sizes
As can be seen, the discrepancy between different mesh sizes regarding the internal energy is insignificant (3–6%). As the crushing resistance is concerned, the discrepancy of force is around 8–12%. The 25mm mesh size is then used for simple cylinder punching simulation, since the simulation related to this simple model is not time consuming. However, the 100mm mesh size is used for the jacket impact simulations because of considerably number of elements.
4.2 Sensibility Study to $\beta$ Ratio

4.2.1 Finite element numerical analysis

In this subsection simple cylinder models of different $\beta$ ratios ($\beta=0.32, 0.48, 0.64, 0.8,$ and 1) are built for verifying the sensibility of the new developed super-element to the $\beta$ ratio. The absorbed energy and resistance of different $\beta$ ratios obtained using LS-DYNA are shown in Fig.4-7 and Fig. 4-8 respectively. As can be seen, the internal energy and crushing resistance increase along with increasing $\beta$ ratio.

![Figure 4-7 Variation of internal energy with change in brace diameters](image1)

![Figure 4-8 Variation of crushing resistance with change in brace diameters](image2)
Basically, all load-displacement curves reach their first peak at more or less the same indentation, \( d = 0.1 \text{m} \), then start load shedding and reach a trough. The load starts increasing subsequent to the load shedding as a result of the change in load-carrying mechanism from chord bending to membrane force in the chord, which is a phenomenon mentioned by (Lee and Llewelyn-Parry, 1999).

In addition, it is worth to mention that even though peak values differ from each \( \beta \) ratio, but in the later stage of the crushing process, they all close to a value which is the maximum axial force without bending moment. However, the variations still exist, especially for high \( \beta \) ratio. As can be seen in Fig. 4-9, the crushed ring section of \( \beta = 0.8 \) slightly differs from the other three \( \beta \) ratios and \( \beta = 1 \) is obviously different from the others. For high \( \beta \) value, different crushing mechanism may explain why the load shedding is not significant and the maximum axial force differs from the case in low \( \beta \) value.

![Comparison of crushed ring sections between different \( \beta \) ratios](image)

Figure 4-9 Comparison of crushed ring sections between different \( \beta \) ratios
The global displacement of the chord for different $\beta$ ratios are depicted in Fig. 4-10, which is measured at the backside of the cylinder (point D in Fig.3-8). As can be seen, the higher the $\beta$ value is, the faster the global mode is triggered. For small $\beta$ values, the global mode starts after 4 seconds in the process; inversely, it starts almost at the beginning of the crushing process in large $\beta$ values. It also demonstrates that the load curve behavior is not taken over by the starting of the global mode, because the first peak is reached at $d=0.1m$, $t=2.5s$, and the global mode happens earlier or later in different cases.

![Figure 4-10 Comparison of global displacement between different $\beta$ ratios](image)

The contours of effective plastic strain for different $\beta$ ratios at the same time step $t=9.6s$ are shown in Fig 4-11~4-16.

- $\beta=0.32$

![Figure 4-11 Contour of effective plastic strain, $\beta=0.32$](image)
• $\beta = 0.48$

Figure 4-12 Contour of effective plastic strain, $\beta=0.48$

• $\beta = 0.64$

Figure 4-13 Contour of effective plastic strain, $\beta=0.64$

• $\beta = 0.8$

Figure 4-14 Contour of effective plastic strain, $\beta=0.8$

Master Thesis developed at ICAM, Nantes
As can be seen, plastic strain occurs at both the interface between cylinders and the chord extremities. Moreover, the plastic strain distribution near the interface are similar for $\beta = 0.32$ and 0.48, $\beta = 0.64$ and 0.8, but the strain distribution for $\beta = 1$ is completely different from others. For $\beta = 0.64$ and 0.8, there are some regions where the plastic strain is almost zero, this is due to the wide brace diameter which makes the leg dented non-perpendicularly. For $\beta = 0.8$ and 1, the displacement is shifted by the shell near the interface and then the other parts of the leg is squeezed, as shown in Fig.4-17. Especially for $\beta = 1$, the plastic strain occurs on the backside of chord which is squeezed by the brace.
4.2.2 Comparison analytical and numerical results

The comparison between the results obtained from the developed super-element and LS-DYNA simulation results are exhibited hereafter for each $\beta$ ratio.

- $\beta = 0.32$

Energy

![Internal Energy](image1.png)

Figure 4-18 Comparison of internal energy between two methods, $\beta = 0.32$

Resistance

![Crushing Resistance](image2.png)

Figure 4-19 Comparison of crushing resistance between two methods, $\beta = 0.32$

As shown in Fig. 4-18 above, the absorbed energy by super-element coincides well with LS-DYNA simulation except for $d = 0.2\sim0.4$m where the absorbed energy is overestimated with 35% discrepancy by the super-element. This is due to the overestimate of the first peak and to the fact that the load shedding is neglected in the formula, as shown in Fig.4-19.
• $\beta = 0.48$

Energy

![Internal Energy Comparison](image)

Figure 4-20 Comparison of internal energy between two methods, $\beta = 0.48$

Resistance

![Crushing Resistance Comparison](image)

Figure 4-21 Comparison of crushing resistance between two methods, $\beta = 0.48$

For the jacket-like dimension, $\beta = 0.48$, the absorbed energies obtained from super-element and LS-DYNA are in good agreement, the maximal discrepancy of the absorbed energy is around 8%. The crushing resistance is shown in Fig. 4-21 and the discrepancy of peak value is around 0.7%.
• $\beta = 0.64$

Energy

![Internal Energy](image)

*Figure 4-22 Comparison of internal energy between two methods, $\beta = 0.64$*

Resistance

![Crushing Resistance](image)

*Figure 4-23 Comparison of crushing resistance between two methods, $\beta = 0.64$*

As can be seen in Fig. 4-22, the absorbed energy obtained by super-element is also consistent with the result from LS-DYNA and the maximal discrepancy is around -8.4%. The crushing resistance is shown in Fig.4-23, the discrepancy of the peak value is -16.6% which is significant, but the underestimate is compensated by neglecting the load shedding.
• $\beta=0.8$

Energy

![Figure 4-24 Comparison of internal energy between two methods, $\beta=0.8$](image)

Resistance

![Figure 4-25 Comparison of crushing resistance between two methods, $\beta=0.8$](image)

The maximal deviation of the absorbed energy and resistance between super-element and LS-DYNA in Fig. 4-24 & 4-25 are around -14.7% and -14% respectively. In this case, the load shedding is insignificant and the resistance given by the super-element is underestimated during entire crushing process. This can be attributed to the derivation of the peak value. Therefore, if the prediction of the peak value could be improved, a better result should be obtained.
• $\beta = 1$

Energy

Figure 4-26 Comparison of internal energy between two methods, $\beta = 1$

Resistance

Figure 4-27 Comparison of crushing resistance between two methods, $\beta = 1$

For $\beta = 1$, as can be seen in Fig.4-26, the absorbed energy in the anterior crushing process is predicted correctly by the super-element. Nevertheless, the derivation between two methods starts to increase from $d=0.8$m and the maximal deviation is around -6.8%. The crushing resistance in Fig.4-27 reveals that the first peak is well predicted, but there is no load shedding occurs in this case. In addition, the in the later stage, the force does not increase as fast as expected. The reason of aforementioned phenomena may be the different crushing mechanism as we saw earlier in this section.
To sum up, the super-element predicts quite well the punching of the cylinder for different $\beta$ ratios. Although, discrepancy between super-element and LS-DYNA still exists, but the deviation is not tremendous; hence, these results are acceptable since they are on the safe side. In addition, it is cumbersome to obtain closed-form expressions which can perfectly describe the crushing force evolution but the main target is to approach the absorbed energy well, and regarding the load-displacement curve, the objective is to catch the trend of the curve. It is also worth to mention that the load-displacement curve obtained by super-element starts with a non-zero value because of supposing a rigid-plastic material.
4.3 Sensibility analysis to the angle between brace and chord

4.3.1 Finite element numerical analysis

In this section, models with different brace angles are built in order to study the sensitivity of the result obtained using the super-element to angle between the brace and the chord. The angle $\alpha=90^\circ$ is selected because braces sometimes are used to rely perpendicularly on the legs in the lower part of the jacket. The contour of effective plastic strain is shown in Fig. 4-28, and the angle affects the shape of the intersection which is circular instead of oval.

The internal energies calculated by LS-DYNA for different angles are shown in Fig. 4-29, which reveals that the variation between different angles is not significant, the maximal diversity is around 10%.

![Intersection](image)

**Figure 4-28** Contour of effective plastic strain, $\alpha=90^\circ$

**Figure 4-29** Comparison of LS-DYNA results between different angles
4.3.2 Comparison of analytical and numerical results

For a leg punched by a perpendicular brace, the absorbed energy and crushing forces obtained numerically and analytically are compared in Fig.4-30 and 4-31 respectively.

The maximal discrepancy of absorbed energy is 15.4% at \( d=0.22 \) m and the discrepancy of first peak in load-displacement curve is around 14.4%. Generally, the results obtained by developed super-element and LS-DYNA simulations match well with each other.
4.4 Sensibility study to the span

4.4.1 Finite element numerical analysis

In general, the span between joints differs from points on the jacket foundation. The higher the point is, the smaller the span will be. The contour of effective plastic strain of span $L=23.5m$ is shown Fig.4-32, as can be seen that the fringe level is a bit lower than span $L=17m$ (Fig.4-12) at both intersection and extremities. The comparison of absorbed energy and crushing resistance between two spans which are selected from jacket dimension is shown in the Fig.4-33 and 4-34 respectively.

![Figure 4-32 Contour of effective plastic strain for L=23.5m](image1)

![Figure 4-33 Comparison of internal energy by LS-DYNA results between different spans](image2)
Figure 4-34 Comparison of resistance by LS-DYNA results between different spans

Figure 4-34 shows that the first peaks are more or less the same, but regarding the process subsequent to load shedding, the increasing speed is slower for longer span. Explained by (De Oliveira et al., 1982 [24]), due to the axial flexibility at the supports, the beam’s ends can shrink by a distance and extension occurs at the colliding point.

NORSOK, which is published by (Standards Norway, 2004 [27]), states that "relatively small axial displacement have a significant influence on the development of tensile forces in members undergoing large lateral deformation". The force-displacement curves associated with different axial flexibility for center collision are shown in Fig.4-35. $C$ is the non-dimensional spring stiffness and can be expressed as eq. (4-2).

$$C = \frac{4c_1K_2}{f_yAl}$$  \hspace{1cm} (4-2)

where $K =$ axial stiffness

$c_1 = 2$, for clamped beams

1, for pinned beams

$W_c = \frac{d}{2}$, characteristic deformation for tubular beams

$l =$ member length, $f_y =$ yield strength, $A =$ cross-sectional area
Figure 4-35 Force-displacement curve for tubular beam with axial flexibility for a center collision [27].

As can be seen from the formulas and figure above, a longer span leads to a smaller resistance increasing speed while the span is the only variable, which explains the reason of different absorbed energy and resistance obtained in Fig.4-33 and 4-34. Although, two extremities are set to be fixed, all other elements near extremities still possess flexibility.

4.4.2 Comparison of analytical and numerical results

The comparison of absorbed energy and resistance between super-element and LS-DYNA for a longer span, \(L=23.5\)m, is shown in Fig. 4-36 and 4-37.

![Comparison of internal energy between two methods, L=23.5m](imageURL)
The absorbed energy obtained by super-element coincides well with result obtained by LS-DYNA except the overestimate at the beginning of the crushing process, which results from neglect of the elastic part. Although, the discrepancy of the first peak value between two methods is around 25.5%, the subsequent process is predicted correctly, which finally leads to a good prediction of the absorbed energy.

### 4.5 Two Braces Punching Analysis

By observation, three different scenarios are usually observed in jacket collision, as will be discussed in next chapter. One of these scenarios is that the rear leg is punched by two braces simultaneously, while two junctions of leg and braces are collided by the bow and the bulb of the striking ship. Therefore, it is necessary to study the case of two braces punching in the simple cylinder as well. The deformation mechanism is assumed to be as the sketch shown in Fig. 4-38. Generally, the formula used in this case is largely identical but with minor differences, instead of $a$, $2a + 0.7 \times gap$ is used in the equation (3-33) and (3-34) and the corresponding analytical expressions are shown in eq. (4-3) and (4-4).

\[
E' = E_r + E_g \\
= E_b \left( 2a + 0.7 \times gap + \frac{\xi_1 + \xi_2}{2} \right) + \dot{E}_m \left( \frac{1}{\xi_1 - (2a + 0.7 \times gap)} + \frac{1}{\xi_2 - (2a + 0.7 \times gap)} \right) \tag{4-3}
\]

\[
\xi_1 = \min(2a + 0.7 \times gap + \sqrt{\frac{2E'_m}{E_b}; L1}) \tag{4-4}
\]
\[ \xi_2 = \min(2\alpha + 0.7 \times \text{gap} + \frac{2E'_m}{E_b}; L2) \]

Different cases for one cylinder punched by two braces are studied by varying some parameters such as the gap between two braces and span.

4.5.1 Two brace without gap and span=17m

First, the case of two braces without gap is studied and the dimension of the cylinders are selected as before, \( D_L = 1.25 \text{m}, \beta = 0.48 \) and \( \alpha = 45^\circ \). The contour of effective plastic strain is shown in Fig. 4-39, for avoiding the improper mesh, a small gap of one element wide (≈25mm) is modeled in the simulation.
Figure 3-39 Contour of effective plastic strain, two braces without gap

As can be seen in Fig.3-39(b), the plastic strain near the intersection between cylinders is still elliptic, which is a bigger ellipse formed with two braces. The deformation pattern coincides quite well with the scenario from simulation.

The comparisons of absorbed energy and crushing resistance between 1 and 2 braces punching situation are shown in Fig. 4-40 and 4-41. As can be seen, the absorbed energy in the case of two braces punching is a bit higher than one brace punching, and the deviation of first peak value of resistance between two cases is 34.3%. In addition, the load curve subsequent to load shedding both cases tend to a similar value.

Figure 4-40 Comparison of absorbed energy between different numbers of braces
The comparison of absorbed energy and crushing resistance obtained analytically and numerically are compared in Fig. 4-42 and 4-43. It reveals that the absorbed energy is well predicted by super-element with small discrepancy around -5.7%. The discrepancy of the peak value between two methods is around -18.9% and the load-displacement curve is underestimated by super-element almost during all the process which is on the safe side.

Figure 4-41 Comparison of crushing resistance between different numbers of braces

Figure 4-42 Comparison of internal energy between two methods, two braces without gap
Figure 4-43 Comparison of crushing resistance between two methods, two braces without gap

4.5.2 Two Braces without gap and span=23.5m

In this section, the cylinder is punched by two braces without gap in a span=23.5m. The comparison of absorbed energy and crushing resistance between super-element and LS-DYNA are shown in Fig. 4-44 and 4-45. As can be seen, the absorbed energy is not predicted as good as before. The maximal discrepancy between two methods is around -16.4% at $d=0.4$m.

Figure 4-44 Comparison of internal energy between two methods, two braces without gap, $L=23.5$m
In this case, the predicted crushing energy by super-element and LS-DYNA both are different from which we have seen before. As can be seen in Fig.4-45, the peak value of load-displacement curve obtained by LS-DYNA remains for a certain period after reached the peak value, and then starts load shedding. On the other hand, the threshold is activated at the starting of the process, it may be attributed to the longer span which lowers the threshold according to the formula.

4.5.3 Two Braces with wide gap and span=17m

In this subsection, the cylinder is punched by two braces separated by a significant gap. The distance of the gap, $g=0.82$m, is selected from the jacket dimension of the model provided by STX in the framework of the CHARGEOL project. The contour of effective plastic strain and the section view of the punched leg generator are shown in Fig. 4-46. It reveals that the plastic strain near the intersection between cylinders forms two ellipse surrounded by a bigger ellipse. Although the plastic strain region is a bit different from the one without gap (the plastic strain in gap region is slighter than at the interface of cylinders), previous formulas (eq. 4-3 and 4-4) are applied for the sake of simplicity and considering the compensation by the surrounded ellipse.
The internal energy and crushing force between super-element and LS-DYNA are compared in Fig.4-47 and 4-48 respectively. As shown in Fig. 4-47, the internal energy predicted by super-element coincides with the one obtained from LS-DYNA simulation. The deviation between two methods is around -9.6% and basically the internal energy is conservatively underestimated by super-element during the process. For the load-displacement curve, the threshold (transition from local to global mode) is activated at beginning of the crushing process. This may be due to the punched region is wider and the span is longer as well. The threshold can therefore be reached easily in comparison with one brace punching and shorter span.

Figure 4-47 Comparison of internal energy between two methods, two braces with wide gap, $L=17m$
4.5.4 Two Braces with wide gap and $L=23.5m$

The 23.5m span cylinder, punched by two braces separated by a 0.82m gap as previous section, is studied in this subsection. The comparison of internal energy and crushing resistance between the results obtained from the developed super-element and LS-DYNA simulation are shown in Fig.4-49 and 4-50.

Figure 4-48 Comparison of resistance between two methods, two braces with wide gap, $L=17m$

Figure 4-49 Comparison of internal energy between two methods, two braces with wide gap, $L=23.5m$
As internal energy is connected, significant discrepancy, -18.4%, is found between results obtained by super-element and LS-DYNA at $d=0.6m$. This can be attributed to the fact that the threshold is underestimated by the super-element, as shown in Fig.4-50. The threshold is reached at the beginning of the process because of wide punching area and long span. Moreover, the deviation of obtained resistance peak value is around -19%. However, once the threshold is reached, the slopes of both curves are similar.
5 VALIDATION ON A REAL JACKET FOUNDATION

5.1 Model Description

5.1.1 Jacket Foundation Particular

The model of jacket foundation provided by STX France Solution is the same model used in the former stage of CHARGEOL project, which is built using Belytschko-Lin-Tsay shell element. The wind turbine and the mast are represented by a punctual mass, attached to the transition piece by means of rigid beam elements. Moreover, the thickness of all the elements is 50mm except boat landing unit. The dimension and model information and the sketch of the model are shown in the Table 5-1 and Fig. 5-1 respectively.

<table>
<thead>
<tr>
<th>Jacket</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height</td>
<td>63.7 (m)</td>
</tr>
<tr>
<td>Width</td>
<td>18 (m)</td>
</tr>
<tr>
<td>Waterline</td>
<td>34 (m)</td>
</tr>
<tr>
<td>Weight</td>
<td>540 (tons)</td>
</tr>
<tr>
<td>Leg Tube O.D.</td>
<td>1.22 (m)</td>
</tr>
<tr>
<td>Leg Tube Thick.</td>
<td>50 (mm)</td>
</tr>
<tr>
<td>Brace Tube O.D.</td>
<td>0.61 (m)</td>
</tr>
<tr>
<td>Brace Tube Thick.</td>
<td>50 (mm)</td>
</tr>
</tbody>
</table>

Table 5-1 Jacket dimension and model information [29]

Figure 5-1 Sketch of jacket foundation model
5.1.2 Offshore Supply Vessel (OSV) Particular

OSV is a typical vessel for maintenance of offshore wind turbines, which is used as a striking ship in this study for numerical validations. The finite element model of the OSV bow has been built in the former stage of the CHARGEOL project. Only the shell plating has been modeled as the ship is considered as rigid. The rest of the ship is modeled by defining a rigid body, characterized by a mass and inertia matrix associated with the ship center of gravity. The LS-DYNA code PART_INERTIA card is used for defining the center of gravity of the ship and the components of inertia tensor. This card allows the inertia properties and conditions to be defined rather than calculated from the finite element mesh. Moreover, it should be mentioned that this card only applies to rigid bodies (LSTC, 2013 [28]). The particular and the sketch of the OSV are shown in Table 5-2 and Fig.5-2 respectively.

<table>
<thead>
<tr>
<th>OSV Particular</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Length</strong></td>
<td>102.4 m</td>
</tr>
<tr>
<td><strong>Breadth</strong></td>
<td>23.23 m</td>
</tr>
<tr>
<td><strong>Depth</strong></td>
<td>25.89 m</td>
</tr>
<tr>
<td><strong>Draft</strong></td>
<td>4.117 m</td>
</tr>
<tr>
<td><strong>Displacement</strong></td>
<td>5000 tons</td>
</tr>
<tr>
<td><strong>Added mass</strong></td>
<td>5250</td>
</tr>
<tr>
<td><strong>Element Size</strong></td>
<td>65 mm</td>
</tr>
</tbody>
</table>

Table 5-2 OSV particular [29]

![Figure 5-2 Sketch of OSV model](image)

5.1.3 Materials

Mild steel is considered for the jacket foundation and is modeled using the LS-DYNA MAT_PIECEWISE_LINEAR_PLASTICITY card, which is the same material as mentioned in 4.1.1. On the other hand, as aforementioned, the OSV is modeled as rigid by using the LS-DYNA MAT_RIGID card for the bow and the rest of the ship is represented by using PART_INERTIA card. The properties of jacket and OSV are shown in table 5-3.

<table>
<thead>
<tr>
<th>Property</th>
<th>Jacket</th>
<th>OSV</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m$^3$)</td>
<td>7850</td>
<td>7850</td>
</tr>
<tr>
<td>Young’s Modulus (GPa)</td>
<td>200</td>
<td>210</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.3</td>
<td>0.3</td>
</tr>
<tr>
<td>Failure Strain (%)</td>
<td>0.25</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 5-3 Property of jacket and OSV model [29]
5.1.4 Simulation Setup

The duration of the simulation is 2 seconds which is deemed to be sufficient for simulating the entire crushing process. The duration of simulations are set up in the LS-DYNA code CONTROL_TERMINATION card and each time step is calculated automatically by LS-DYNA based on both mesh size and material characteristics as explained in 2.5.2.

5.1.5 Boundary condition and contact

The boundary condition on the leg near mud line are considered to be fixed, and the constraint is defined by using BOUNDARY_SPC_SET_BIRTH_DEATH card in LS-DYNA as mentioned in 4.1.1. The deformation of the jacket can be very large and predetermination of where and how contact will take place may be difficult or impossible. In this case, the automatic contact is recommended as these contacts are non-oriented, meaning they can detect penetration coming from either side of a shell element. Therefore, the contact between jacket foundation and rigid striking ship is defined by using the LS-DYNA CONTACT_AUTOMATIC_SURFACE_TO_SURFACE card. The impacted leg and rigid ship are defined as the Master and Slave respectively. As aforementioned, the striking ship is assumed to be rigid in order to be conservative. In the case of rigid body contact, the rigid body should have a reasonable fine mesh so as to capture the true geometry of the rigid part, which prevents the contact instability. The node spacing on the contact surface of the rigid body should be no coarser than the mesh of any deformable part which comes into contact with the rigid body. Moreover, no stress, strain or displacement is calculated for rigid body, so fine mesh used for the rigid has minor effect on CPU requirement (LSTC, 2014. [30]). In addition, the leg near mud line is set to be rigid in order to prevent overall shearing movement of the jacket for catching precisely the displacements associated to the punching phenomenon.

5.1.6 Punching Scenarios

From ship-jacket collision simulations, three different punching scenarios are observed:

I. The jacket is collided by the bow and the bulb of the striking ship at the junction and mid-span of the cylinder respectively. The rear leg is punched by the impacted leg through one brace and the adjacent brace is free to move. Therefore, the intersection between the adjacent brace and the leg sustains only global mode.

II. Only one junction on the jacket is collided by the bow of the ship but the cylinder is not collided by the bulb. The rear leg is punched by the impacted leg through one brace but adjacent brace is restrained by the collided leg since it is not collided by the bulb. Therefore, the span of the cylinder is considered to be shorter, and only local indentation occurs at the intersection between punching brace and leg during the collision process.
III. Two junctions on the jacket are collided by the bow and the bulb of the striking ship simultaneously, and the rear leg is punched by the impacted leg through two braces. The sketches of these three phenomena are shown in the figure below:

Figure 5-3 Sketch of different punching scenarios
5.2 Study of Different Scenarios and Colliding Angles

5.2.1 Assessment of the rear leg indentation

In this section, the leg joint collision with three different colliding angles are studied for validation of the new developed super-element, namely 45°, 60°, 90°, and the sketches of ship-jacket collisions at different angles are shown in Fig.5-4. As aforementioned, there are three scenarios observed in the crushing process, therefore, five cases are studied, which are 90° (I, II and III), 45° (I) and 60° (I).

(a) 90°  (b) 60°  (c) 45°

Figure 5-4 Top view of ship-jacket collisions

In order to post-process the absorbed energy accurately in the punched region and the associated cylinder between two support junctions, the rear leg is divided into several parts as shown in Fig. 5-5. In addition, the internal energy of braces which transfer the displacement is post-process as well.

Figure 5-5 Layout of different parts

By observation, the total displacement of the punching area can be classified into three groups,
which are local denting, global bending, and overall movement of junctions which support the punched cylinder. Nevertheless, the movement of junctions are not participating in dissipating the energy while only the punching process is considered. Therefore, the displacement of junctions should be excluded while evaluating the indentation of punched leg. The displacement at different points for three scenarios are shown in Fig. 5-6–8.

![Figure 5-6 Displacement at different points, Scenario I](image)

For scenario I, the displacements at both support junctions are appreciable and should be excluded from the displacement of punched region. By observing the animation and the displacement resultant curves above, it is found that all the points have the same displacement at the beginning of the process ($t<0.3s$) and the process subsequent to $t=0.3s$ is going in the same trend except lower junction (point A). Therefore, the indentation of the punched rear leg is considered as the difference between displacements at the punched point (E) and the upper junction (B).

![Figure 5-7 Displacement at different points, Scenario II](image)
For scenario II, the displacement of points B, C, D and F are nearly the same. This demonstrates that points B and D can be considered as the support instead of B and A; moreover, there is almost no global bending between point B and D. Therefore, the displacement at punched region (E) should exclude the displacement of the support (B). As mentioned in previous section, the adjacent brace is restrained by the collided leg, hence, only the local mode obtained from the super-element is compared to the results obtained by LS-DYNA.

Figure 5-8 Displacement at different points, Scenario III

For scenario III, the indentation involved in the punching process is considered as the difference between the displacements at punched region and upper junction. In addition, the local and global modes are considered in this scenario, even though only minor difference exists between the displacements of the two punched region and their backside.
5.2.2 Case 1: 90°- Leg punched by one brace without adjacent brace restraint (scenario I)

In this subsection, scenario I is studied for 90° collision angle and the contour of effective plastic strain is shown in Fig.5-9. Since the colliding angle is equal to 90°, the plastic strain mainly happens in legs 1 & 2, and the plastic strain near the mud line is due to shear force at the root of the leg. In order to catch correctly the displacement of punched region, the jacket transversal displacement due to shearing effect is eliminated by considering the base of the legs as rigid. The plastic strain occurs at some joints as well, which is due to the relative movement of different parts on the jacket. Nevertheless, it is negligible in comparison to collided and punched legs. In addition, the plastic strain highlighted by a green ellipse is due to the global movement of the collided leg, which is out of the scope of the thesis. Therefore, this part will not be discussed here.

The energy absorbed by the leg which is punched by colliding joint through the brace in the red ellipse is compared to the energy predicted by new developed super-element. The span between two supports is 23.5m ($L_1$=9m and $L_2$=14.5m).

![Figure 5-9 Contour of effective plastic strain and length of span, 90°, scenario I](image)

The internal energies obtained from super-element and LS-DYNA simulations are compared in Fig. 5-10. As can be seen, the internal energy is predicted quiet accurately by the developed super-element for indentation lower than $d$=0.6m, and the maximal discrepancy at $d$=0.6m is
around 8%. The absorbed energy for indentation higher than \( d = 0.6 \) m is obviously overestimated as the slope of energy curve is suddenly dropped.

![Jacket_90°, Scenario I_Internal energy](image)

**Figure 5-10** Comparison of internal energy between two methods, 90°, scenario I

By observing the crushing process in LS-DYNA simulation, buckling is observed in the brace which transfers the displacement from collided leg to rear leg at \( t = 0.62 \) s, \( d = 0.6 \) m. The curve of indentation versus time in the punched region is shown in Fig 5-11. This curve reveals that the indentation is zero at the beginning of the collision process, and the punching process starts after certain time, then the indenting speed decreases at \( d = 0.6 \) m due to the starting of the buckling which prohibits the brace from transferring the displacement. The punching process terminates at around \( d = 0.81 \) m when the braces are completely buckled.

![Jacket_90°, Scenario I_Indentation](image)

**Figure 5-11** Punching indentation versus time, 90°, scenario I
It also can be demonstrated by plotting the energy absorbed by the punched region (part 596 only) and by the braces, as shown in Fig. 5-12 and 5-13 respectively.

![Image of graph showing energy absorbed by punched region](image)

**Figure 5-12 Energy absorbed by punched region**

As can be seen in Fig. 5-12, the energy rate of part 596 decreases at $t=0.62\text{s}$ and then decreases again at $t=0.8\text{s}$. This phenomena are first due to the starting of the brace buckling and then to its complete buckling. Nevertheless, the absorbed energy still increases after $t=0.8\text{s}$, and this may be attributed to different moving speeds of different points in punched region which results from buckling.

![Image of graph showing energy absorbed by braces](image)

**Figure 5-13 Energy absorbed by the braces**

The rates of energies absorbed by the two braces 559 and 567, which transfer the displacement decrease at $t=0.62\text{s}$, and then the absorbed energies stop increasing at $t=0.8\text{s}$. The rate of energy absorbed by brace 560 almost stops increasing since braces 559 and 567 stop transferring the displacement from impacted leg at $t=0.8\text{s}$.
5.2.3 Case 2: 90°- Leg punched by one brace with adjacent brace restraint (scenario II)

Scenario II in 90° collision angle is studied in this subsection. As aforementioned, the punched region during the punching process is restrained by the adjacent brace. Therefore, the span between both supports is considered as 10.5m ($L_1=9 \text{ m}$ and $L_2=1.5 \text{ m}$), as shown in Fig. 5-14 where the contour of effective plastic strain is depicted.

![Figure 5-14 Contour of effective plastic strain and length of span, 90°, scenario II](image)

The energy absorbed by the leg which is punched by colliding joint through the brace in the red ellipse is compared to the energy predicted by new developed super-element, as shown in Fig.5-15. As can be seen, the absorbed energy obtained by super element is only in good agreement at the very beginning of the process. For the process subsequent to $d=0.03\text{ m}$, the energy suddenly increases due to the different moving speed of two braces, the maximal discrepancy is around -50.95% at $d=0.06\text{ m}$. While two braces are moving in different speed, not only the upper brace is restrained by the lower brace, but also the lower brace is sustaining punching shear force which is an axial tension force to the chord. This punching shear force is not accounted for by using the super-element, which makes the energy higher. Moreover, as mentioned before, only local mode is considered in this case, and the transition is almost the end of the punching process which is indicated in Fig.5-15. Therefore, it is deemed to be comparable.
The braces start buckling at around $t=0.55s$, $d=0.23m$, which can be seen in Fig. 5-16, and there are only small indentation and fluctuation after buckling.

The influence of bucking can also be observed in Fig. 5-17 and 5-18, where the time evolution of energies dissipated in punched leg and braces are plotted. The absorbed energy rate obviously decreases after buckling in the interface between leg and brace (part 596). On the other hand, the energy absorbed by braces is more complicated. The absorbed energy rate decreases when the braces start buckling and increases after a short period then decreases again. This fluctuation is due to the behavior of the brace buckling, which is not studied in this thesis, but it is an interesting task for further study.
5.2.4 Case 3: 90° - Leg punched by two braces (scenario III)

In case 3, scenario III for colliding angle 90° is studied. The contour of effective plastic strain is shown in Fig.5-19. As can be seen, the plastic strain occurs on legs 1 & 2. More precisely, the plasticized areas highlighted by green ellipses are due to the global modes of the collided leg. Plasticized area highlighted by red ellipse which corresponds to rear leg punched area will be studied here, and the absorbed energy in this area is compared to the energy predicted by new developed super-element. In the studied section, the span between two supports is 23.5m ($L_1=9m$ and $L_2=14.5m$).
The internal energies obtained by developed super-element and LS-DYNA simulation are compared in Fig.5-20. As can be seen, the absorbed energy obtained by developed super-element is in good agreement with LS-DYNA as far as the indentation is lower than $d=0.23\text{m}$. While the indentation exceeds $d=0.23\text{m}$, the absorbed energy rate suddenly decreases then restarts increasing with the same rate. This is due to an abrupt global movement and asynchronous displacement of each nodes in the punched region at $t=0.468\text{s}$, $d=0.23\text{m}$. While the global movement occurs abruptly, there is almost no absorbed energy, which can be seen in Fig.5-21.
After the abrupt global movement, the displacements of different nodes in the punched region are slightly different, as can be seen in Fig. 5-22. Therefore, the absorbed energy rate is not constant after \( d = 0.23 \text{ m} \), as shown in Fig. 5-20.

The buckling of the braces occurs around \( t = 0.718 \text{ s} \), \( d = 0.46 \text{ m} \), which makes the absorbed energy rate obviously decreasing (see Fig. 5-20). It is also demonstrated by the indentation time evolution depicted in Fig. 5-23; as can be seen, the slope of the displacement curve starts decreasing at \( t = 0.71 \text{ s} \). The maximal discrepancy between analytical and numerical results before buckling is 10.8% at \( d = 0.36 \text{ m} \), as shown in Fig. 5-20.
The abrupt global movement may be due to the buckling of brace (part 559), as can be seen in Fig.5-24, which also leads to no more absorbed energy in other braces (Fig.5-25). The braces (part 558, 562 and 566) buckle at around $t=0.7s$. 

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**Figure 5-23** Punching indentation versus time, 90º, scenario III

**Figure 5-24** Absorbed energy of braces, 90º, scenario III

**Figure 5-25** Absorbed energy of braces, 90º, scenario III
5.2.5 Case 4: 45° - Leg punched by one brace without adjacent brace restraint (scenario I)

In case 4, the jacket is collided by the striking ship with a 45° angle and scenario I is studied. As the collision angle is not a right angle, the assumption in section 3.1.1 is used, and it can be verified by using the displacement time evolutions depicted in Fig. 5-26. As can be seen for the process before $t=0.43s$, the global displacement of the collided leg is 0.793m and the displacement of the rear leg on both side are 0.552m. Based on the assumption, the displacement of the rear legs suppose to be $0.793 \times \sin 45° = 0.56$m, the discrepancy between assumption and simulation is around 1.4%. Nevertheless, as can be seen, the displacements of two rear legs become different for the process subsequent to $t=0.43s$, it is due to the unsymmetrical mesh that makes the ship going either side rather than going straight forward. To demonstrate this, the large inertia is given to the ship to keep the ship going straight forward, as can be seen in Fig. 5-27, although the difference still exists because of unsymmetrical mesh, but the difference between the displacement of two rear legs is smaller. It can be demonstrated that if ship navigates straight forward for entire process, the displacements of two rear legs should be nearly the same.

![Figure 5-26 Displacement of collided leg and punched rear legs](image)

![Figure 5-27 Displacement of collided leg and punched rear legs with large ship inertia](image)
The contour of effective plastic strain is shown in Fig. 5-28, and as can be seen, the plastic strain mainly occurs on legs 1, 2 and 4. Therefore, the internal energies calculated by super-element and by LS-DYNA in punched leg region (highlighted by green ellipse on one of the rear legs) are compared in Fig. 5-29.

As shown in Fig. 5-29, the energy absorbed by the punched leg is predicted correctly until $d=0.19\text{m (}t=0.51\text{s)}$, and the energy absorbed rate abruptly increases at $d=0.19\text{m}$ due to buckling of the brace 562. In this case, buckling occurs on both braces 560 and 562, but two braces buckle at different time step. While buckling occurs on the brace 562, the indentation drops as shown in Fig. 5-30, and the punched region is restrained by the buckled brace. The buckling of brace 562 not only affects the transformation of the indentation but also provides a restraint while it buckles, it is also the reason why the absorbed energy rate increases while buckling occurs instead of decreasing. The brace 560 starts buckling at $d=0.23\text{m (}t=0.65\text{s)}$, and two braces move at the same speed until the ship starts moving backward. Therefore, the punching process is considered to terminate at $d=0.23\text{m}$, it can also be demonstrated in Fig. 5-30.
It should be mentioned that there are some fluctuation of indentation at the beginning of the process in this case, and the energy is absorbed by the other parts of the leg, as shown in Fig. 5-31 and 5-32. As shown in Fig.5-31, the fluctuation occurs from \( t=0s \) to \( t=0.17s \), and this fluctuation makes a small difference (≈0.02m) between two positions. It is observed that the energy from the fluctuation is mainly absorbed by the other part of the punched leg, namely part 529, as shown in Fig.5-32. Therefore, it is necessary to eliminate the difference between two positions in order to obtain correct indentation for punching. The punching process is considered starting at \( t=0.17s \), and the indentation associated to punching is considered to be zero for the duration \( t=0-0.17s \). The position of different parts are shown in Fig.5-33.
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Figure 5-31 Fluctuation of indentation

Figure 5-32 Energy absorbed by different parts of the punched leg

Figure 5-33 Positions of different parts

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5.2.6 Case 5: 60° - Leg punched by one brace without adjacent brace restraint (scenario I)
In case 5, the jacket is collided by the striking ship with a 60° angle and scenario I is studied. The contour of effective plastic strain is shown in Fig.5-33, as can be seen, the plastic strain mainly occurs on legs 1 and 2. Therefore, the internal energies calculated by super-element and by LS-DYNA in punched leg region (highlighted by red ellipse) are compared in Fig.5-34.

![Figure 5-33 Contour of effective plastic strain, 60°, scenario I](image)

![Figure 5-34 Energy absorbed by punched region, 60°, scenario I](image)
In general, the absorbed energy is predicted correctly by the developed super-element until \( d = 0.61 \) m. For the process subsequent to \( d = 0.61 \) m, the buckling occurs on braces 558 and 559, and the indentation is obviously decreased, as shown in Fig.5-35. Although both braces 558 and 559 buckle, extent of buckling is different and the brace 558 is more severe. The buckling of brace 558 not only affects the transformation of the evolution indentation but also provides a restraint while it buckles. It is also the reason why the absorbed energy rate is increased while buckling occurs instead of decreasing.

![Jacket_60°, Scenario I_Indentation](image)

Figure 5-35 Indentation versus time, 60°, scenario I

The restraint provided by brace 558 can be demonstrated by plotting the displacement of the nodes in punched region, as shown in Fig.5-36. As can be seen, the displacement are almost the same at the beginning of the punching process, and then the nodes start to move at different speed at \( t = 0.7 \) s. But later they move at the same speed again, which is due to the buckling on brace 559.

![Displacement of punched region](image)

Figure 5-36 Displacement of different nodes in punched region
The energy absorbed by the punched region is shown in Fig.5-37. Because of brace 558 buckling, the absorbed energy rate increases at $t=0.7s$. Later on, the absorbed energy rate decreases a bit at $t=0.76s$, which due to buckling of brace 559.

![Figure 5-37 Energy absorbed by punched region, 60°, scenario I](image1)

The energies absorbed by different braces are shown in Fig.5-38 and 5-39 respectively. As can be seen, the absorbed energy rate decreases at $t=0.7s$ and $t=0.76s$ for braces 558 and 559 respectively. These results are consistent with the resultant mentioned above.

![Figure 5-38 Energy absorbed by braces, 60°, scenario I](image2)
Figure 5-39 Energy absorbed by braces, 60°, scenario I
6 SUMMARY AND CONCLUSION

In this thesis, a new super-element for predicting the deformation energy of wind turbine foundation due to punching in ship-jacket foundation collision is developed based on virtual work principle. By observation of finite element simulation results, the proper displacement and velocity fields are chosen for calculating the internal energy rate. In general, the absorbed energy is predicted well by the new developed super-element, which has been demonstrated by both the simple cylinder punching cases and ship-jacket collisions.

In the cases of simple cylinder punching, several sensibility studies to different factors are done, such as $\beta$ ratio (ratio between brace and chord diameters), angle between brace and chord, span of cylinder, different number of braces, and the gap between braces. For sensibility to $\beta$ ratio, five different $\beta$ ratios are studied and the discrepancy increases along with the increasing $\beta$ value, but absorbed energy for all cases is in good agreement with numerical solution in substance. For sensibility to angle, span, number of braces, and gap, the absorbed energies are well-predicted. Nevertheless, while combining the parameters two braces and long span, the accuracy decreases for all the process after $d=0.2m$ with -19% discrepancy; however, it is considered to be acceptable since super-element results remains conservative.

In the case of ship-jacket collision, the rear legs are punched by the collided leg through the braces, and three punching scenarios are found and studied. Several conclusions are made for different cases as below:

1. For scenario I and III, the energy absorbed by the punched leg is predicted accurately by the new developed super-element for the punching process before buckling occurs on the braces.

2. For scenario II, the absorbed energy is not predicted as good as other two scenarios which results from extremely short span. Since it remains conservative, it is considered to be acceptable. Nevertheless, it should be noted that this scenario should be improved in further study.

3. In the cases of different collision angle, 45°, 60° and 90°, the absorbed energies predicted by new developed super-element are in good agreement with LS-DYNA simulation in all the cases.

In order to treat a complete real jacket, analytical calculations are performed by using:

- The super-elements developed in a previous study [6] to estimate the absorbed energy when the leg is impacted by bow and bulb (red and orange ellipses in Fig.6-1),

- The new super-element developed in this study to assess the energy absorbed in all the
regions where punching occurs (green ellipse in Fig. 6-1).

The absorbed energies predicted analytically are then compared to total energy absorbed by LS-DYNA simulation, as shown in Fig. 6-2.

Figure 6-1 Positions of super-element applied, 90 °, scenario I.

Figure 6-2 Comparison of overall absorbed energy between LS-DYNA and super-elements
In comparison to overall absorbed energy obtained by LS-DYNA, the distribution of absorbed energies predicted by super-elements for different parts are shown as following:

- bow: 19.75%,
- bulb: 22.98%,
- punching 1: 5.80%,
- punching 2: 4.64%,
- punching 3: 15.55%.

While taking into account the punching phenomenon, the result is improved by 25.99% of overall energy. Nevertheless, the new developed super-element is used in punching 2 and 3 for an approximate prediction. Since the assumption of indentation is not applicable and the relative displacement between nodes is unknown, the discrepancy may exist. In addition, the difference between total energy absorbed by jacket and the energies predicted by super-elements is around 31.28%.

The absorbed energies predicted by super-elements (bow+bulb+punching3) for impacted and rear legs are compared to LS-DYNA results, as shown in Fig.6-3 and Fig.6-4. As impacted leg is concerned, the absorbed energy predicted by super-elements is in good agreement with LS-DYNA simulation.

![Comparison of energy absorbed by impacted leg between LS-DYNA and super-elements](image)

In terms of the rear leg, the absorbed energy is slightly overestimated by super-elements when the ship displacement varies from 1.9m to 2.7m, which is due to the buckling of the braces. While the ship displacement exceeds 2.7m, the absorbed energy remains constant, which means that the two punching processes are terminated. As can be seen, the difference between the
absorbed energy predicted by super-elements and LS-DYNA simulation is still appreciable. It implies that some parts of rear leg which are involved should be taken into account. By observation, the part near mudline absorbs 29% of the total rear leg energy, which is due to shearing effect, as shown in Fig.6-5. Therefore, it should be accounted for a more accurate prediction.

Figure 6-4 Comparison of energy absorbed by rear leg between LS-DYNA and super-elements

Figure 6-5 Effective plastic strain caused by shearing effect
7 FUTURE WORK

I. Knowing relative displacements between nodes on the jacket model is required, since the displacement in punching process is difficult to derive directly. The energy predicted by super-element is strongly connected with the indentation of each time step during the process. Therefore, deriving relative displacement between brace/leg joints would noticeably improve the results.

II. Another punching scenario is also found at the junction between two joints which are collided by the bow and the bulb of the striking ship, the sketch is shown in Fig. 7-1. This scenario is that the collided led is indirectly punched by brace due to the global moving of the collided leg. The energy absorbed by this punched region is around 15% of the energy absorbed by collided leg, as observed in the F.E. simulation. In this case, the displacement field considered in this thesis may be used, but the indentation at each time step need to be derived. In order to compute the absorbed energy of this scenario, it is necessary to know the relative displacement between joints.

![Figure 7-1 Scenario of collided leg punching](image)

III. Buckling of the brace which transfer the displacement from collided leg should be considered, since it affects the punching process and absorbs some energy before the buckling happens.

IV. The energies absorbed by different parts derived in LS-DYNA simulation are compared in Fig. 7-2. As can be seen, the energy dissipated by other legs or braces is still appreciable. Although, the energy absorbed by collided leg and rear leg is around 80% of overall energy, the internal energy absorbed by collided and rear leg is not only due to bow/bulb impact and direct punching. Therefore, it is necessary to consider more phenomena in the super-element, ex. buckling of braces, shearing near the mudline and any other patterns of punching.
Figure 7-2 Energies absorbed by different parts of jacket in LS-DYNA simulation, 90 (J)
8 ACKNOWLEDGEMENTS

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Analytical formulations for ship-offshore wind turbine collisions

9 REFERENCE

11. MIT OpenCourseWare Mechanical Engineering, 2004. Lecture note. Available from:

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Books
APPENDIX A DISPLACEMENT FIELD DETAIL

The ring of the cylinder can be split up into three parts, which are highlighted in different color in Fig. A-1. The displacement field of each part on the ring is defined by the indentation $\delta$ and an angle $\alpha$ on original ring, which is proposed by (T. Wierzbicki and M.S. Suh, 1988. [16]).

For $0 \leq \alpha \leq \frac{(R_1-R_2)}{R} \sin \phi$ (in red),

$$\left( \frac{W_0}{R} \right)^2 = \left( \alpha - \sin \alpha \right)^2 + \left( 1 - \delta - \cos \alpha \right)^2$$  \hspace{1cm} (A-1)

For $\frac{(R_1-R_2)}{R} \sin \phi \leq \alpha \leq \pi - \frac{R_1}{R} \phi$ (in green),

$$\left( \frac{W_0}{R} \right)^2 = \left[ \left( \frac{R_1}{R} + \frac{R_2}{R} \right) \sin \phi + \frac{R_2}{R} \sin \beta - \sin \phi \right]^2 + \left[ 1 - \delta - \frac{R_2}{R} (1 - \cos \beta) - \cos \alpha \right]^2$$

where $\beta = \frac{R}{R_2} \alpha - \left( \frac{R_1}{R_2} - 1 \right) \sin \phi$.  \hspace{1cm} (A-2)

For $\pi - \frac{R_1}{R} \phi \leq \alpha \leq \pi$ (in blue),

$$\left( \frac{W_0}{R} \right)^2 = \left[ \frac{R_1}{R} \sin (\phi - \psi) - \sin \alpha \right]^2 + \left\{ \frac{R_2}{R} \left[ 1 - \cos(\phi - \psi) \right] - 1 - \cos \alpha \right\}^2$$

where $\psi = (\alpha - \pi) \frac{R}{R_1} + \phi$.  \hspace{1cm} (A-3)

![Figure A-1 Deformation pattern and symbols used by T. Wierzbicki and M.S. Suh](image)

The velocity field is also proposed by T. Wierzbicki and M.S. Suh, which is derived from the differentiation with respect to $\phi$. 

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For $0 \leq \alpha \leq \frac{(R_1 - R_2)}{R} \sin \phi$ (in red),

$$\frac{w_0 \dot{w}_0}{R^2} = (1 - \delta - \cos \alpha)(-\delta)$$ \hspace{1cm} (A-4)

For $\frac{(R_1 - R_2)}{R} \sin \phi \leq \alpha \leq \pi - \frac{R_1}{R} \phi$ (in green),

$$\frac{w_0 \dot{w}_0}{R^2} = \left[ \frac{R_1}{R} \frac{R_2}{R} \sin \phi + \frac{R_2}{R} \sin \beta - \sin \alpha \right] \left[ \frac{R_1}{R} \frac{R_2}{R} \sin \phi + \frac{R_3}{R} \frac{R_2}{R} \cos \phi + \frac{R_2}{R} \sin \beta + \frac{R_2}{R} \cos \beta \frac{\dot{\phi}}{R} \right]
+ \left[ 1 - \delta - \frac{R_2}{R} (1 - \cos \beta) - \cos \alpha \right] \left[ -\delta - \frac{R_2}{R} (1 - \cos \beta) - \frac{R_2}{R} \sin \beta \frac{\dot{\beta}}{R} \right]$$ \hspace{1cm} (A-5)

where $\beta = \frac{R}{R_2} \alpha - \left( \frac{R_1}{R_2} - 1 \right) \sin \phi$.

For $\pi - \frac{R_1}{R} \phi \leq \alpha \leq \pi$ (in blue),

$$\frac{w_0 \dot{w}_0}{R^2} = \left[ \frac{R_1}{R} \sin(\phi - \psi) - \sin \alpha \right] \left[ \frac{R_1}{R} \sin(\phi - \psi) + \frac{R_1}{R} \cos(\phi - \psi)(1 - \psi) \right]
+ \left( \frac{R_1}{R} \right) \left[ 1 - \cos(\phi - \psi) - 1 - \cos \alpha \right] \left( \frac{R_1}{R} \left[ 1 - \cos(\phi - \psi) \right] + \frac{R_1}{R} \sin(\phi - \psi)(1 - \psi) \right)$$ \hspace{1cm} (A-6)

Note:
1. $w_0$ is function of $\alpha$ and $\delta$ (= $w(\alpha, \delta)$)
2. The symbols used by T. Wierzbicki and M.S. Suh are different from the ones used in this thesis. The relation between these two is as following:

$$\alpha = \theta$$
$$\phi = \pi - \psi$$
$$\beta = \frac{R}{R_2} \theta - \left( \frac{R_1}{R_2} - 1 \right) \sin \pi - \psi.$$  
$$\psi = (\theta - \pi) \frac{R}{R_1} + \pi - \psi$$

(Symbols in red are used by T. Wierzbicki and M.S. Suh.)

Figure A-2 Deformation pattern and symbols used in this thesis
APPENDIX B VALIDATION FOR DIFFERENT LEG DIAMETER

An extra validation is done for testing a bigger leg diameter. By observation, the ring is considered completely crushed at $\psi = 1.25\pi$, and $\frac{1}{3}\pi$ is taken for $\psi_0$ as before for high resistance at the beginning. Since the leg diameter is bigger, the local denting can go deeper which makes $\psi > \pi$ possible, as the sketch shown in Fig.B-1. The plastic hinges are depicted on the figure. Although the curve between two hinges is not an arc, it is assumed that the curve is represented by an arc with radius $R_2$ from point A to B. The internal energy and crushing resistance obtained by super-element is compared to the resolution from LS-DYNA simulation, as shown in Fig.B-2 and B-3.

![Figure B-1 Crushed ring section, $D_L=2.5\text{m}$, $\beta=0.48$](image1)

![Figure B-2 Comparison of internal energy between two methods, $D_L=2.5\text{m}$, $\beta=0.48$](image2)
As can be seen, energy and resistance obtained by super-element is in good agreement with LS-DYNA simulation. Moreover, it is found that different parameters may be used for different leg diameters.

Figure B-3 Comparison of crushing resistance between two methods, $D_L=2.5\text{m}$, $\beta=0.48$