

# **COLLISION ANALYSIS OF OFFSHORE WIND TURBINE JACKETS IMPACTED BY A SHIP**

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## INTRODUCTION

Safety and reliability is the main concerned topic for every industries that related to offshore activities. Nowadays, these activities keep actively occur due to the increasing number of offshore wind farm projects especially in European area which may leads to a higher probability to have a collision incident. If this accident effect has not been taking into account since in the early stage of the project, there will be a huge cost to overcome all the problems afterwards. Therefore the collision analysis become more important to ensure the safety of such operations.

The aim of this study is to developed an simplified analytical tool to calculate the resistance of the offshore wind turbine jackets when it gets impacted by considering the buckling and punching phenomenon. This will help the engineers to have a better judgment when working at preliminary design stage. The behavior of the jacket will be studied in several cases by modifying the impact velocity and location. Finally, the results of this tool will be compared with the nonlinear finite element analysis from LS-DYNA in order to validate the performance of this simplified tool. In addition, a study for validate these results with the rules from DNV-GL, BV, and LR in terms of collision analysis will be performed as well.

## OFFSHORE WIND TURBINES

Offshore wind turbines (OWT) has become popular in the last decade due to the fact that it has a better performance compare to the onshore wind turbine. But, it is quite challenging to built an offshore structures which are subjected to many loads such as wave, wind, potential ship impacts, and wind turbine component itself. So, there are several types of offshore wind turbine structures which mainly based on the operational characteristics and its dimension. These support structures may be defined as the Figure 1,

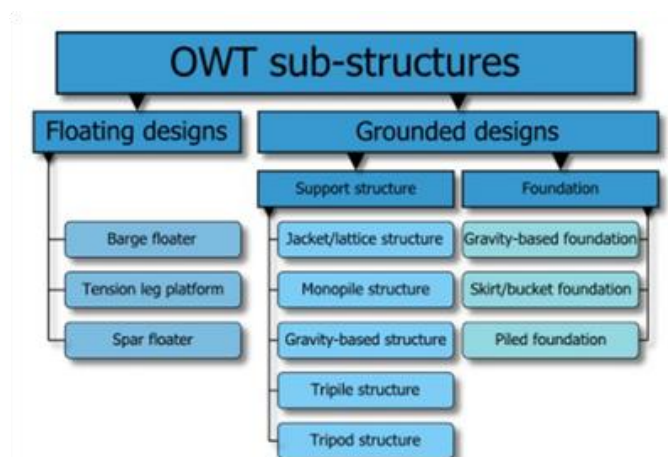


Figure 1. Offshore Wind Turbine Sub-Structures Classification [14]

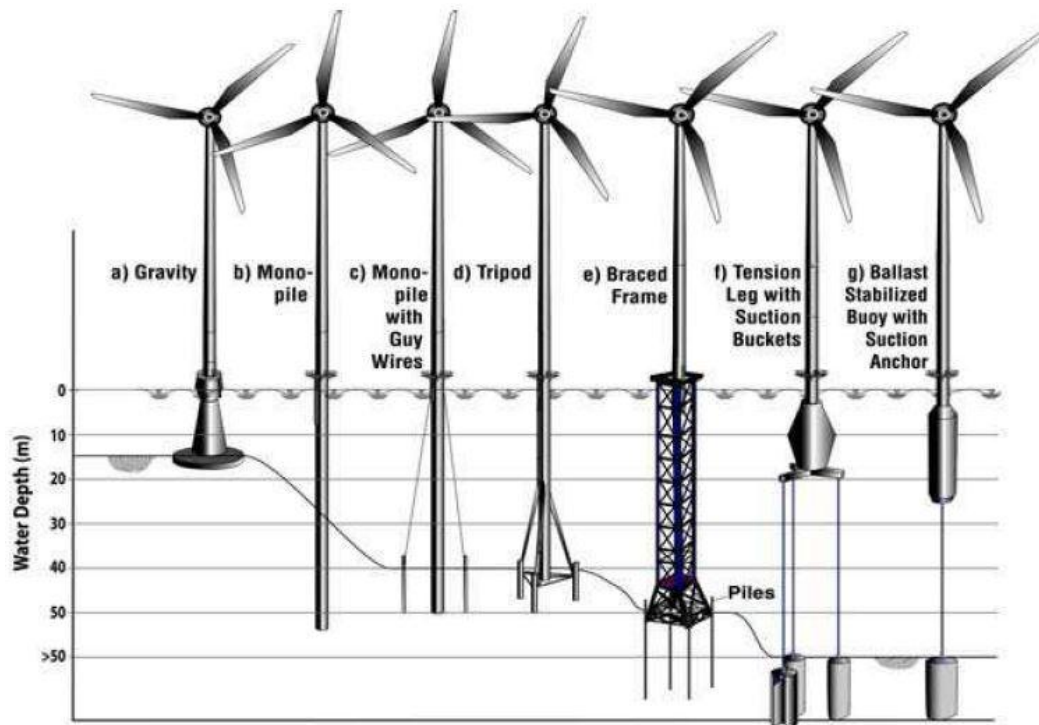


Figure 2. Commonly Known OWT Sub-Structures [23]

### GRAVITY BASED STRUCTURES

The principal system of the structure could be imagined as its name, the gravity force will help the structure to resist the tensile loads between the bottom of the structure and the seabed. It is been used in relatively shallow water as in Figure 2 or when the seabed is rocky. Generally, concrete is the main material to create this structures and it leads to a better investment when the environmental loads are relatively low. The transport of this structure could be easier by implementing the water ballast system which could make it floating and towable with the tugs. This is one of a good solution to perform the OWT installations where the heavy lift or offshore service vessel could not provide.

### MONOPILE STRUCTURES

This type is the most commonly used for offshore wind turbines until now. Normally, it is built from the large diameter steel pipe up to 6 m and 150 mm thickness. It is hammered to the seabed by using a large impact or it will be connected to a sockets that had been drilled to the seabed. Due to the simplicity of its shape, it is easier to predict the force acting on this structure based on some equations. The usual range of water depth to implement this system is  $\leq 30$  m. It may be used on deeper water but a tensioned guy wires shall be used in order to stabilize the structures. A limiting condition of this type is the overall deflection, vibration, and subjected to large cyclic, lateral loads and bending moments also axial loads.

**TRIPOD STRUCTURES**

When the location of the OWT installations getting deeper, the environmental loads will also getting worse. This leads the engineer to design a stronger structures such as this tripod system. This system consist of several elements such as the jacket leg and the piles. Usually these item are constructed at the yard and will be transported by a supply vessel or a barge. One of the advantages of this system is that the seabed preparation has no longer become an issue. Also, the stiffness of this structure is bigger compared to the previous type and it will provide a good stability. This system require a certain depth depend on the design due to its operational activity of the approaching vessel which may have some problem with the draught. In 2009, tripods were installed in 30 m of water for the Alpha Ventus wind farm located 45 km from the island of Borkum, Germany.

**JACKET/LATTICE STRUCTURES**

This system basically is a modification of a tripod system by using more structural members. Generally it has 3 or 4 jackets leg which stiffened by a smaller tubular steel pipe bracing to create a higher stiffness. It is used for a deeper water depth which is around 20-50 m and it will be improved as the demand of OWT is increasing in terms of deeper installation until 70 m. Due to its complexity of the fabrication and level of the transportation difficulty, it will require a higher initial investment and probably the maintenance cost as well. But, as the market grows, there will be a lot of industries try to develop a smarter way to built such structure around the world which leads to a broader choice of the OWT producer.

**FLOATING TENSION LEG STRUCTURES**

This floating systems has the ability to perform in deeper water depth due to its simplicity without using any structural members attached continuously to the seabed. It could be performed by using a submerged part as can be seen in Figure 2 as a dampener of its own motion due to environmental loads. Moreover, an anchoring system that attached to the submerged part also become another solution. By using a ballast system, it is possible to control the behavior of the OWT in order to ease the installation operation. But, as the physics become really dynamic in terms of the motion and impact forces, a well-adapted design is needed to overcome the problems. This is the main reason that such system still being studied in order to broaden the wind farm location which is necessary to achieve the European Union target by 2020.

## RISK ASSESSMENT

The demand for the energy will never end as long as people live in this earth. But, there is a problem on how people will generate this energy due to environmental reasons which has been really important in this era. The extraction of the fossil fuel always create some disturbance to its environment and the safety of the nuclear power plant brings too much risk when subjected to the natural disasters. Therefore, the idea to generate the energy with the renewable sources has become really popular in the last decade. One of the most popular sources are coming from the Offshore Wind Turbines Farms as the Figure 3 below. In European Union, it is become more important since they need to achieve the targeted power capacity between 20-40 GW which shall be operating by 2020 and the planning for the following year projects had been studied as Figure 4.

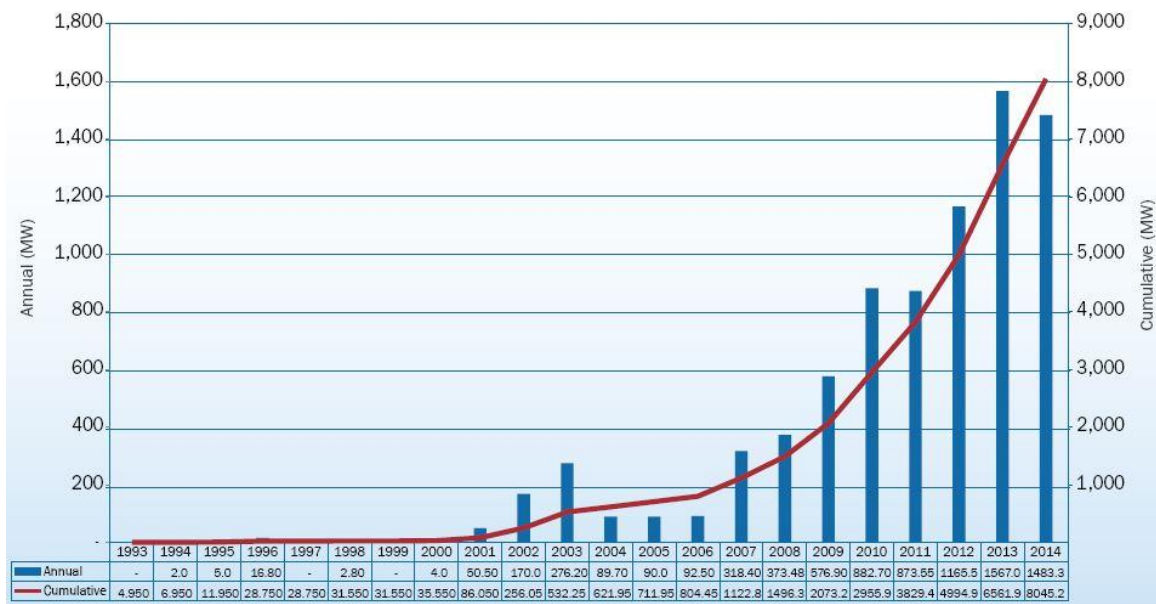


Figure 3. Cumulative and Annual Offshore Wind Installations [14]

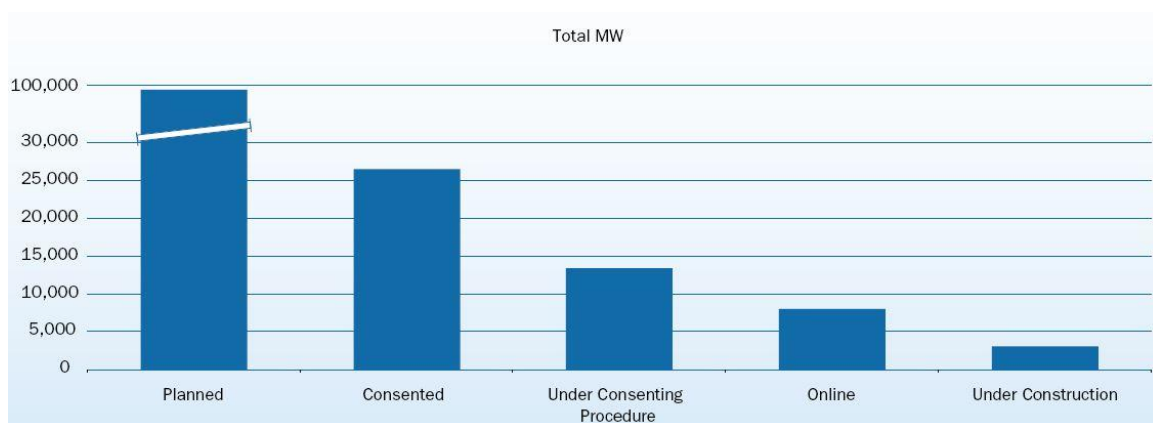
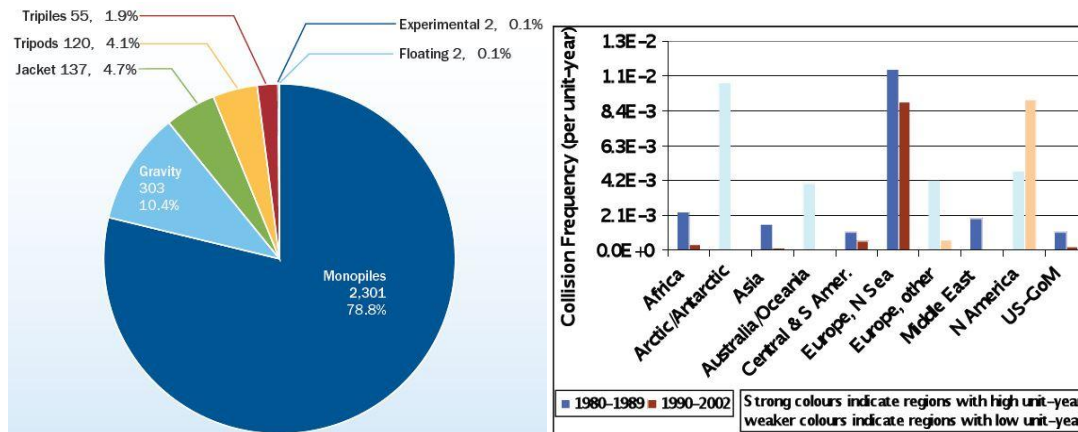


Figure 4. Offshore Wind Turbine Markets [14]

In order to narrow down which kind of the OWT structures need to be studied, a statistical data from the European Wind Energy Association Report 2014 shows that most of them used a monopile structures. In addition, the vessel type that has a higher probability to have a collision with the offshore structures is provided. Finally, to know how important this collision need to be studied, another statistical data of collision frequency being shown here as well. All the Figures 5 below represent the fact declared above.



| Vessel Type    | Production Units | MODUs |
|----------------|------------------|-------|
| Supply Vessel  | 34%              | 60%   |
| Standby Vessel | 19%              | 11%   |
| Working Vessel | 34%              | 16%   |
| Rig            | 7%               | 6%    |
| Shuttle Tanker | 3%               | 1%    |
| Other          | 3%               | 5%    |
| Unknown        | 0%               | 1%    |

Figure 5. Installed OWT Sub-Structures , Collision Frequency with correspond Vessel Type [14], [13]

Although every approach had been taken to prevent the collision in the maritime industries by improving all the navigational technologies and create the certification from the Classification Societies for people, the collision incident still happened and it will have a really wide consequences as shown in Figure 6 below,

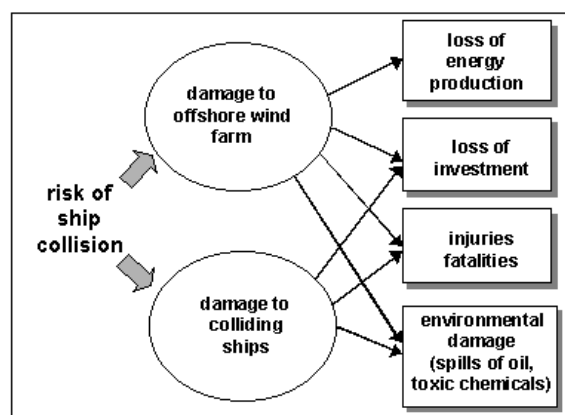


Figure 6. Classification of the ship's collision risks

As we know that when a collision happened, both of the structures and ship will be damaged. But the most important question would be, which one will have a bigger damage? As we can see from the Figure 6 above that the general consequences would be loss of investment, fatalities, loss of energy production, and environmental damage. Each damages would be depend on many factors such as, wind turbine capacity, how big the ship is, how many crew on board, wind, substances inside the ship, etc. For example, a collision between supply vessel and the offshore wind turbine structures in an extreme case will cost the company hundreds of millions Euro as loss of property and energy production. Moreover, in a collision between a oil tanker and offshore structures will create a large degradation of marine ecosystem due to oil spill on the sea like Exxon-Valdes incident. The propagation rate of the impact area will mostly depend by the substances itself and the environmental conditions like wind and current. Therefore, as the consequences of a collision may be very catastrophic, the research had been actively conducted in order to minimize the collision risk and damage.

There are several ways to analyzing the risk and damage from the collision incident such as : Formal Safety Assessment, Failure Mode and Effects Analysis, Hazard and Operability Study, What-If, and Fault Tree Analysis. Also, some pioneer in marine related industries had developed some collision models in order to determine the probability of the collision on sea. For example , COLLIDE (Safety Nordic AS), SOCRA (MARIN), SAMSON (MARIN), CRASH (DNV), COLWT (GL), COLLRISK (Anatec UK Ltd), DYMITRI (British Maritime Technology), and SSPA (Ellis et al, 2008). The SSPA methods is developed as the needs from the industry due to a higher sea traffic around the Offshore Wind Farms. This topic had been explained briefly by the previous work from Andres Barrera in 2013 [7].

In the other hand, the following research is an effort to developed an tool that could help people to answer the underlying question of collision, which one is better to experienced a bigger damage? The problem is there are too many collision scenario that should be analyzed since the results will be different depending on the ship collision's velocity, location of impact, and type of structures. It will take a really long time to compute all this scenario by a present Finite Element Approach, while it is possible to developed an analytical tool that may compute such things in an efficient manner. Finally, a deeper analysis could be performed in order to develop the Classification Societies generate a new guidance to built the offshore structures or ship that satisfy the conclusion of this research.

## STATE OF THE ART

As the offshore activities has been increasing in the last decade, a deeper analysis regarding the collision analysis is needed in order to evaluate the collision accident that always has a negative impact afterwards. In engineering point of view, there are two ways to evaluate this problem, these method shall be explained in this part.

The characterization of the design regarding the collision phenomenon had been introduced based on the kinetic energy which basically governed by the ship's mass including the added mass and the impact velocity. Then, this energy should dissipate into another form as the energy conservation law states. This dissipation could be explained briefly in the Figure 7.

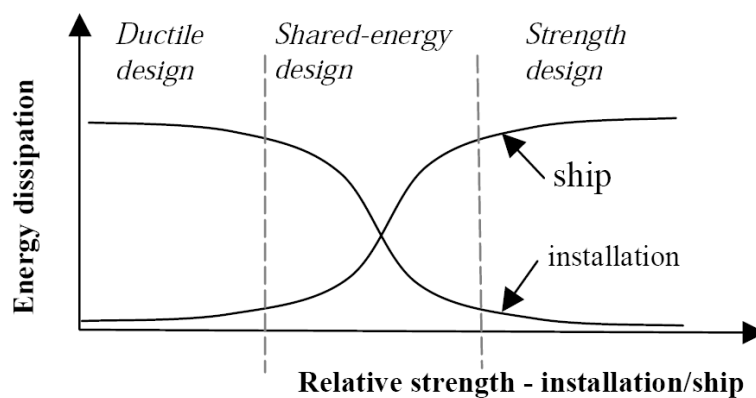


Figure 7. Concept of Design Against Collision [20]

There are three types of the design which commonly known as Ductile Design, Shared-Energy Design, and Strength Design. The strength design means that the offshore structures is strong enough to handle the impact thus the ship will have more dissipated energy. In the other hand, the ductile design means the opposite way that the ship has a stronger structures that leads to less energy dissipation. While the combination of both design shall be called as Shared-Energy Design.

In order to assess where this energy dissipation flows, P.Visser had established the resistance components of the fixed platforms into several types :

- Local Denting
- Elastic Beam Bending
- Plastic Bending / Hinge Formation
- Plastic Tensile Strain
- Global Deflection of the Installation
- Local Deformation of The Ship

By knowing the classification of the impact energy affecting the structures, the evaluation of the ship collision problems shall be conducted by using either the numerical method or an simplified analytical tool based on certain consideration such as accuracy and computation time.

Generally, the collision study will consist of several scenario with different parameters such as impact velocity and impact location. If the study conducted by using only numerical approach, then it will takes a longer time which is becomes the biggest problem to overcome the engineering problem. Therefore, there are many research about using the simplified analytical tool in order to compute many collision scenario in an efficient time with acceptable results. A. Bela et al [5] conducted some research about sensitivity analysis of an impacted monopile offshore wind turbine structures with different parameters such as impact velocity, wall thickness, gravity loads, wind, impact point, and soil stiffness. This analysis will become a basis for future analytical developments in order to estimate the crushing resistance and external dynamics of impacted monopile structures. The similar study also being conducted by Le Sourne et al [6] on offshore wind turbine jackets to establish the hypothesis to develop an analytical tool based on a super-element approach, this paper had concluded that the assumption of rigid striking ship is acceptable.

## NUMERICAL METHOD

This method is commonly known as the Finite-Element Method which is usually being used to solve any structural problems which in this case to evaluate the collision event. As the rapid development of the computing technology in the past decade, nowadays there are many FEM solvers provided such as Abaqus, Ansys, LS-DYNA, etc. In this study, LS-DYNA solver shall be used because the majority of the research conducted in this field of analysis use it.

In general, the equation of motion can be written with under following form,

$$[M + M_a]\ddot{u} + C\dot{u} + f_{int}(u) = p(t)$$

Where  $u$  represent the displacement of the body,  $M + M_a$  represent the structural and added mass matrices, and  $C$  is the damping coefficient. The right term represent the external forces that might occur on the reality. This equation shall be integrated using the explicit central difference scheme since the collision problem considered as a non-linear phenomenon as the internal force function ( $f_{int}$ ) varies according to the displacement leading to create a nonlinear ordinary differential equations.

### The Central Difference Method

The semi-discrete equations of motion at time  $n$  are:

$$\mathbf{M}\mathbf{a}^n = \mathbf{P}^n - \mathbf{F}^n + \mathbf{H}^n$$

where  $\mathbf{M}$  is the diagonal mass matrix,  $\mathbf{P}^n$  accounts for external and body force loads,  $\mathbf{F}^n$  is

the stress divergence vector, and  $\mathbf{H}n$  is the hourglass resistance. To advance to time  $tn+1$ , we use central difference time integration:

$$\begin{aligned}\mathbf{a}^n &= \mathbf{M}^{-1}(\mathbf{P}^n - \mathbf{F}^n + \mathbf{H}^n) \\ \mathbf{v}^{n+1/2} &= \mathbf{v}^{n-1/2} + \mathbf{a}^n \Delta t^n \\ \mathbf{u}^{n+1} &= \mathbf{u}^n + \mathbf{v}^{n+1/2} \Delta t^{n+1/2}\end{aligned}$$

where

$$\Delta t^{n+1/2} = (\Delta t^n + \Delta t^{n+1}) / 2$$

and  $\mathbf{v}$  and  $\mathbf{u}$  are the global nodal velocity and displacement vectors, respectively. We update the geometry by adding the displacement increments to the initial geometry:

$$\mathbf{x}^{n+1} = \mathbf{x}^0 + \mathbf{u}^{n+1}$$

We have found that, although more storage is required to store the displacement vector the results are much less sensitive to round-off error.

#### Time Step Calculations for Beam and Truss Elements

For the Hughes-Liu beam and truss elements, the time step size is given by:

$$\Delta t_e = L / c'$$

where  $L$  is the length of the element and  $c$  is the wave speed:

$$c = \sqrt{E / \rho}$$

For the Belytschko beam the time step size given by the longitudinal sound speed is used unless the bending-related time step size given by [Belytschko and Tsay 1982]

$$\Delta t_e = 0.5L / c \sqrt{3I \left[ \frac{3}{12I + AL^2} + \frac{1}{AL^2} \right]}$$

is smaller, where  $I$  and  $A$  are the maximum value of the moment of inertia and area of the cross section, respectively.

Comparison of critical time steps of the truss versus the elastic solid element shows that if Poisson's ratio,  $\nu$  is nonzero the solid elements give a considerably smaller stable time step size. If we define the ratio,  $\alpha$ , as:

$$\alpha = \frac{\Delta t_{continuum}}{\Delta t_{rod}} = \frac{C_{rod}}{C_{continuum}} = \sqrt{\frac{(1 + \nu)(1 - 2\nu)}{1 - \nu}}$$

we obtain the results in Table below where we can see that as  $\nu$  approaches .5  $\alpha \rightarrow 0$ .

|          |    |       |       |       |       |       |     |
|----------|----|-------|-------|-------|-------|-------|-----|
| $\nu$    | 0  | 0.2   | 0.3   | 0.4   | 0.45  | 0.49  | 0.5 |
| $\alpha$ | 1. | 0.949 | 0.862 | 0.683 | 0.513 | 0.242 | 0.0 |

### Belytschko-Lin-Tsay Shell

The Belytschko-Lin-Tsay shell element ([Belytschko and Tsay 1981], [Belytschko et al., 1984a]) was implemented in LS-DYNA as a computationally efficient alternative to the Hughes-Liu shell element. For a shell element with five through thickness integration points, the Belytschko-Lin-Tsay shell elements requires 725 mathematical operations compared to 4050 operations for the under integrated Hughes-Liu element. The selectively reduced integration formulation of the explicit Hughes-Liu element requires 35,350 mathematical operations. Because of its computational efficiency, the Belytschko-Lin-Tsay shell element is usually the shell element formulation of choice. For this reason, it has become the default shell element formulation for explicit calculations.

The Belytschko-Lin-Tsay 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. The co-rotational portion of the formulation avoids the complexities of nonlinear mechanics by embedding a coordinate system in the element. The choice of velocity-strain or rate-of deformation in the formulation facilitates the constitutive evaluation, since the conjugate stress is the physical Cauchy stress.

The above explanation of the numerical method is compiled from the LS-DYNA Theory Manual 2015 [23] and 4<sup>th</sup> European LS-DYNA Users Conference [17] for the reference on afterwards topics. Further explanation shall be found inside the document in order to have a better understanding about how the software solve the problem numerically.

## **SIMPLIFIED ANALYTICAL METHOD**

As described by Pedersen in 1995 and Petersen 1982, the ship collision mechanics can be divided into inner collision mechanics and outer collision dynamics.

- The inner collision mechanics is the collision problem which governed by the buckling, yielding, and rupture of the material on every assemblies of the structure. The collision forces shall be calculated by using the explicit finite element code.
- The outer collision dynamics will use the MCOL program to be coupled with LS-DYNA to compute the global motions of the two rigid bodies subjected to collision forces and hydrodynamic pressure forces. Hydrodynamic loads are calculated with hydrostatic restoring matrix. While the added mass and damping matrices computed by using the 3D sea-keeping code AQUA+.

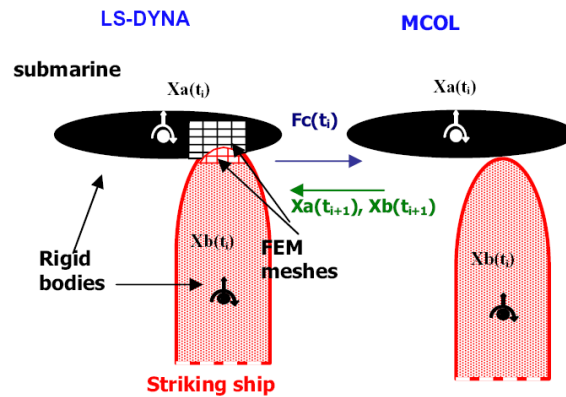


Figure 8. LS-DYNA MCOL Collision Simulation System [17]

As the model above, only the structures around the collision location are meshed while the remaining structure will be described by its inertia matrix and the center of mass position. The improved MCOL programs shows a good results agreement to taking into account the strong rolling movement which generated by the crushing force and surrounding water forces (added mass, wave damping, and restoring forces) as in Figure 9. Without this coupled, LS-DYNA only able to calculate the translational movement. This MCOL analytical tool shall be used to developed another tool known as SHARP which will coupled the internal and external mechanics of the ship collision to accommodate the reality that the striking and struck vessels may be damaged. Based on this knowledge and derivation, a new tool shall be developed to conduct the collision study between ships and offshore wind turbine jackets, it will be known as COLEOL.



Figure 9. LS-DYNA MCOL Simulation Results vs Reality

A theoretical basis is needed in order to create the mathematical model of the internal mechanics. The upper bound theorem shall be used which according to Jones[24], states that *"if the work rate of a system of applied loads during any kinematically admissible collapse of a beam is equated to the corresponding internal energy dissipation rate, then that system of loads will cause collapse, or incipient collapse, of the structure"*. Therefore the external energy rates shall be evaluated as below,

$$\dot{E}_{ext} = P \cdot \dot{\delta}$$

Where P is the lateral load applied on the cylinder and  $\delta$  is the penetration. This could be obtained by calculating the area under the curve of obtained load function (P-  $\delta$ ).

Then the internal energy rate for the cylinder body could be expressed as ,

$$\dot{E}_{int} = \iiint_V \sigma_{ij} \cdot \dot{\epsilon}_{ij} \cdot dV$$

Where V is the volume of solid body,  $\sigma_{ij}$  is stress tensor , and  $\dot{\epsilon}_{ij}$  is the strain rate tensor. Then some simplifications shall be applied in order to solved this analytically, these assumptions are :

- The rigid plastic material which means that the  $\sigma_0$  (flow stress) is equal to the yield stress  $\sigma_y$ . This will neglecting the elastic part of deformation and the strain rate effect.
- The internal energy derivation which will be based on super element method [4]

As already mentioned before that the analytical tool used here will be based on the super-elements method as had been further studied by Loïc Buldgen et al [4]. The general principles of this method is that dividing the full-structure into several sub-structures and then generate the mathematical model to represent the interaction of it. Therefore in this particular study, the jackets shall be divided into three super-elements which are impact on the vertical cylinder, impact on the horizontal cylinder, and impact on the oblique cylinder as shown,

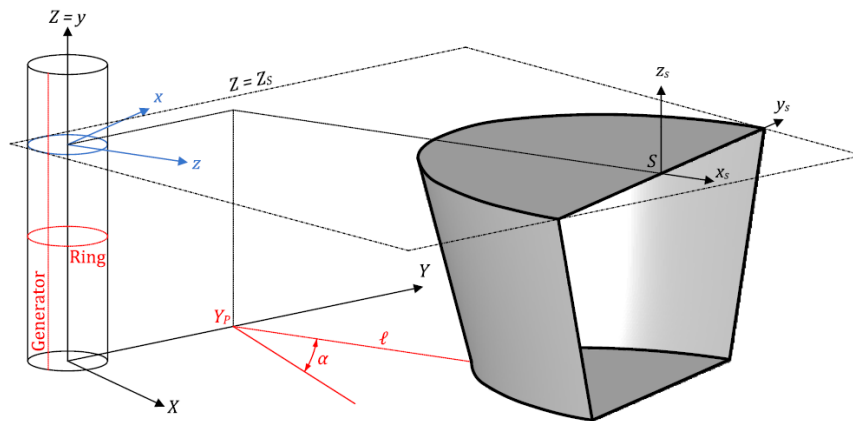


Figure 10. Impact on Vertical Cylinder Model[4]

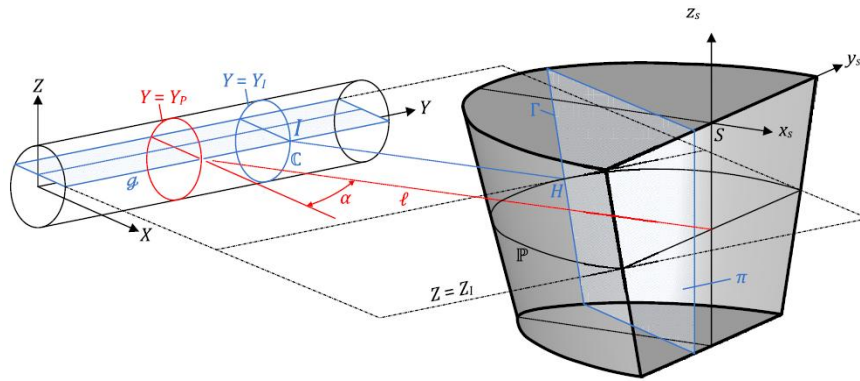


Figure 11. Impact on Horizontal Cylinder Model[4]

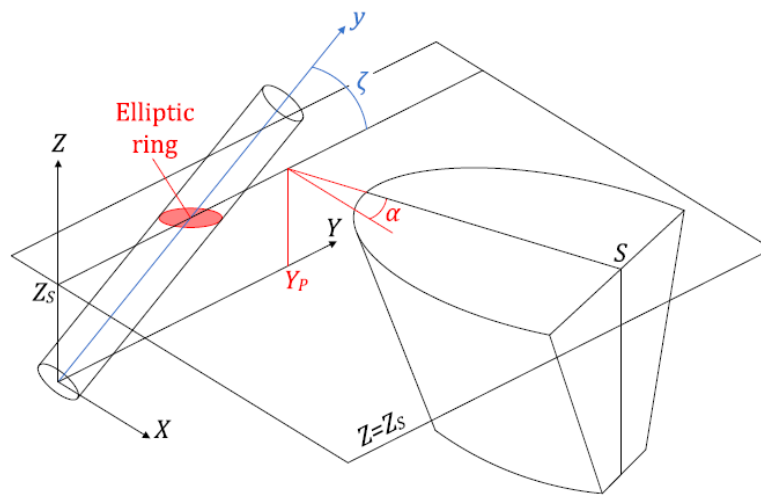
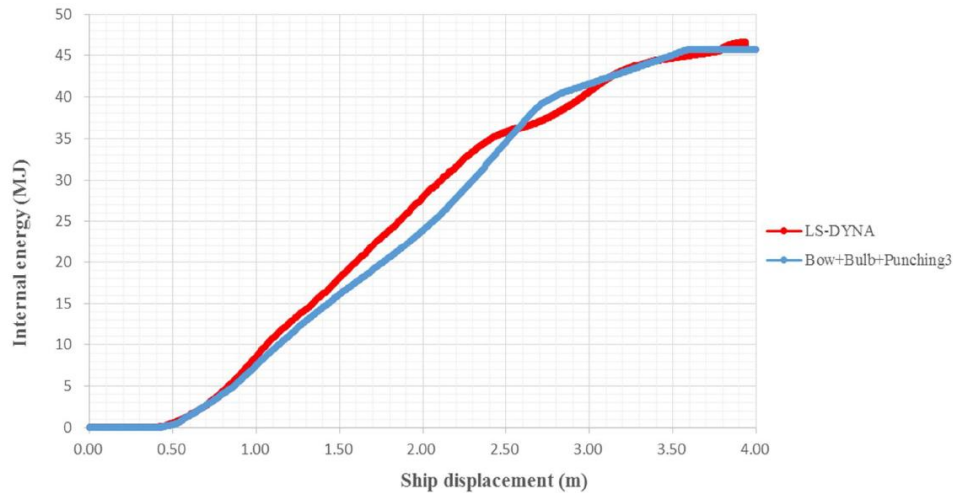
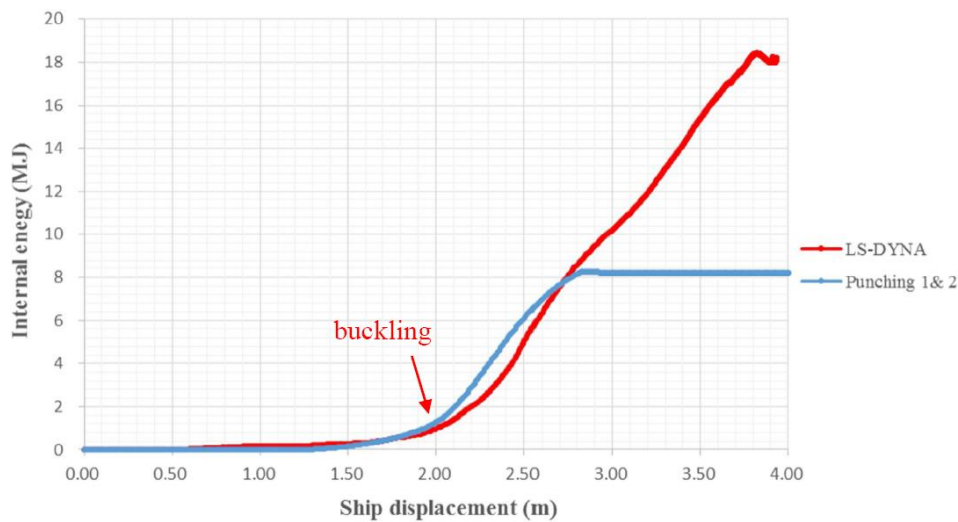


Figure 12. Impact on Oblique Cylinder Model[4]

In order to obtain the local and global crushing resistance of each super-element, the deformation mechanism and displacement field should be evaluated first. But for the last super-element on oblique cylinder, it is difficult to derive the deformation and displacement field during the collision process, the main reason is the fact that the rings have an elliptic shape which make it harder to be imagined how the deformation looks like. Therefore, the corresponding crushing resistance shall be evaluated by using a linear interpolation between the other solutions. Moreover, further development of above study had been conducted by Jing-Ru Hsieh [9] in order to taking into account the punching and buckling phenomenon. The results from this development could be seen in the Figures 13 and 14 below,



**Figure 13. Absorbed Energy Comparison on the Impacted Leg of OWT Jackets**



**Figure 14. Absorbed Energy Comparison on the Rear Leg of OWT Jackets**

From this adaptation, it has been concluded that the absorbed energies predicted by the super-elements for impacted legs has a good agreement with LS-DYNA Simulation, while for the rear leg has some overestimation by super-elements methods due to the buckling of braces.

In the other hand, there are two points that could be improved from this latest study. First, is the implementation of relative displacement between brace/leg joints in order to have a better prediction of absorbed energy in super-element method. Then, since the absorbed energy from collided leg and rear leg is around 80% of overall energy, it is necessary to adapt the super-element method with another phenomena such as buckling of braces, shearing near the mud-line, and any other patterns of punching. Generally, this study has a really close relation between the conducted research paper as mentioned all above. Therefore, the corresponding paper shall be studied in order to understand all the derivation of necessary equation.

## **RULES REGARDING COLLISION ANALYSIS**

The possible objective of this study is to updating the rules regarding the collision based on the simplified analytical tool as explained before. Thus, the world leading classification societies shall be selected to analyze the present rules with the results from the analytical tool. These classification are BV, DNV-GL, and LR as will be explained respectively.

### **BV (Bureau Veritas)**

#### **Scope**

Due to operations around the unit collisions can happen. The scope of the following requirements is to provide rules for the verification of the structure safety with respect to such events. The requirements are eligible for the verification of the resistance of side shell structure submitted to a collision from a vessel which dimensions are small compared to the dimensions of the unit.

#### **Safety principle**

The energy of the colliding vessel is to be absorbed by the deformed side shell of the unit without risk of flooding. Therefore the safety criterion is that the structural elements may suffer permanent deformations without any rupture.

#### **Calculations to be submitted**

The following documents are to be submitted to the Society for information:

- list of shuttle tankers and supply vessels intended to be operated during the unit life and selected vessels
- colliding speeds and justification
- colliding energy calculation of the selected colliding vessels
- side shell energy deformations versus bow indentations until first shell plate rupture.

#### **Risk analysis**

A risk analysis is to be performed taking into account the boats operating around the unit, such as supply boats, shuttle tanker, etc. The risk analysis is to determine, for each vessel operating around the unit, the speed, the mass and the associated probability of collision.

#### **Assessment conditions**

##### **Areas to be considered**

The following areas of the unit are to be considered:

- bow and free side shell between two transverse bulkheads
- free side shell at the level of the first transverse ring from a transverse bulkhead.

In addition, the following areas are to be considered, when relevant:

- side shell at spread mooring zones
- side shell at the offloading line protection zones
- side shell at the riser zones
- side shell at the turret.

### Collision characteristics

The colliding vessel is considered hitting the side shell by the bow at 90°. The colliding vessel sizes and bow shapes (with or without bulb) to be considered are selected from the list of shuttle tankers and supply vessels intended to be operated during the unit life. The colliding vessel speed is to be fixed considering the operation instructions. Without any informations, the speed is to be taken equal to 10 knots.

### Colliding energy

The colliding vessel energy  $E_c$ , in kJ, is to be taken equal to:

$$E_c = \frac{1}{2} \frac{M_a M_b}{M_a + M_b} V^2$$

$M_a, M_b$  : Unit and colliding vessel mass, in tons, including the added water mass

$V$  : Speed of the colliding ship, in m/s.

### Criteria

The considered limit state is the first rupture of a plate in the indented (crushed) area of the unit.

### Methodology

#### Bow behavior

The bow of the colliding ship is considered non deformable. Therefore the geometrical contour of the bow defines the indented area of the unit.

#### Finite element calculation

Finite element calculations are to be performed step by step of indentation until the limit state is reached, using a non linear elasto-plastic recognized software.

#### Analytical methods

As an alternative, analytical methods, such as Mc Dermott and Rosenblatt methods can be applied.

### Results

The results to be provided are:

- the absorbed deformation energy versus indentation steps, until the limit state is reached
- the deformed structure at the limit state
- the list of the cracked plates.

**DNV-GL (Det Norske Veritas-Germanischer Lloyd)****General**

Collision by a supply vessel against a leg of a self-elevating unit shall be considered for all elements that may be hit either by sideways, bow or stern collision. The vertical extent of the collision zone shall be based on the depth and draught of visiting supply vessels.

A collision will normally only cause local damage of the leg. However, the global strength and overturning stability of the unit shall also be checked. With lattice type legs the damaged chord or bracing and connections may be assumed to be non-effective for check of residual strength of the unit after collision. Assessment of dynamic effects and non-linear structural response (geometrical and material) should be performed as part of the impact evaluation.

**Ship impacts and collisions**

Boat landings, ladders and other secondary structures in and near the water line shall be designed against operational ship impacts in the ULS. The primary structure in and near the water line shall be designed against accidental ship impacts in the ALS. Furthermore, if an accidental ship impact against a secondary structure results in larger damage of the primary structure than an accidental impact directly against the primary structure, then this load case shall also be considered in the ALS.

When primary structural parts such as the support structure and its foundation are exposed to ship impacts, these structural parts shall not suffer such damage that their capacities to withstand the loads they are going to be exposed to become compromised. In the ULS, secondary structural parts, such as fenders, boat landings and ladders, shall not suffer damage to such an extent that they lose their respective functions as access structures. In the ALS, secondary structural parts are allowed to become torn off, e.g. by including weak points or by local strengthening of supporting structural parts, thereby to avoid excessive damage to these supporting primary structural parts.

For design against operational ship impacts, the characteristic impact load shall be taken as the expected impact load caused by the maximum authorized service vessel approaching by bow and stern in the most severe sea state to be considered for operation of the service vessel. A vessel-specific speed shall be assumed. The speed shall not be assumed less than 0.5 m/s. Effects of wind, wave and current shall be included as well as effects of added mass, which contributes to the kinetic energy of the vessel.

The expected impact load is part of the results from the risk analysis. When specific loads are not given, the contact area can be designed by assuming an impact force

$$F = 2.5 \times D$$

where  $F$  is the impact force in units of kN and  $D$  is the fully loaded displacement of the supply vessel in units of tons. This is based on an assumption of ship impact against a hard structure. When a damper or spring device such as a fender is provided in the area subject to the impact, a lower impact force can be used. For further background and guidance, reference is made to the DNV High Speed Light Craft Rules. IEC61400-3 states that if no information about the service vessel is known, the impact force can normally be accounted for by applying 5 MN as a horizontal line load over the width of the support structure. This load is meant to include dynamic amplification.

### **Bow Strength when pushing against or connecting with other structures**

When arranged for such, the area of the bow intended to be pushed against another structure shall be strengthened to withstand the applied loads. The design contact area(s) shall be shown on the drawing of the bow structure. The contact area shall be considered to be the part of the bow that makes regular contact with another structure (e.g. a wind turbine tower boat landing structure), including any load spreading effects from the fender system. The contact area shall be dimensioned for a horizontal force  $F_H$ , a vertical force  $F_V$  and a side force  $F_S$ . The horizontal force  $F_H$  shall be taken as the static bollard push as stated by the designer. The vertical force,  $F_V$ , shall be taken as:

$$F_V = \mu F_{BP} \text{ (kN)}$$

$\mu$  is the coefficient of friction acting between the fender and the connection, unless otherwise stated, this shall be taken 1.0. The side force,  $F_S$ , shall not be less than 30 kN. A combined load case shall also be considered where the horizontal force,  $F_H$ , vertical force,  $F_V$ , and side force,  $F_S$ , are applied to the contact area simultaneously. The allowable stress shall be taken from the relevant chapters for plate, stiffeners or girders using allowable stresses as for ships side subjected to sea loads. In addition, the bow structure shall be designed for an accidental collision. The collision load shall be taken as the worst combination of  $F_{RD}$ ,  $F_V$  and  $F_S$ , where  $F_{RD}$  is the horizontal force caused due to a rapid deceleration:

$$F_{RD} = (\Delta + A_m) a_{coll} \text{ (kN)}$$

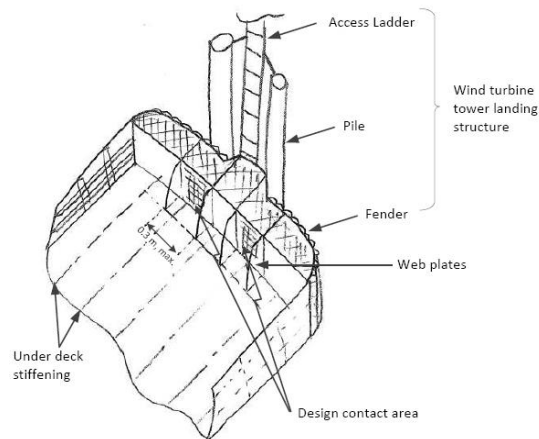
$\Delta$  is the displacement of the craft at the design water line (tonnes)

$A_m$  is the surge added mass (tonnes). Unless otherwise stated, this shall be taken as  $0.2 \times \Delta$

$a_{coll}$  is the collision deceleration ( $\text{m/s}^2$ ). Unless otherwise stated, this shall be taken as  $0.25g_0$ .

The contact area shall be supported at intervals not exceeding 0.3 metres by web plates or girders aligned with deck longitudinal stiffeners or other similar structure. As a minimum,

one web plate or girder shall be fitted on either side of the contact area, see Figure below. Unsupported plating in the contact area shall be a minimum of 12 mm unless buckling stiffeners are fitted. A lower plate thickness may be accepted if supported by buckling calculations.



For design against accidental ship impacts, the characteristic impact load shall be taken as the impact load caused by unintended collision by the maximum authorized service vessel in daily operation. For this purpose, the service vessel shall be assumed to be drifting laterally and the speed of the drifting vessel shall be assessed. The speed shall not be assumed less than 2.0 m/s. Effects of added mass shall be included. Effects of fendering on the maximum authorized service vessel shall be considered.

### **Accidental strength (ALS)**

#### **General**

In the safety evaluation of a platform concept, accidental loads should be identified and taken into consideration. For impact types of accidental load such considerations lead to recommendation of structures of a certain robustness or ductility or redundancy. Requirements for Accidental Limit States (ALS) and structural redundancy of slender main load bearing structural elements are given in DNV-OS-C101 Sec.6.

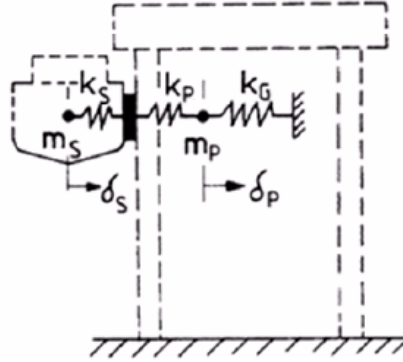
The damaged condition may be divided in two main groups:

- Structural redundancy comprising fracture of bracing, chord or joint for legs
- The unit shall be designed for environmental loads with return period not less than 1 year after damage, see DNV-OS-C101 Sec.3 B100.

#### **Ship impact**

The present section describes only the impact from a ship collision. Such an impact load is most likely to cause local damage to one of the legs only, but the possibility for progressive

collapse and overturning should also be considered. The overall lateral stiffness of a jack-up platform is rather small, normally an order of magnitude smaller than the stiffness of a corresponding jacket structure. Because a ship collision is a hypothetical event, it is allowable to use very simplified methods of analysis. A simple two-degree-of-freedom system may normally be considered appropriate, see Figure below.



Under the assumption that  $k_L \gg k_G$  it is found that the maximum interaction force between ship and platform is:

$$P_L = k_L(\delta_s - \delta_p) = v \sqrt{\frac{m_s k_l}{1 + \frac{m_s}{m_p}}}$$

The maximum global load effect is:

$$P_G = k_G \delta_p = v \sqrt{\frac{m_s k_g}{1 + \frac{m_p}{m_s}}}$$

where :

$$k_l = \frac{1}{\frac{1}{k_s} + \frac{1}{k_L}} = \text{combined local stiffness of ship and leg}$$

$k_s$  = stiffness of ship.

$m_p$  = mass of platform.

$k_p$  = local stiffness of platform leg.

$m_s$  = mass of ship including added mass.

$k_G$  = overall lateral stiffness of platform.

$v$  = ship velocity before collision.

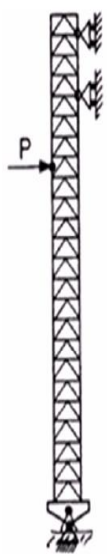
The values of  $k_G$  and  $m_p$  are to be evaluated with due consideration of ship/platform configuration. If the platform is hit in such a way that the resulting platform displacement is a pure translation (no rotation) then:

$$k_G = n k_e$$

$$m_p = n m_e$$

where  $n$  is the number of legs and  $k_e$  and  $m_e$  are defined in 4.4.3.

The value of  $k_p$  may generally be determined as shown in below Figure. However, if stresses in chords or bracings are found to exceed the elastic limit before the lateral load reaches the maximum value, PL, the calculations should be based on non-linear methods. Such methods should account for formation of plastic hinges and buckling of compression members. In addition the strength of tubular joints has to be checked according to the methods described in 6.3. If ductile energy absorption shall be possible, it is necessary that the joints do not fail due to excessive punching shear stresses.



Lateral load:  $P$

Lateral displacement:  $\delta$ .

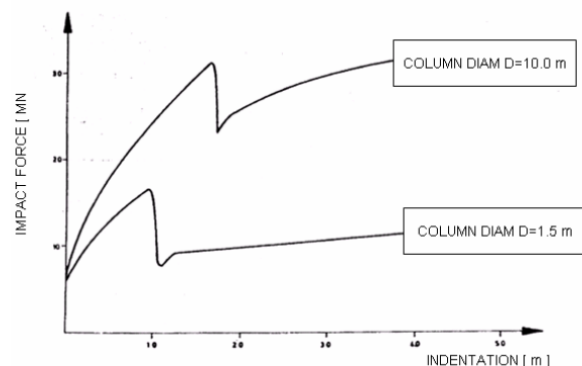
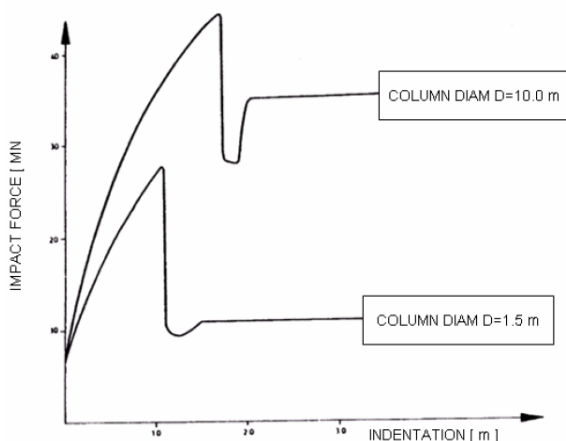
Local stiffness:  $k_p = P / \delta$

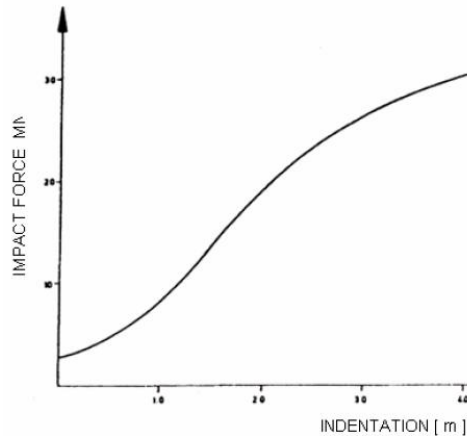
Displacement contributions:

- Shear and bending deflection of leg
- Bending of chord
- Compression of bracings
- Deformation of chord wall.

The value of  $k_s$  represents the local stiffness of the ship. Because the ship plating is relatively thin, the deformation of the ship will always be inelastic. Typical force

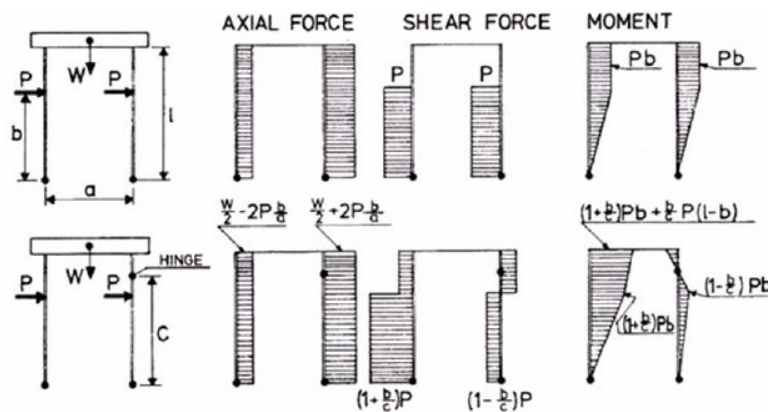
indentation curves are shown in three Figures below. For the purpose of these calculations the selected value of  $k_s$  should be an average value for the actual range of displacements. The value of PL will in the first place be decisive for the extent of local damage, while PG will be decisive for the risk of overturning.





### Damaged structure

For a jack-up leg of sufficient robustness a credible accidental load as described above will normally not lead to any significant damage. However, for slender lattice legs it may be found that one chord element is damaged and made inefficient. According to the Rules, a platform with a credible damage shall survive under environmental conditions corresponding to a return period of 1 year. If one chord element in a triangular lattice leg, see Figure 4-1, is removed, a hinge will be formed. The leg will then be unable to transfer horizontal loads to the sea bottom, but it may still be able to carry axial loads. This means that horizontal loads have to be redistributed to the intact legs, which will then be subjected to increased loading. However, the environmental loads corresponding to a 1 year return period are significantly smaller than those corresponding to a 100 year return period. It is also possible to allow higher stresses in a damage condition than in a normal intact operating condition. A simplified two-dimensional model is used in order to illustrate the residual strength of a jack-up platform after the formation of a hinge in one of the legs, see below Figure.



**LR (Lloyd's Register)**

Collision loads imposed by attending vessels which may be approaching, mooring or lying alongside the unit are to be considered in the design. The unit is to be designed to withstand accidental impacts between attending vessels and the unit and be capable of absorbing the impact energy.

The kinetic energy to be considered is normally not to be less than:

- 14 MJ for sideways collision;
- 11 MJ for bow or stern collision; corresponding to an attending vessel of 5000 tonnes displacement with impact velocity 2 m/s. A reduced impact energy may be accepted upon special consideration, taking into account the environmental design criteria.

The energy absorbed by the unit during a collision impact will be less than or equal to the total impact kinetic energy, depending on the relative stiffnesses of the relevant parts of the unit and the impacting ship/unit and also on the mode of collision and ship/unit operation. These factors may be taken into account when considering the energy absorbed by the unit, *see also* Ch 4,1 and Ch 4,3 for column-stabilised and self-elevating units respectively.

Collision is to be considered for all elements of the unit which may be hit by sideways, bow or stern collision. The vertical extent of the collision zone is to be based on the depth and draught of attending ships/units and the relative motion between the attending ships/units and the unit.

**Collision scenarios**

Supply vessels arriving to the field normally have three modes of collision with the platform:

- On arrival
- Waiting in standby position
- During loading (manoeuvring and drifting)

Supply on arrival

The vessel is heading for a way point set some nautical miles away from the platform. A collision model based on historical accident frequency is presented below:

Collision frequency :  $f_c = N \times f$

Collision energy :  $E = \frac{1}{2} \times m \times (1+a) \times v^2$  (k Joule)

where :

$f_c$  = collision frequency (per year)

$v$  = powered velocity of vessel (m/s)

$N$  = number of arrivals per year (visits per year)

$m$  = mass of vessel (ton)

$a$  = added mass factor ( $a = 0.1$ )

$f$  = collision frequency (per visit)

*Collision frequency per visit (f)*

The collision frequency will likely depend on the installation and the specific location the supply vessel is visiting. However, it is reasonable to use a generic historical collision frequency per visit. When using a generic frequency (which is e.g. based on world-wide data) the risk analysis should include assessments/discussions regarding adjustment of the frequency to the area subject to analysis. Relevant sources for accident data are:

- Petroleum Safety Authority Norway, [www.psa.no](http://www.psa.no)
- Health and Safety Executive UK, [www.hse.gov.uk](http://www.hse.gov.uk)
- IHS Fairplay, [www.ihs.com](http://www.ihs.com)
- Norwegian Maritime Authority, [www.sjofartsdir.no](http://www.sjofartsdir.no)
- The International Association of Oil and Gas Producers, [www.ogp.org.uk](http://www.ogp.org.uk)

*Powered velocity of vessel (v)*

To be able to assess the consequences of powered supply vessel collision during approach, it is necessary to include velocity of vessel. Normally these types of collisions are split into low speed collisions and high speed collisions. There are few sources available describing the velocity in historical events. The Petroleum Safety Authority Norway is one known source that has velocity included in the event description [14]. A report published by the Health and Safety Executive UK [15] does not include absolute velocity Figures but rather categorizes the events with consequence classes which may be used to assess the velocity at impact.

*Mass of vessel (ton)*

The mass of the relevant supply vessel in question should be applied (in displacement tonnes). It should be noted that in the previous years the supply vessels on the NCS and UKCS have increased in size and it is now normal with sizes around 8000 tons.

*Added mass factor (a)*

An added mass factor for water displacement of  $a = 0.1$  is applied for collision on arrival since this is most likely a head on collision.

Supply vessel in standby position

After arrival at the field, when the supply vessel joins the standby vessel in standby position waiting for admittance to approach the platform. The supply vessel normally performs DP preparations in this position. A possible collision scenario may be that the vessel has a total blackout and drifts towards the platform. A collision model based on historical accident frequency is presented below:

Collision frequency :  $f_c = N \times t \times f_{\text{Drift}} \times P$

Collision energy :  $E = \frac{1}{2} \times m \times (1+a) \times v^2$  (k Joule)

where;

$f_c$  = Drift collision frequency (per year)  $v$  = velocity of vessel (m/s)

$N$  = number of visits per year  $m$  = mass of vessel (ton)

$t$  = Duration in standby position (hour)  $a$  = added mass factor ( $a = 0.4$ )

$f_{\text{Drift}}$  = Frequency of total blackout and drift-off during standby (per hour)

$P$  = Probability of drifting towards installation and hit, given drift-off during standby

*Frequency of drift-off during standby ( $f_{\text{Drift}}$ )*

The frequency for a modern supply vessel should be derived from historical failure data, e.g. loss of position data provided by International Marine Contractors Association (IMCA, [www.imca-int.com](http://www.imca-int.com)).

*Probability of drifting towards installation ( $P$ )*

Probability of drifting towards installation and hit given drift-off during standby,  $p$  should be estimated specifically for the installation in question, e.g.:

$f = (D+L) / (\pi \times R) \times (1 - \text{Pleeward})$

where;

$D$  = Platform diameter (m)  $R$  = Distance from vessel to platform (m)

$L$  = Length of vessel (m)

$\text{Pleeward}$  = Probability for vessel being positioned at leeward side

The equation  $(D+L) / (\pi \times R)$  finds the probability for the vessel's location along half the circumference (i.e. half 500 m zone), while  $(1 - \text{Pleeward})$  corrects for the probability of actually drifting towards the installation. If one does not correct for  $(1 - \text{Pleeward})$ , then the whole circumference should be included  $(2 \times \pi \times R)$ .  $\text{Pleeward}$  is estimated by considering wind and wave directions, probabilities for using deck cranes, operational strategy for supply vessel approach etc. It is normal to assume that the vessel approaches the installation and waits in standby position at the point closest to the onshore supply base.

*Velocity of vessel ( $v$ )*

Data from simulations or historical data is preferable. However, the velocity at impact is normally assumed to be low, i.e. below 3 knots.

*Added mass factor ( $a$ )*, An added mass factor for water displacement of  $a = 0.4$  is applied for drifting collision, since this is most likely a sideways collision.

Supply vessel during on/offloading – drive-offCollision frequency :  $f_c = N \times t \times f_{\text{Drive}} \times P$ Collision energy :  $E = \frac{1}{2} \times m \times (1+a) \times v^2$  (k Joule)

where;

$f_c$  = Powered collision frequency (per year)       $P$  = Probability of hit given drive-off  
 $v$  = velocity of vessel (m/s)

 $N$  = number of visits per year $m$  = mass of vessel (ton) $t$  = Duration in standby position (hour) $a$  = added mass factor ( $a = 0.1$ ) $f_{\text{Drive}}$  = Frequency of drive-off during on/offloading (per hour)*Frequency of drive-off during on/offloading ( $f_{\text{Drive}}$ )*

The frequency for a modern supply vessel should be derived from historical failure data, e.g. loss of position data provided by International Marine Contractors Association (IMCA, [www.imca-int.com](http://www.imca-int.com)).

*Probability of driving towards installation ( $P$ )*

Probability of collision,  $P$ , needs to be estimated based on the collision target in question. The relevant target may be columns, riser balcony, living quarter etc.

*Velocity of vessel ( $v$ )*

Data from simulations or historical data is preferable. However, the velocity at impact is normally assumed to be low.

*Added mass factor ( $a$ )*

An added mass factor for water displacement of  $a = 0.1$  is applied for drive-off collision during on/offloading since this is most likely a bow/stern collision.

Supply during on/offloading – drift-offCollision frequency:  $f_c = N \times t \times f_{\text{Drift}} \times P$ Collision energy:  $E = \frac{1}{2} \times m \times (1+a) \times v^2$  (k Joule)

where;

$f_c$  = Drift collision frequency (per year)       $v$  = velocity of vessel (m/s)

 $N$  = number of visits per year $m$  = mass of vessel (ton) $t$  = Duration in standby position (hour) $a$  = added mass factor ( $a = 0.4$ )

on/offloading

 $f_{\text{Drift}}$  = Frequency of total blackout and drift-off during standby on/offloading (per hour) $P$  = Probability of drifting towards installation and hit given drift-off during standby

*Frequency of drift-off during standby ( $f_{\text{Drift}}$ )*

The frequency for a modern supply vessel should be derived from historical failure data, e.g. loss of position data provided by International Marine Contractors Association (IMCA, [www.imca-int.com](http://www.imca-int.com)).

*Probability of drifting towards installation ( $P$ )*

Probability of drifting towards installation and hit given drift-off during loading/unloading,  $f$ , should be estimated specifically for the installation in question. Probability of loading on leeward side, wind direction and usage of cranes should be taken into consideration.

*Velocity of vessel ( $v$ )*

Data from simulations or historical data is preferable. However, the velocity at impact is normally assumed to be low.

*Added mass factor ( $a$ )*

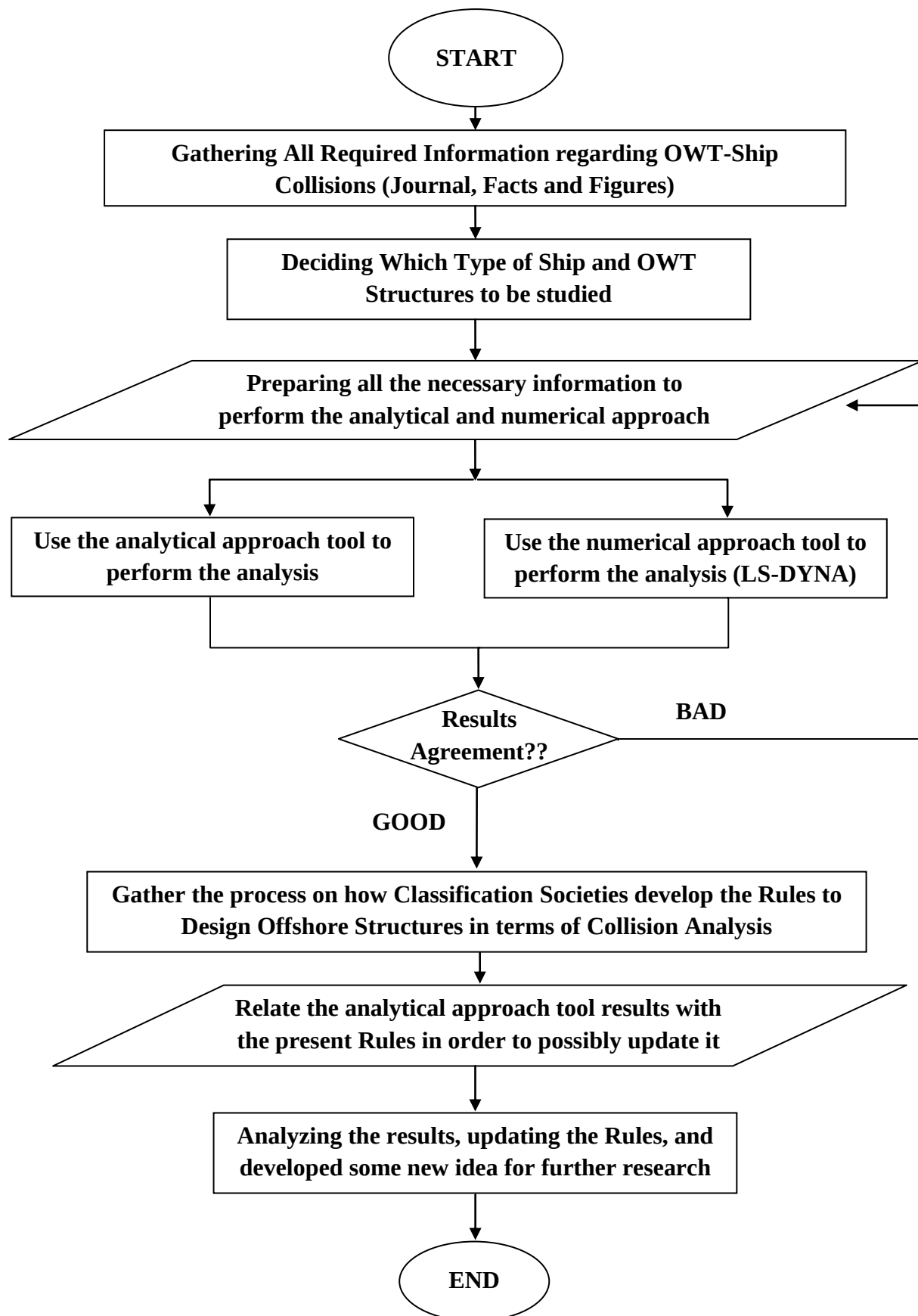
An added mass factor for water displacement of  $a = 0.4$  is applied for drift collision since this is most likely a sideways collision.

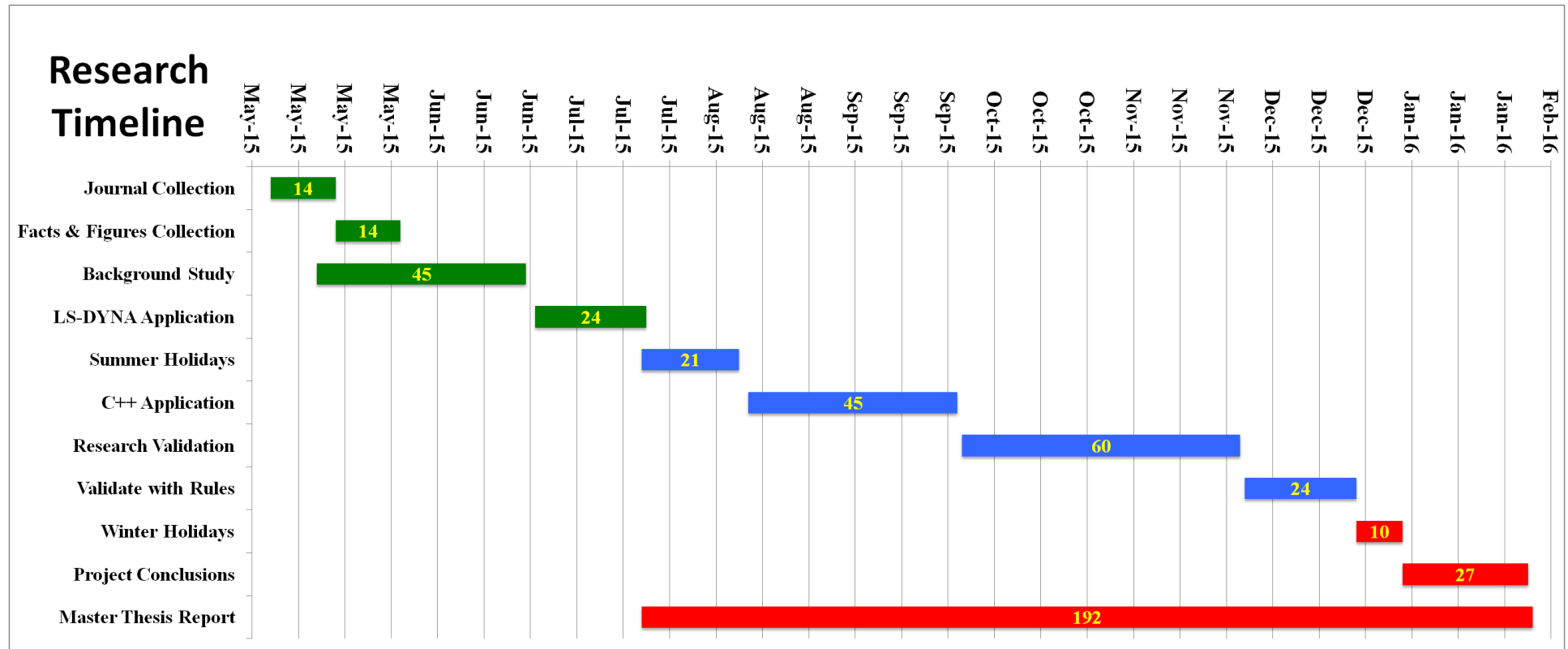
Visiting vessel collisions – standby vessels

If present on the field, a standby vessel is circulating around the platform. A standby vessel normally stays outside 500m zone of the platform. A possible collision scenario may be that the standby vessel has a total blackout and starts drifting towards the platform. However, the standby vessel should normally be positioned on the leeward side of the installation (outside 500m) and her potential drifting path should be away from any installations in the field. Therefore, a drifting standby vessel should not drift towards the installation. If there is a possibility for not being positioned on the leeward side, a similar methodology as for “supply vessel in standby position” may be applied.

All the stated rules above are compiled from its rules and guidelines standards as mentioned in the reference page, [10], [11], and [12].

## RESEARCH FLOWCHART AND TIMELINE





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