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MASTER'S THESIS

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ABSTRACT

Technical solutions for subsea field developments have seen a rapid growth in complexity over the last years. Today the oil & gas industry is getting ready for a step change by implementing subsea processing facilities. Implementing the equivalent of a topside processing facility on the seafloor brings several challenges, one being the inspection, maintenance and repair (IMR).

The main objective of this thesis is to define the challenges related to IMR on a subsea processing facility and to evaluate which vessels are able perform the required IMR operations such a facility would require. The evaluation will be based on the components of the Åsgard subsea compression station. To evaluate vessels ability to perform the IMR operations the data simulation program ORCAFLEX is used. The most critical part of the operation is the splash zone crossing phase so the analysis is based around this. The results from ORCAFLEX are also combined with the weather criteria found at the location.

The result of the analysis shows that the operations the vessels can perform varies greatly depending on what module is lifted with what vessel and that some modules cannot be lifted in high sea states no matter what vessel is used. The analysis also indicates that the largest vessel used in the simulation may be over dimensioned and that a smaller vessel could be used for the same operations. If medium sized vessels are able to the IMR work for the subsea processing facility, the existing IMR vessel fleet can be put to use to prevent unnecessary expenditures.

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ABBREVIATIONS

AHC	–	Active Heave Compensation
CBM	–	Condition Based Monitoring
CFD	–	Computerized Fluid Dynamics
CM	–	Condition Monitoring
CAPEX	–	Capital Expenditure
CPDU	–	Control Power Distribution Unit
DAF	–	Dynamic Amplification Factor
DNV	–	Det Norske Veritas
DP	–	Dynamic Positioning
FAT	–	Factory Assurance Test
FPSO	–	Floating Production Storage and Offloading
GSC	–	Gullfaks Subsea Compression
HSE	–	Health, Safety and Environment
IMR	–	Inspection, Maintenance and Repair
IOR	–	Increased Oil Recovery
JIP	–	Joint Industry Project
KPI	–	Key Performance Indicator
LARS	–	Launch And Recovery System
MBL	–	Minimum Break Load
MEG	–	Ethylene Glycol
MHS	–	Module Handling System
MTTF	–	Mean time to failure
NCS	–	Norwegian Continental Shelf
OEM	–	Original Equipment Manufacturer
OPEX	–	Operational Expenditure
OREDA	–	Offshore Reliability Data
O&G	–	Oil and Gas
RAO	–	Response Amplitude Operator
ROV	–	Remotely Operated Vehicle
SCS	–	Subsea Compression System
SHS	–	Special Handling System
SRSWI	–	Subsea Raw Seawater Injection
SSBI	–	Subsea Separator, Boosting and Injection
SSF	–	Statoil Subsea Factory™
SWL	–	Safe Working Load
VSD	–	Variable Speed Drive
WI	–	Water Injection

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1 INTRODUCTION

1.1 PROBLEM DESCRIPTION AND BACKGROUND

Subsea technology has seen a rapid expansion over the last years, where new concepts that have proven themselves useful have been added to the ever expanding subsea technology toolbox. Today the subsea industry is getting ready to implement its latest technological wonder, the subsea processing facility. This technology would bring with it a step change that could forever amend the offshore industry and in the future make offshore platforms obsolete. The long term goal of subsea processing is to move the topside production facilities in its entirety to the seabed.

The concept behind subsea processing developments started out as a way to reduce topside weight, due to the continued success of subsea processing the motivation for applying this technology changed from reducing weight to replacing topside facilities. Subsea processing has already been used to extend the production life on declining fields, greatly increasing the oil recovery. The key driver for these installations is increased oil recovery, but additional benefits linked to subsea processing includes reduced cost, optimized production and reduced HSE risks.

Today (2014) there are three major projects (Åsgard, Gullfaks and Ormen Lange) ongoing that's based around subsea compressor stations. These will be the first commercial projects applying this technology. The complexity of the equipment and systems installed in these units surpasses any previous installations found on the seafloor. The subsea compression project is part of Statoil's larger concept called Statoil Subsea Factory™ (SSF) which aims to combine all necessary technologies needed for a full subsea process plant that will challenge the topside processing facilities used today.

Having the equivalent of a topside processing facility on the seabed introduces a range of challenges regarding the monitoring of the processes and the degradation of the equipment. Good procedures for inspection, maintenance and repair (IMR) will be a governing factor for the operability on such a complex and versatile facility. The components or modules on SSF will all have different dimensions, weights, life times and how critical the functionality of the module is for the production. Vessels specialized in IMR work will be needed to ensure optimal conditions for the SSF.

The SSF is still just a concept and it has no designated location, but the benefits for the concept are largest in areas where traditional topside facilities have a hard time operating, such as ultra-deep water (3000m) and where the climate is harsh i.e. the arctic, which gives the IMR vessels another hurdle to overcome.

1.2 SCOPE AND OBJECTIVE

The scope of this thesis is to define the challenges related to IMR on SSF and to evaluate how the implementation of a full subsea processing facility will affect IMR operations. Investigate and categorize the vessels needed/eligible/available to perform IMR on SSF. Elaborate on the need for condition monitoring of the critical components of the factory, to ensure timely IMR operations. Where the main objective is finding an optimal IMR vessel regarding module handling and ROV operations, which is able to ensure a high operability of the SSF and thus challenge topside processing facilities.

1.3 METHODOLOGY

SSF is still a concept and has not been accomplished yet so one will have to look at the installations happening today to get an overview of what the subsea factory will look like. This thesis will base its information on the subsea compression module that is to be installed on Åsgard later this year (2014). To achieve the objectives defined for this thesis the following steps will be taken.

1. *Describe SSF*
   In order to evaluate the vessel need, a fundamental understanding of how the different equipment (modules) on the subsea factory work is required.

2. *Define and discuss IMR regarding the subsea factory*
   Categorize the different operations related to IMR. Discuss the critical failures for the different component and how they are noticed. Discuss the specification of an IMR vessel and typical IMR operations.

3. *Evaluate the eligible vessels that can perform the IMR operations on Åsgard*
   Based on the frequency of critical failures of the modules on the compression station and the typical weather conditions found in the Norwegian Sea the weather window for different vessels and operations is established.

4. *Simulate a splash zone lifting operations in ORCAFLEX©*
   Combine the eligible IMR vessels with the modules that needs to be replaced and reinstalled. The lifting of these modules through the splash zone with different vessels is simulated in ORCAFLEX©.

5. *Analyze and combine the results and propose the vessel best suited for the operations*
   Based on the weather analysis, the splash zone lifting operation and the failure rates for the different modules, an evaluation of the most suited vessel for the operation is performed.

1.4 LIMITATIONS

Since the subsea factory has yet to be built the modules used in this thesis are based on existing technology found on different fields on the Norwegian continental Shelf (NCS). Special emphasis is given to components that are to be used on Åsgard for gas compression since these modules are currently the state of art equipment used in subsea processing which are likely to resemble the components that will be used for future subsea factories.

As no other Response Amplitude Operators (RAO) were available, the vessels RAO curves in this thesis will be based on the RAO curves acquired from the computer simulation program ORCAFLEX. The effects of heave compensating systems, guide wires, lifting slings and hull specializations i.e. bilge keels or bulbous bows are not included in the analysis. ORCAFLEX had no way to implement active stabilization systems for roll and pitch so this had to be neglected as well. For the analysis the lowering speed of the crane is also neglected since this is assumed to be significantly lower than the wave particle velocity and the crane tip velocity.

The critical failure data acquired from the offshore reliability data handbook (OREDA) is based on topside equipment as subsea processing had not yet been implemented in 2002 – 2003, the years the OREDA handbook takes its data from. The critical failures and mean time until failure (MTTF) found in this thesis are based on the topside data, but the contribution from elements not included in the subsea design are removed and not accounted for in the total critical failure rate.

2 BACKGROUND

2.1 DEFINING STATOIL'S SUBSEA FACTORY™

Statoil Subsea Factory™ (SSF) is a concept Statoil launched in 2012 at the Underwater Technology Conference in Bergen. The goal of the concept is to combine different aspects of subsea production and processing, thus removing the need for topside processing facilities and be able to achieve remote subsea to beach transportation of hydrocarbon for any offshore location. Subsea processing and production will have the potential to increase the recovery rate and accelerate the production as well as leaving a smaller environmental footprint [1].

Subsea processing can be divided into four different applications:

- Single and multiphase hydrocarbon boosting
- Separation systems
- Raw seawater injection
- Gas compression

The first three of these technologies have already been qualified and are operational on the NCS and by 2015 Statoil aims to have achieved subsea gas compression on two of their fields, Åsgard and Gullfaks and by 2020 the goal is then to have a complete subsea factory available. On an even longer term perspective, the vision is to implement a subsea factory that can operate in ultra-deep water and in arctic regions. This concept is a natural evolvement of Statoil's subsea processing where proven technology from earlier field developments is combined to create a fully operational subsea factory [2].

The natural evolvement often talked about is based on that if one subsea component, i.e. a separator or a subsea water injection station is successfully qualified it would be a great achievement by itself, but both of these components synergizes very well and their value increments when they work together. When connected these components would make it possible to not only separate the water from the well stream, but also re-inject the water into the reservoir. In other words, advances in subsea technology are in itself a drive for further subsea development [3]

Since no subsea field is exactly the same, it would not be possible to define a "one and only subsea factory", it has therefore been suggested that subsea factory realizations could be divided into the following subgroups, depending on the objective the factory aims to achieve:

- The brown field factory
- The green field subsea factory to host
- Subsea factory to marked

The brown field factory will be located close to existing facilities and will use smart solutions to extend the lifetime of fields in decline. The main goal of these factories will be to increase the recovery rate and maintain or accelerate the production. To achieve this seabed boosting and compression will be the main technologies that are to be used [2].

The green field subsea factory to host will be specialized and purpose build factories meant for fields which today is inaccessible. It aims to conquer the challenges of deeper water, longer step out and colder climate. New oil fields will typically be able to have a max production, also called plateau phase of 6-8 years, but if the green field factory will connect different fields together in a field center. This will prolong the production period and better utilize the transport capacity and the existing host facility. For these potential developments flow assurance will be of upmost importance to avoid hydrate or wax plugging in the long distance pipeline systems. New technologies that will be crucial for the green field factory will be compact subsea separation and coalescer technology, boosting and gas treatment as well as cold flow oil transport [2].

As previously mentioned the subsea factory is a gradual development of technology where the end goal is to bypass the need for platforms all together. Subsea factory to marked belongs to the future past 2020 where the processing and transport is of such a quality that further treatment will not be needed before being introduced to the marked. It's harder to define the technologies needed for this, since it's so far into the future, but it's safe to say it will be based on the technology used in both the brown field- and green field factory [4].

2.2 ACHIEVEMENTS RELATED TO STATOIL'S SUBSEA FACTORY™

Subsea processing is a technology that has gradually evolved to where it is today through constant focus on improvement. The technology that is used on today's subsea boosting systems would not have been possible without knowledge gained from close to 20 years of subsea installation experience. These achievements which can be seen in Figure 2.1 are the cornerstones in what is to be Statoil's subsea factory [2].

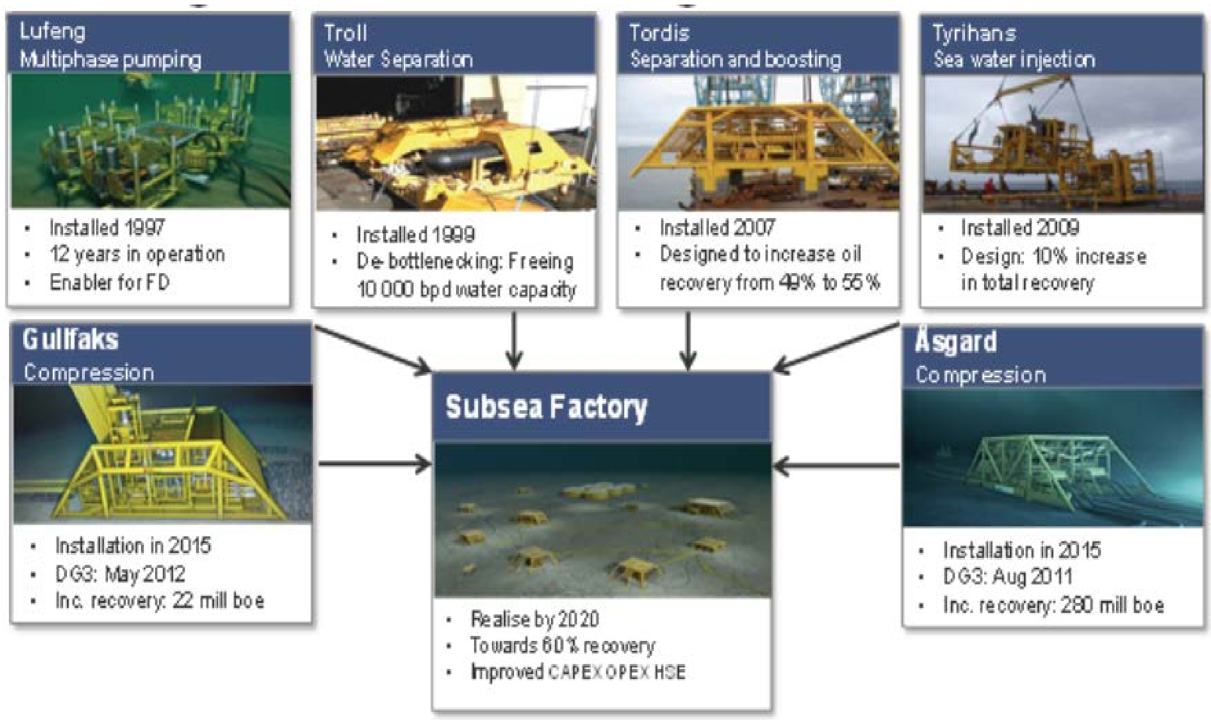

Figure 2.1: Statoil's existing subsea processing toolbox [2]

The effectiveness of these technological solutions in regards to increased oil recovery (IOR), CAPEX and OPEX reduction sheds light on how useful a complete subsea factory will be. It shows that this technology not only works when tested and qualified in safe and controlled environments, but also out in the real world. When these concepts are used more frequently the oil and gas (O&G) industry becomes less skeptical of their advantages. The O&G industry is known for being conservative and is slow to adapt new technology since this is associated with large uncertainties and risks [4].

2.2.1 MULTIPHASE SUBSEA BOOSTING ON LUFENG

The installation of single phase subsea pumps on the Lufeng 22-1 field in 1997 marks the beginning of the subsea processing for Statoil. Lufeng is a small oil field that was found in 1986, it's located approximately 265 km southeast of Hong Kong in the South China Sea. Due to the reservoir properties of Lufeng, it was deemed necessary to introduce artificial lift to keep an acceptable production rate up. Subsea booster pumps were chosen to do this, much because it was possible to install intervention and replacement facilities onboard the floating production storage and offloading (FPSO) unit, where the retrieval and replacement of a malfunctioning pump could be completed in less than one day [5].

Another challenge with Lufeng was that it's located in an area that is prone to typhoons, it was expected that the FPSO would have to leave the field twice a year. To overcome this problem a subsea quick connect/disconnect turret was installed on the vessel. By dropping this to 30m below sea level when a typhoon was incoming the FPSO could leave in 6-8 hours and easily recover the turret when back on location. This field went on producing oil until it was decommissioned in early 2009 having produced 17 million barrels more than expected [5].

2.2.2 SUBSEA SEPARATION WITH THE TROLL PILOT

In 2001 Statoil successfully deployed a subsea water separation system called "Troll Pilot" on the Troll field, this was then the first subsea water separation system in the world. The Troll field is a 750 km² large oil and gas field located in the northern part of the North Sea approximately 65km west of Kollsnes in Hordaland. Around 60% of the total natural gas reserves found on the NCS is located here which makes it the largest gas field in the North Sea and one of the largest gas fields in the world [6]

The oil production on the Troll field cannot be retrieved using the traditional vertical wells and is therefore based on a multitude of horizontal wells which are completed in the narrow oil zone. This configuration would cause large quantities of produced water shortly after start-up and is the reason why a subsea separation system was chosen. There were two main objectives for the Troll Pilot. Firstly separating the water subsea increased the fluid treatment capacity of the Troll C platform and in turn the hydrocarbon production. The second objective which could be seen as even more important was to demonstrate how subsea separation and boosting was a viable alternative to topside separation [5].

2.2.3 SUBSEA SEPARATION, BOOSTING AND INJECTION ON TORDIS

The Idea of subsea separation was taken a step further and in 2007 a subsea separator, boosting and injection system (SSBI) was installed on the Tordis field which is located in the Tampen area in the North Sea. This field was developed as a tieback to Gullfaks C facility and production on Tordis started in 1994. The reason for installing the SSBI here was because it was anticipated that increasing quantities of water and sand would be produced as the reservoir matured which in turn would create a bottleneck for the water treatment capacities at Gullfaks C [5]

The hydrocarbons separated by this system are pumped back to Gullfaks C through a multiphase pump while the produced water is re-injected into a disposal well. The SSBI system is more complex than the Troll Pilot and consists of six different modules. The separator, manifold, de-sander multiphase & WI pumps, water flow module and multiphase module. Each of these modules can be individually retrieved when repair is needed [5]

2.2.4 RAW SEAWATER INJECTION ON TYRIHANS

Another important building block for the portfolio of the subsea factory was the installation of a raw seawater injection system that was installed on the Tyrihans field in 2009. Tyrihans consists of two separate fields, located on Haltenbanken offshore mid Norway at approximately 300m water depth. To maintain high enough reservoir pressure while the field is depleted and as an increased oil recovery (IOR) method, gas and water injection had to be considered. Since the step-out distance from the closest production facility is over 30km, the capacity of gas injection on the closest production unit was limited. The probability of formation water breakthrough to the oil producing wells was also considered to be low, so there wouldn't be any risk of scaling. The subsea raw seawater injection (SRSWI) concept seemed like the best solution. It is anticipated that the field will have an IOR of 10% due to the SRSWI system [5].

2.2.5 GAS COMPRESSION ON ÅSGARD

The Åsgard field is located 200km from the cost of Trøndelag, like Tyrihans in the area called Haltenbanken in the Norwegian Sea. The oil, gas and condensate found here are produced through the topside infrastructure which consists of Åsgard A, B and C. Where Åsgard A is an FPSO used for production and storage of oil, Åsgard B is a processing platform that processes the gas and condensate and Åsgard C is used for storage of this condensate. This is one of the most developed fields on the NCS with over 50 wells drilled in 16 different templates [5]

The gas production from the satellite fields Mikkel and Midgard to Åsgard started in 2000 and so far the natural pressure in the reservoirs has been enough to maintain the production, but due to continued pressure loss and decreasing production it would be impossible to maintain a stable gas and condensate flow without intervention in the form of compression. With no compression the ethylene glycol (MEG), a hydrate inhibitor, flowing from Åsgard A, through Mikkel and Midgard and back to Åsgard B would accumulate in the flowlines due to the gas flow rate being too low. It was decided early in the production phase that Åsgard would be a suitable field to take use of the world's first subsea compression facility. The reasoning behind choosing this concept was the location and the field's contribution to the Norwegian gas export, also none of the existing facilities on Åsgard had sufficient payload or space for a traditional topside compression facility [7].

The gas compression system on Åsgard will be further discussed in the chapter 2.3 and is the basis for the analysis done in this thesis.

2.2.6 GAS COMPRESSION ON GULLFAKS

In May 2012 Statoil started a life extension project for Gullfaks and as part of this project Statoil is in the process of qualifying multiphase compressor technology. Gullfaks subsea compression (GSC) is expected to be in operation next year (2015) and will be the world's first wet gas compressor system. It is anticipated that a successful implementation of GSC will open the door for the development of several fields which of today are uneconomical to develop both on the NCS and worldwide [2].

The GSC system includes two 5MW wet gas compressors, coolers, gas mixers, power & utility systems and protective structures. Since the compressor system is fluid tolerant i.e. can handle liquid/wet gas there is no need for any liquid removal in the form of scrubber or separators. This makes the GSC system less complicated and the modules are easily retrieved [8]

2.3 CASE STUDY – ÅSGARD GAS COMPRESSION

There are many advantages with subsea compression systems (SCS) compared to the traditional topside application. By placing the compression system on the seafloor one will see an increase in the production, enhanced recovery from the reservoir and experience a reduction of the environmental footprint. Subsea compression brings benefits such as safer operation, due to being an unmanned operation, larger flexibility in field developments and simpler flow assurance strategies. There are currently two main types of compression systems under development, a multiphase compression system that is mainly applicable for smaller fields that are based on simplicity, low CAPEX and relatively low boost pressure. Larger fields with a larger step out distance will need more technologically enhanced systems including gas scrubbing upstream the centrifugal compressors to remove any remaining liquids from the dry gas are being employed [7]

The configuration and placement of the compression system is seen on Figure 2.2. The system is supplied with electrical power through 40km of high voltage power cables from Åsgard A, while the production is routed through the compressor station and back to the existing pipeline system and to Åsgard B [9]

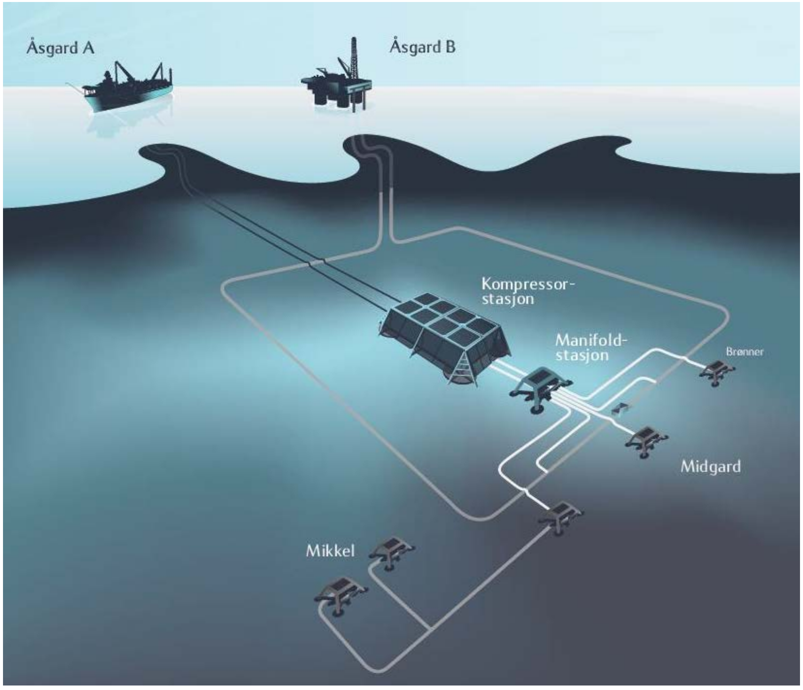

Figure 2.2: Åsgard subsea gas compression field layout [9]

According to estimates done by Statoil, the SCS installed on Åsgard will increase the lifetime of the field by 15 years and increase the recovery by 278 million barrels of oil equivalent. This is achieved by compressing and separating the condensate and gas from the well production on the seafloor. The gas is then boosted back into the flowlines and transported to Åsgard B 40km away [7].

The subsea compression station will consist of an enormous template (74m x 45m x 26m) weighing over 1800Te with two parallel 11.5 MW compression trains located in this template. All the dimensions and weights for the different components can be found in Appendix A.1. A new subsea compression manifold station will also be installed as can be seen in Figure 2.2 in front of the compression station. This manifold station will provide an efficient tie-in of the station to the existing pipeline system. It will also give routing flexibility of the production going back to Åsgard B for further processing [9]

Each of the two compression trains as can be seen in Figure 2.3 consists of [7]:

- Inlet and anti-surge cooler module
- Scrubber module
- Compressor module
- Pump module
- Discharge cooler module
- Subsea control system
- Subsea power system
- MEG distribution system

The whole compression process is explained in Appendix A.1.

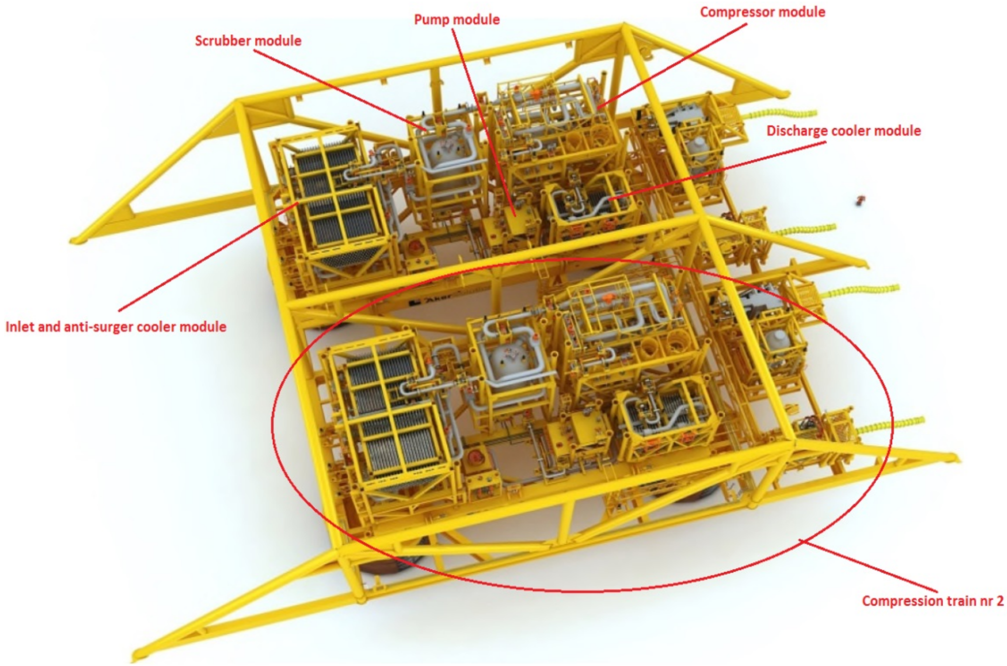

Figure 2.3: Top view of the subsea compression station [7]

The templates and power modules were installed in 2013, while the remaining modules making up the rest of the compression system are to be installed in the 3rd quarter of 2014. The system is meant to go online in 2015 [7]

2.3.1 INLET AND ANTI-SURGE COOLER MODULE

The well stream that enters the compression station will be cooled down in the inlet and anti-surge cooler module which can be seen in Figure 2.4. This is done to enhance the compressor efficiency and to increase the liquid volume fraction so more condensate is removed in the gas scrubber module. During shutdown/startup, compressed gas with be looped through the anti-surge line to keep the compressor running to ensure a safe startup/shutdown [10]. The cooling of this operation is also achieved by the inlet and anti-surge cooler module. The module weighs 235Te and the dimensions are 15m x 10m x 8m [9].

As the cooler module cools the well stream towards or even below the hydrate formation temperature, flow assurance is achieved by distributing MEG to all cooling pipes during all operational scenarios. Cooler modules are also prone to biological growth and contaminations caused by mineral deposition which in turn may degrade the cooler performance over time. In anticipation of this the cooler/heat exchanging surface is normally over dimensioned [10]

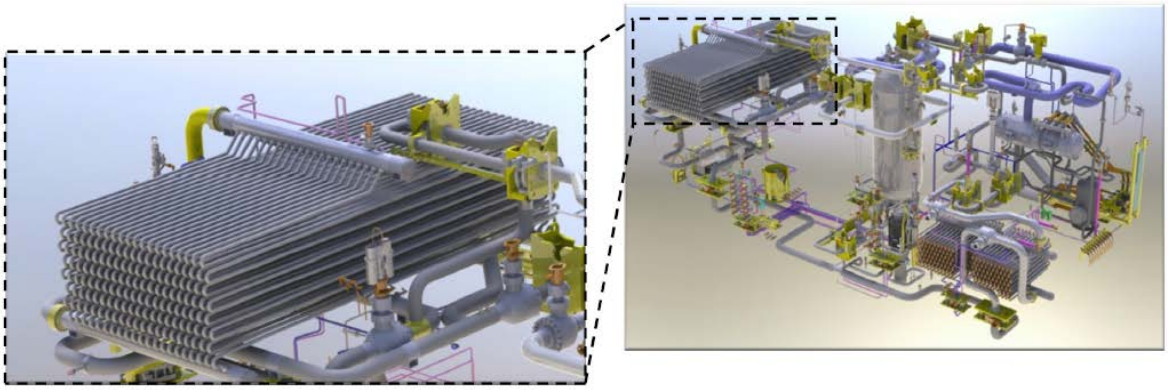

Figure 2.4 The inlet and anti-surge cooler module seen together with the whole compression system [9]

2.3.2 SCUBBER MODULE

The scrubber module will work as a slug catcher with the required volume to handle the slugs being produced due to well start-up or pressure transits. The innards are therefore designed to withstand the forces from the expected slugs for the entirety of the scrubber's lifetime. To reduce the forces generated by the slugs, the liquid levels are monitored and detected upstream, and controlled by the liquid pump's variable speed operations. The scrubber module weights 210Te and the dimensions are 8m x 8m x 12m [9].

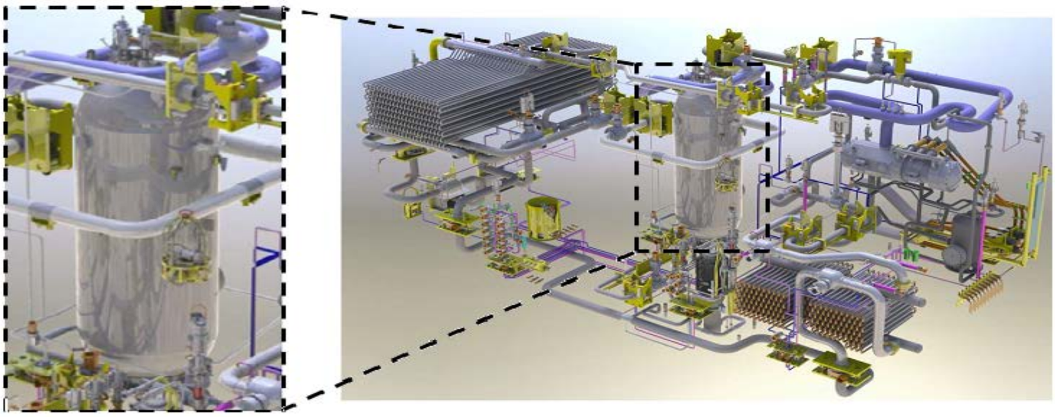

Figure 2.5: The scrubber module seen together with the whole compression system [9]

To effectively separate the liquids and the gas a spinlet inlet arrangement is used on the scrubber. This accommodation distributes the flow in such a way that liquid is guided downwards on the scrubber wall, while the gas is pressed upwards through the center of the scrubber. To prevent the build-up of solids in the bottom of the scrubber continuous liquid jetting is required. This is achieved by continuous recycling from the condensate pump to the lower part of the scrubber [11].

2.3.3 PUMP MODULE

A flowchart of a typical subsea pumping system can be seen in Figure 2.6 where the red part is the pump module and the blue part is the pump manifold. The pump module (red part) can easily be disconnected from the manifold (blue part) by opening two clamp connecters, and replaced with another module. Should the pump fail, the production can be diverted around the pump, but at a lower flow rate. The pumping system has a high voltage motor that is controlled by a variable speed drive that drives a screw type pump. To prevent overheating and lubrication for the system a barrier fluid is used [12]. The main purpose of this fluid is to:

- Cool the high voltage motor
- Lubricate and cool the motor and bearings
- Lubricate and cool the seals between the pump and the motor

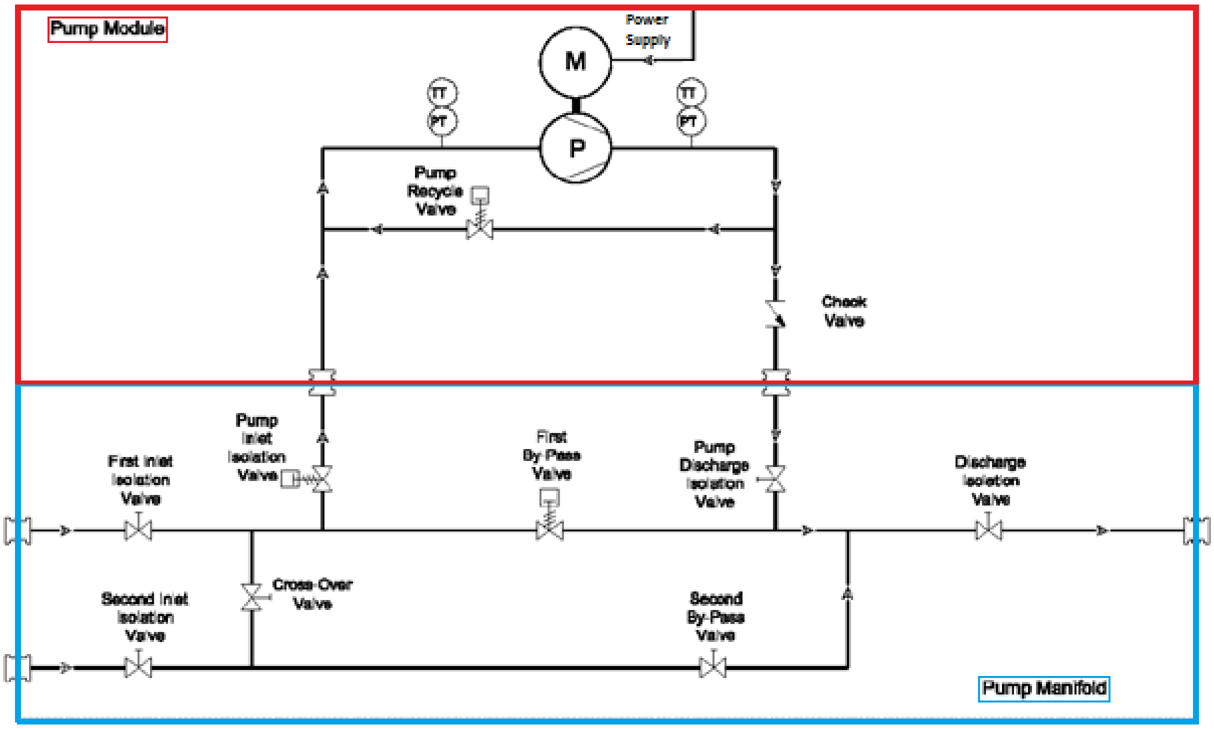

Figure 2.6: A retrievable pump module and the pump manifold [12]

The barrier fluid system is kept at a slight overpressure relative to the process, so that if there are any leaks, no processed fluid enters and contaminates the barrier fluid. This overpressure is controlled by a pressure and volume regulator, which takes into account that changes in temperature, also changes the volume of the barrier fluid. The consumption of barrier fluid is closely monitored as this is a way of detecting leaks in the system and it's used for planning interventions on the pump system [12]

The pump module is the smallest of all the modules on the Åsgard compression train weighing 45Te with the dimensions 5m x 5m x 6m [9]. The pump module is one of two components that have rotating machinery in the subsea compression train and is therefore expected on a general basis to require maintenance more frequently than other components. According to data gathered from OREDA 2009 shown in Appendix A.6, the mean time to failure (MTTF) for a subsea pump module is estimated to 5 years. It is therefore especially important to monitor the pump performance to be able to anticipate and plan the needed IMR operation [12].

2.3.4 COMPRESSOR MODULE

This module will be the core of the whole compressor train and it's here the incoming cooled gas has its volume decreased and pressure increased, also known as compression. The high speed motor and the centrifugal compressor are installed in a common pressure vessel. The enclosure is pressurized and a barrier system separates the motor from the compressor. This ensures a clean operating atmosphere for the electrical motor. To improve the reliability of the compressor module, magnetic bearings are used between the electrical motors drive shaft and the compressor [10]. These bearings are oil-free, frictionless and therefore virtually maintenance-free compared to mechanical bearings. So instead of lubrication the bearings only require low voltage power to function. The compressor module is the heaviest module on the compressor train weighing 289Te with the dimensions 11m x 9m x 10m [9]

The electrical motor is located slightly higher up than the compressor so in the event of any liquids entering the unit the motor will still stay dry as gravity will force any droplets or solid particles to the bottom of the compressor vessel. The motor is kept at a safe temperature by a cooling gas that is circulated in the motor enclosure [9]. The gas is cooled in an external seawater cooler. This closed cooling loop prevents the motor from any exposure to the unprocessed gas that is being compressed in the compressor and thereby increases the reliability of the motor [10].

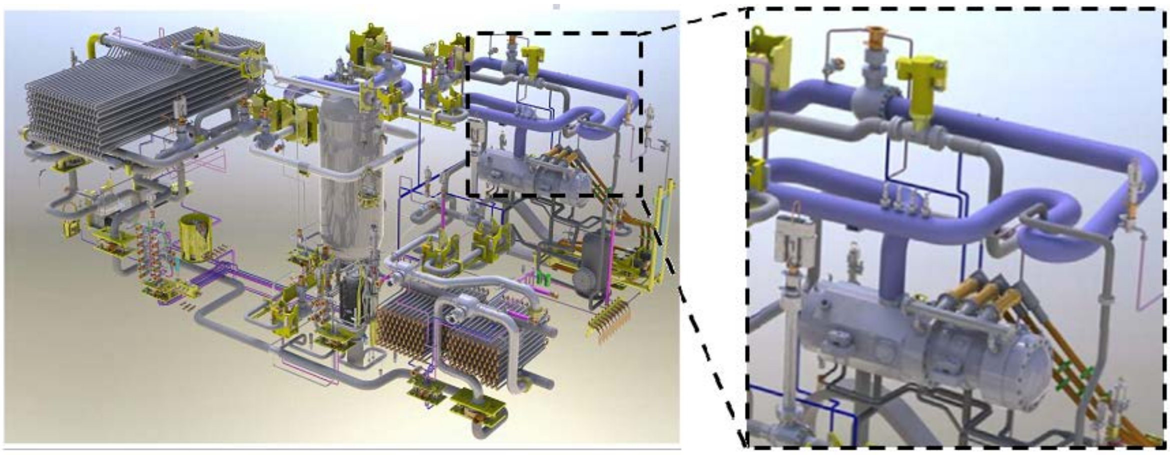

Figure 2.7 The compressor module seen together with the whole compression system [9]

The huge dimensions of the compressor module are one of the main reasons why Statoil want to use Technips new immense IMR vessel; the "North Sea Giant" to do the IMR workover for the Åsgard compression station. It's hard to predict a trend for subsea processing, but it is believed that future compression stations will have smaller modules than the ones seen at Åsgard [1]. In fact a JIP called "Subsea compact GasBooster™" have been launched that aims specifically at making the compressor module more compact thus making it more flexible and cost effective to install and retrieve [13].

2.3.5 DISCHARGE COOLER MODULE

As the compressed gas exits the compressor module it has again been heated up by the compression process. The discharge cooler module is there, like the inlet cooler module, to cool the flow coming before it is reconnected to the flowlines. The main difference between the inlet cooler and the discharge cooler is that for the latter the liquid concentration will be almost non-existent as the liquid has been separated in the scrubber. This prevents the discharge cooler from having the enormous dimensions of the inlet cooler [10].

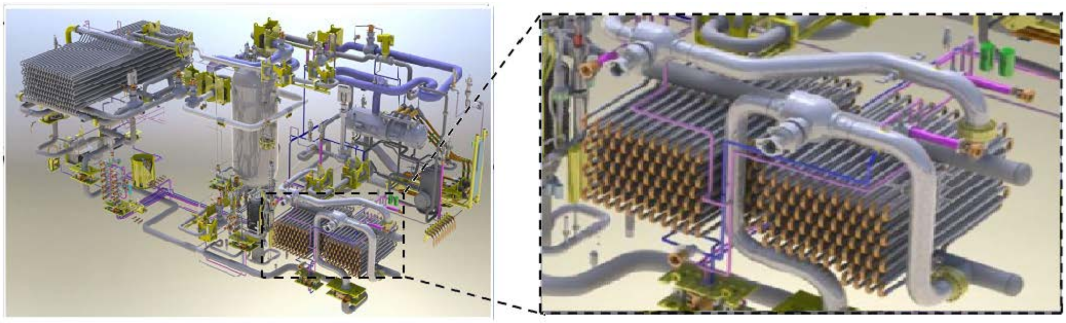

Figure 2.8: Discharge cooler module seen together with the whole compression system [9]

From Figure 2.8 one see that the dimensions approximately half of the inlet cooler. The discharge cooler module weighs 107t with the dimensions 9m x 7m x 5m [9]

2.4 MODELLING AND SIMULATION

2.4.1 ORCAFLEX©

ORCAFLEX is a program frequently used within the oil & gas industry due to its easy to use graphical interface. The program is capable of analyzing a range of different marine operations such as pipe lay-, riser- and lifting analysis. ORCAFLEX is a finite element program developed by Orcina that is used for nonlinear time domain analysis. The program uses elements with six degrees of freedom with a “lumped mass” to simulate structural elements such as pipes, plates or beams. The elements are used to simplify the mathematical formulation which reduces the overall computational time. The hydrodynamic forces are calculated based on cross flow assumptions and the Morison equation [14]

2.4.2 OREDA

ORDEA (Offshore REliability DAta) is a project organization supported by eight of the largest oil and gas companies with operations all over the world. The main purpose of OREDA is to gather and exchange reliability data between the participating companies and to be the standard for management of reliability data collection for the oil and gas industry. Reliability data has been collected from over 265 installations with more than 16000 equipment units. In total the database contains over 38000 failure and 68000 maintenance records related to topside and subsea equipment.

The 5th edition of the OREDA handbook was released in December 2009. The OREDA 2009 handbook is a unique data source for failure mode distribution, failure rates and repair times for equipment used in the oil and gas industry. The handbook is divided into two volumes, where volume 1 contains all the topside equipment and volume 2 covers subsea equipment.

Since subsea gas compression is a new technology, the data gained from OREDA is based on the equivalent topside equipment with some modifications e.g. the contribution from elements not included in the subsea design was removed from the total failure rate [15]

3 STATE OF THE ART

3.1 INTERVENTION, MAINTENANCE AND REPAIR

IMR is the term commonly used for intervention on operational subsea structures. This includes operations such as inspection of pipelines, anchor handling chains or platform legs using the onboard ROVs. The IMR vessels also perform repairs and removal on subsea installations, which could be everything from valves, control pods and chokes, or even sometimes whole modules [16]. Due to the limitations of the ROV when it comes to repair, malfunctioning modules must often be extracted and replaced instead of being repaired subsea, which emphasizes the need for cranes and module handling systems that can handle heavy lifts in harsh weather. It is therefore critical that subsea installations are divided in smaller modules that can be separately retrieved [16]. The sequence for a typical IMR operation can be seen in Figure 3.1, where Subsea 7's "7 Viking" can be seen in the middle [17].

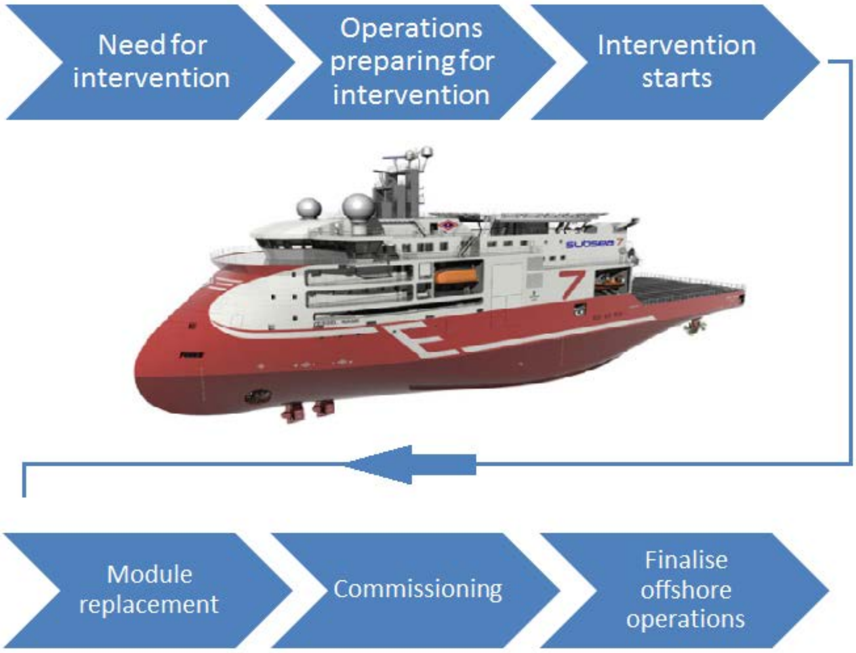

Figure 3.1: The sequence of operation for an offshore IMR operation [17]

There are three different scenarios leading up to an IMR operation. There can be an immediate need for an intervention, also known as an unplanned intervention. These types of interventions are most critical as they reduce the production or even force an entire field to shut down. The second scenario is a planned intervention which is scheduled according to the maintenance plan. There can be an entire shutdown of the production facility lasting one week where all the planned maintenance takes place. The last scenario is opportunistic maintenance, this refers to preventive maintenance which is carried out when an opportunity presents itself. A typical example could be when one component is removed for maintenance and it is decided to remove another part and repair this one ahead of its planned maintenance plan. Opportunistic maintenance is typically carried out in a way that is cost-saving as two maintenance activities are performed at the same time. [18]. Condition monitoring is an important tool when deciding which parts to include in the opportunistic maintenance. These three categories and their frequency can be seen in Table 3.1.

The range of activities covered by the acronym IMR could be divided into three different categories; these are inspection, maintenance and repair operations [18].

The inspection operations normally consist of condition monitoring activities such as:

- Structural inspection
- Pipeline inspection
- Corrosion monitoring
- Inspection of free spans and the need for gravel dumping or placing protective mats

The equipment used for these operations are ROVs with non-destructive testing tools. These operations are fairly common and highly standardized and are normally completed in a combination with more demanding maintenance or repair operations [16].

Maintenance operations are often scheduled interventions or restorations based on data from condition monitoring- and performance degradation systems. This includes replacement of:

- Anodes
- Subsea control modules
- Jumpers
- Subsea pumps or other rotating machinery

Removal and cleaning of debris such as unexploded ordnances or fishing nets etc. left by third parties are also considered maintenance operations. Compared to inspection activities maintenance operations are more demanding as they require more equipment, such as running tools when replacing or removing modules, they are also more dependent on the weather conditions as heavier equipment is lifted and the operations tend to take longer to execute [18].

Repair activities are more specialized and often require custom engineered solutions compared to the other activities, as seen in Table 3.1. Removal of a module uses standardized procedures and equipment that is easily available from original equipment manufacturer, while a pipeline leakage will differ from leakage to leakage [18].

Repair operations may include the following:

- Repair of broken or corroded caissons or suction anchors
- Repair of raiser guides
- Repair of template hatches, locks or hinges
- Arrest of propagating cracks
- Clamping of leaking pipes

Table 3.1: The complexity, types and frequency of the different intervention types [18]

	Inspection		**Maintenance**		**Repair**	
	Recurring	Special	Clean & Clear	Modules	Clamping	Replace
Scheduled						
Unscheduled						
Opportunistic						
	Basic tools and procedures		Standardized tools & procedures		Engineered tools and procedures	

Increasing Frequency

Increasing Difficulty

In some cases the IMR vessel is also used for riser less well intervention where it performs operations such as well perforation, scale squeeze or setting and pulling different plugs. It never takes control over the well, and is thus per definition a ship and not an offshore unit since no hydrocarbons are transported to the vessel. This is important due to different safety regulations for the two cases [19].

The ROVs are either launched over the side or through the moonpools. The “over the side” launch with no guiding through the splash zone will have a lower operational limit than launching the ROV through the moonpool while using the specialized cursor guiding system [16].

Module retrieval and installation is the most critical operation for an IMR vessel. The weight and the size of the module will be the limiting factor when choosing what vessel or crane to use for any IMR operation. As with the ROVs, the modules are either lifted through the moonpool if they’re small enough or over the side using the deck tower or crane. All these operations are weather dependent. As high waves causes large motions, accelerations and thus large forces on any object lifted through the splash zone. In the latest years there have been developments in specialized lifting and lowering towers used for overboard lifting. The specialized IMR vessel North Sea Giant has such a Special Handling System (SHS) which is designed to dampen the splash zone forces and thus increase the weather window in which modules can be installed and retrieved [16].

3.2 IMR FOR THE SUBSEA FACTORY

Failures are a part of any facility and for subsea facilities there are no way of solving these problems in a quick and cheap manner. Good maintenance is therefore especially important to maintain performance and availability. In general, subsea wells have had a much lower recovery rate than topsides platform, this is mainly because the enormous gap in costs between intervention operations topside compared to subsea [20]. The same is true for any subsea installation that is meant to replace topside equipment. The observed failure rates in subsea installations have been shown to follow the bathtub curve as can be seen in Figure 3.2. As more is learned about the failure causes in subsea equipment the futuristic goal is to be able to design components with no early life failures and fewer random failures.

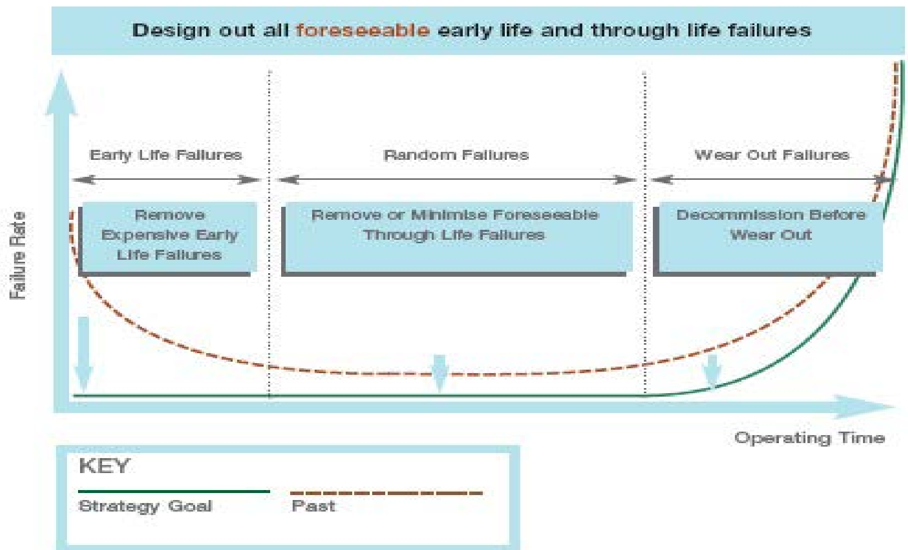

Figure 3.2: Typical failure patterns of subsea facilities - The bathtub curve [21]

The first generation subsea installations had a much higher failure and reliability problems due to, for instance, leakages, material problems, etc. that caused long downtime and high cost, as shown in Table 3.2 [18].

Table 3.2: Early experiences of subsea failures [18]

Project	*Failure mode*	*Direct Cost*	*Downtime*
Foinaven	Super duplex (steel pipe) cracking	$55M	10 months
Foinaven	(Valve) Stem seal leakage	$30M	4 months
Schiehallion	13 subsea control modules suffering hydraulic leakage; 9 modules replaced	$9M	N/A
Troika	Replacement of 8 conductors due to leakage, production was deferred	$20M	N/A

To combat the high early life failure warranties are given by the original equipment manufacturer (OEM) and vigorous testing before and during commissioning. The OEM normally only does the functional unit testing while the whole system goes through a system integration test supervised by the operator i.e. the subsea compression station at Nyhamna for the Ormen Lange field [20].

The operating experience gained from earlier subsea installations together with OREDA handbook will give an indicator of what components that are likely to fail regarding the components used in the development of SSF.

Statoil has also gained a lot of information from their prior installations mentioned in sec. 2.2. Some of these failures are summarized in Table 3.3.

Table 3.3: Failures in prior subsea installations [6].

Project	*Component*	*Failure modes*	*Comments*
Lufeng	Pump	Substantial consumption of barrier fluid	Mechanical seal leakage, replaced the seal
	Control system	Control/signal failure Erratic output	Due to redundancy the control was never lost. The failures represents the loss of redundancy
	Electrical connectors	Deterioration	Lack of refurbishment, retrieved and replaced
Troll	WI Pump	Reduced performance	General intervention and replacement
	Electrical connectors	Earth fault	Retrieved and replaced
	Separator	No inductive level instrumentation	Replace electrical jumper and common control pod
Tordis	Multiphase meters	Reduced performance	Scale-up related calibration, back-up battery problems.
	WI Pump	Increased pressure	Seismic data showed non-optimal placement of water injector

3.2.1 CONDITION BASED MAINTENANCE AND CONDITION MONITORING

For most of the components for the subsea industry, there is a requirement that the design life of the component is as large as the field life [22], but for subsea systems with rotating equipment such a long operation time is impossible. From the OREDA 2009 handbook it is assumed that a subsea pump and compressor will have a mean time to failure (MTTF) of 5 – 10 years, depending how hard the components are operated. For rotating subsea equipment, service is typically done by swapping a broken module with a new one. As these modules are quite heavy an intervention vessel with enough crane capacity might be difficult to find, and take time to mobilize.

Since it's known that maintenance will be needed for any rotating equipment, a good maintenance strategy is needed. Three typical maintenance strategies are [23]:

1. Run until failure
2. Replace after a certain limit is reached (time based maintenance)
3. Replace when performance is reduced, but before critical failure (condition based maintenance)

In practice a combination of these three strategies are used since they each have some merit and since it's not possible to detect every failure and sometimes failures occur without any warning.

If strategy 1 was to be used for a subsea facility the time between each time intervention was required would be maximized, but the downtime would be long whenever a component broke down as the mobilization time for an IMR vessel is typically in the order of a month [23].

Strategy 2 is the safest way of operating and is used in the aircraft industry as this strategy greatly reduces the risk of any unplanned breakdowns. For a subsea facility it would mean a must shorter downtime, typically one day instead of a month. This will be the strategy with the most frequent interventions and there is always a certain risk associated with any subsea intervention as well as an extra cost for refurbishment of the replaced pump or compressor module that is replaced [23].

The 3rd strategy is the most optimal solution if performed correctly. It's situated somewhere between strategy 1 and 2 in frequency of interventions, but if managed optimally the downtime will be just as short as with the time based maintenance since the modules are replaced before critical failure occurs. The challenge with this strategy is to know when to replacement should take place. If it's changed to early, resources that could have been saved is lost while if the module is run until failure nothing is saved compared to strategy 1 [23].

A quick calculation based on the three different strategies found in Appendix A.5 is shown in Table 3.4 which describes the expected yearly expenses using the different strategies.

Table 3.4: Intervention strategies and their expected yearly expenses (given in MNOK/year)

	Pump module	Compressor module
Strategy 1	18,33	93,33
Strategy 2	7,67	12,67
Strategy 3	3,89	6,42

From Table 3.4 one can see that strategy 3; condition based monitoring is the most cost effective solution and that an unexpected breakdown, especially for the compressor module will cause huge yearly expenses. The potential for saving expenses by optimizing the maintenance are huge, particularly if the failure can be detected at least a month before the module breaks down.

This concept is commonly known as condition based maintenance (CBM), which in essence is to complete the maintenance when the need arises. For the subsea factory this need for maintenance would be predicted based on sensors and indicators, should anything indicate that equipment is going to fail or that the process is deteriorating maintenance would be called for. This does not work for every process as random critical failures can occur without warning, but for systems where faults develop slowly over time and where the degradation is measurable, CBM is applicable [23].

The terms for condition monitoring (CM) and CBM are similar and to some extent overlapping, one way to describe these two principles is by the use of a pyramid shown in Figure 3.3. The lower levels of the pyramid refer to CM where the current and past state of the machine is logged and categorized, while the upper part of the pyramid, the CBM takes the collected data, analyzes and estimates kea performance indicators (KPI) and tries to predict how the changes that are seen now will affect the future and thus determine when the component needs maintenance [23].

Logging data can easily be achieved while making sense of all the data is what's proven to be challenging. Ideally the data should be converted into a system that is easy to understand and use, such as a traffic light interface. Green would symbolize that everything is ok, yellow means there is cause for concern or that the process cannot work at full capacity while red would symbolize that immediate action must be taken [23].

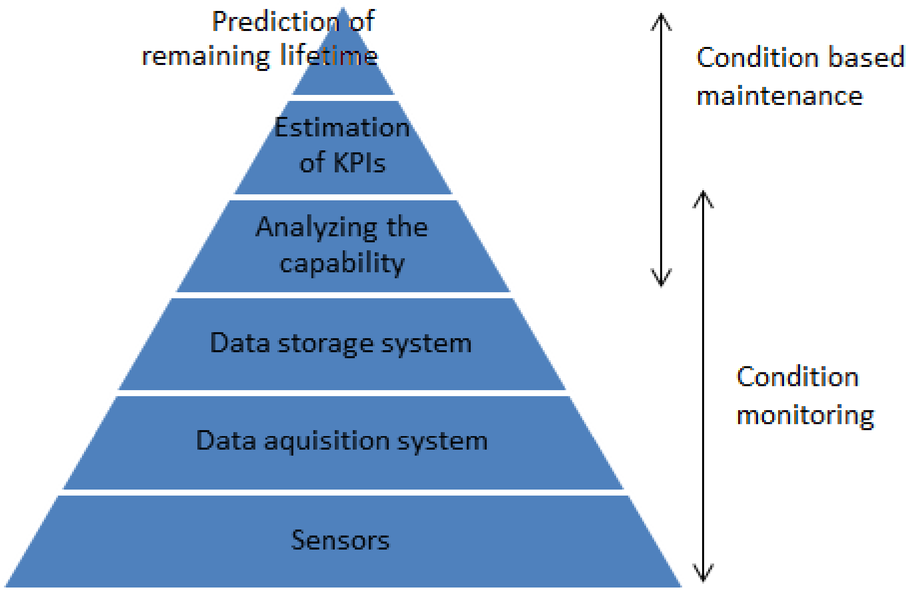

Figure 3.3: Condition based maintenance and condition monitoring [23]

As mentioned in the beginning of this chapter, the time it takes for a machine to degrade depends on how hard it is driven. Take ball bearings as an example; the lifetime for a typical bearing is given as [24]:

$$L_{10} = \left(\frac{C}{P}\right)^{a} \qquad [10^6\ rev]$$

Where

L_{10}	-	lifetime of the bearing (90% reliability) $[10^6\ rev]$
C	-	dynamic load rating $[KN]$
P	-	dynamic equivalent load $[KN]$
a	-	exponent of the life equation, 3 for ball bearings 10/3 for roller bearings $[-]$

If the load, P is doubled the formula for a ball bearing becomes:

$$L_{10} = \left(\frac{C}{2P}\right)^{3} = L_{10} = \left(\frac{C}{P}\right)^{3} * \frac{1}{8} \qquad [10^6\ rev]$$

From this one can see that when the load is doubled the lifetime is reduced by a factor of 8. If the load is reduced to 80% from the original load, the lifetime of the bearing is almost doubled. Thus by logging power, speed and run hours, one can convert the data from actual run hours to effective run hours and estimate the remaining lifetime of the ball bearing.

Another KPI that is important to monitor for subsea pumps is, as mentioned in chapter 2.3.3, the lube oil consumption. The lube oil is supplied through a very long umbilical and since the viscosity of this oil is high, only a certain amount of the lubricant can be supplied to the pump. As the pump seal starts to deteriorate, the lube oil starts leaking out into the process (due to the overpressure) and the lube consumption increases with the leakages. At one point if no maintenance is provided the leakage will exceed the maximum deliverance of lube oil and the pressure differential is lost, the process may then leak into the pump and damage it [12], the lube oil consumptions is therefore closely monitored and maintenance schedules are based on this monitoring.

The compressor module on Åsgard is, as mentioned in 2.3.4, fitted with magnetic bearings in order to achieve a longer lifetime before service. An important KPI is the current supplied to these bearings. As time goes by the compressor becomes more unbalanced, and the magnetic bearings must work harder to keep the shaft centered and more current is needed, there is a limit of how much magnetic force the bearing system can generate, so eventually it may be overwhelmed. By monitoring the current supplied to the bearings it is possible to track how the available magnetic power decreases over time, and replace the module before the bearing system is overwhelmed [23].

All rotating equipment will have a signature vibration, by monitoring the vibration frequency it is possible to say something about the condition of the rotating machinery. The vibration of the machinery is logged when it is has its factory assurance test (FAT). The vibration is then compared to the FAT vibration to evaluate the deterioration of the pump and maintenance is planned based on this [12].

Subsea pumping systems are starting to be known as a stable and reliable technology, with over 50 subsea pumps deployed on the seabed [23]. Subsea compression stations are just now emerging, but are more complex systems. Some of the experience gained from condition monitoring of subsea pumping systems can be transferred to the compression systems as these two technologies are somewhat similar.

3.2.2 IMR VESSELS

An IMR vessel is a highly technical vessel used in the offshore business. The main objective of an IMR vessel is inspection and repair of subsea facilities and installations, but it is also used for other tasks such as light construction activities, scale treatment or as a “tie-in vessel” where it supports different kinds of driverless connections of pipelines and spool pieces. The most essential equipment found on the IMR vessel is the ROVs, the cranes and the module handling tower. Usually there are several highly specialized work class ROVs and one or two smaller ROVs used for observation and inspection [16].

The limiting seastate that the IMR vessel can operate in is one of the most critical vessel specifications and there is a constant chase towards improving this limiting factor. In 2001 the platform supply vessel, Far Saga was modified and equipped for the IMR marked and this vessel was one of the first vessels equipped with an advanced launch and recovery system (LARS) which included active heave compensated (AHC) winches. This increased the limiting H_s of the LARS for the ROV’s from 3.0m to 4.5m [16].

In 2005 Statoil announced a tender for a high capacity vessel and in 2006 Shell followed up with a tender for a vessel with similar capabilities. In 2008 Deep Oceans “Edda Fauna” stood ready at Statoil’s disposal while Shell had Subsea 7s “7 Viking” ready for their IMR work. Both vessels have sheltered hangars for module and ROV handling through moonpools. The limiting seastate for ROV operations for these vessels is 5,0m H_s and their length is both around to 110m long [16].

For an IMR vessel it’s important that the vessel can withstand the motions induced by the waves and wind. There are several design principles that have gained popularity in the IMR business such as bilge keels to prevent roll motion and bulbous bows to increase buoyancy and reduce pitch motion.

A bilge keel is a pair of short, flat plates protruding from the hull that presents a sharp obstruction to roll motion. The bilge keel should be located on a low strake on the vessels hull so to not increase the vessels draft. Bulbous bow is a protruding bulb at the front of the vessel which reduces the drag on the vessel by canceling out the bow wave, where the bow wave is the wave formed immediately in front of the vessel. With a bulbous bow placed below the waterline ahead of the bow wave, water flows over the bow and coincides with the bow waves and the two waves cancel each other out thus reducing the vessels wake [25]. Another design that is crucial for IMR vessels are antiroll tanks which are large tanks situated within the vessel. The design of these tanks are such that a larger amount of water is trapped in the higher side of the vessel, so when the vessel rolls to one side, the tanks counters this movement and prevents some of the rolling.

Ulstein Design came up with a new bow concept called the X-Bow® for an anchor handling vessel in 2006 which has been implemented for several IMR vessels over the years. The bow design is characterized by slender water entry lines, no bulb at the front of the vessel and the bow is sloping backwards. An example of an IMR vessel with the X-Bow® design can be seen in Figure 3.4. According to Ulstein the slender hull water line and smoother volume distribution in the foreship leads to reduced slamming and resistance, increased operational window and increases the vessels seakeeping abilities [16]. Today there are over 40 vessels, either in production or completed, equipped with the X-Bow® and a good part of these are being used for IMR work around the world [26]

Figure 3.4: Ulsteins X-Bow® design on an IMR vessel [26]

As one study the growing portfolio of the IMR vessels today, one clearly notices a trend of building larger and more technologically advanced vessels. The North Sea Giant is such a vessel with its 160,9m length over all and 30m breadth, further specifications can be seen in Appendix A.4. The North Sea Giant is the vessel that will handle all the installations of all the different Åsgard compressor modules.

3.2.3 MODULE HANDLING SYSTEM

For lifting operations done on an IMR vessel a module handling system (MHS) or a deck crane is used [16]. The main components in a standard MHS is:

- Integrated tower in vessel's hangar or stand-alone deck tower structure
- Main lift-line winch system
- Guide wire system
- Cursor system
- Moonpool doors and vessel interface system
- Deck skidding system

The stand-alone deck tower will offer more flexibility and be able to handle larger modules, while the hangar integrated alternative offers more a safer and more comfortable working environment. The main lift-line winch system are typically qualified for a water depth of 3000m and a safe working load (SWL) from 10-300Te, but there are special handling systems with a SWL of 400Te [28]. These winch systems uses active heave compensation and auto tension/constant tension to greatly reduce dynamic loadings and improve the lifting capabilities [27].

The guide wire system is there to ensure accurate load handling and precise mating of subsea equipment onto the seabed during operations in harsh weather and rough seas, while the cursor system is there to guide and securely grasp the lifted module during deployment or retrieval through the moonpool. Controlled lowering of the cursor system and the module prevents it from swinging within the confines of the moonpool, reducing the risk of damage to the load. As the water depth is increasing it becomes a challenge to use guide wire systems, the increasing weight is obviously one concern, but more operationally challenging is the fact that the offset due to current and entanglement of wires are increasing [27].

The moonpool systems can differ from vessel to vessel, some with simple hinged hatches others with complex three-part doors with different wire hatches. The standard size of moonpools are 7,2 x 7,2m, but some vessels have systems that can extend the length of the moonpool to be able to handle larger modules. The deck skid system is there to provide a mean for the operators to safely move the heavy modules around on deck, into and out of hangars and to place them on moonpools, while the vessel is in transit, despite rough weather. These systems contribute to safe and time efficient vessels [27].

The modules for the Åsgard compression station that are able to pass through a standard 7,2m x 7,2m moonpool will be the modules with dimensions less than 6m x 6m x 6m. These are as follows [28]:

- Pump module
- Pump transformer
- Control power distribution unit (CPDU)
- CPDU transformer
- MEG module
- Subsea control module

The pump module will be the component with the lowest mean time to failure (MTTF) according to OREDA 2009 and therefore the module is chosen for the analysis in chapter 5, but to simplify the analysis it is assumed the pump module is lifted over the side of the vessel.

3.2.4 SPECIAL HANDLING SYSTEM (SHS)

When the modules dimensions of the modules are too large to go through the moonpool the alternative, over the side lifting operations must be done. To be able to perform these operations in harsh weather it is important to be able to limit the hydrodynamic forces the modules will experience. Technip have developed a SHS that is installed on the new IMR vessel North Sea Giant. This system will make it possible for the vessel to huge modules in weather where the significant wave height is as high as 4.5m [28]

The SHS consists of a large tower structure which has the ability to rotate around its central axis, while constricting a module from movements in any direction as can be seen in Figure 3.5. When the module is located over the side of the vessel it's lowered down by a railing system. An adapter frame is connected to the module and guides it through the splash zone until the module is lowered as much as possible over the side, as is illustrated in the figure below. The module is connected to a specialized adapter frame through four hydraulic locked pad-eyes, which can be disconnected when the module is safely deployed [29].

Figure 3.5: The SHS lowering a subsea module through the splash zone [29]

The module is skidded into position to be attached to its specialized adapter frame and picked up by the tower. It is then lifted up and swung over the side of the vessel. The towers sliding frame and the module will be lowered first on the cursor rails and then further down onto specialized vessel rails which allows for a deeper deployment of the module which is essential in regards to slamming forces on the module. The lowering speed is 0,1m/s when lowering through the splash zone while 0,5m/s during the rest of the operation [29].

3.3 MARINE OPERATIONS

An arbitrary IMR operation will essentially have many of the same aspects as an installation operation, beginning with mobilization from port. The time from the mobilization begins till the vessel has successfully completed the operation can take from one week up to eight weeks depending on the transit distance and the season [30].

3.3.1 SEAFASTENING

To ensure safe transportation to the offshore location sea fastening is required as wind and waves induces large movements on the ship and the modules if they are not safely secured. Failure to safely secure the cargo could lead to damage to the modules or human injuries or fatalities. Sea fastening is normally obtained by using bolt, chains, frame solutions, nuts, pins or sockets. Welding entire steel profiles or plates to the deck is another way to ensure safety when transporting cargo. Modules will have a predetermined place on the vessel to ensure that the stability of the vessel is intact as the modules can be heavy and numerous enough to influence this. Good planning of the placements of the modules also leads to easier handling [25].

Norsk Standard [31] Sec 8.2.3 states that: “Design of grillages and sea fastening shall facilitate load-out and subsequent release, shall provide adequate vertical and horizontal support and shall be such that welding and flame cutting do not inflict damage to the transported object. The contribution from friction shall be disregarded in the design of sea fastening and grillage.”

3.3.2 TRANSIT

The transit time can be a large factor of the total time it takes to complete the operation, especially for greenfield subsea factories which might be located in remote, deep or arctic regions of the world. The weather situation will also influence when the transit can take place. It is possible to weathervane the vessel, but the extra fuel consumption and the slower transit speed doesn’t always make this the most cost efficient solution [25].

Norsk Standard [31] Sec. 8.2.3 states that: “Prior to sail away, it shall be verified that the vessels, towing system, transported object, sea fastening, navigation aids, voyage protection, etc. as well as preparations for the next phase are completed, all are in every respect seaworthy, in accordance with the design and ready for the voyage, and that all debris is removed.”

3.3.3 DYNAMICALLY POSITIONED UNIT SYSTEM

During the retrieval and installation of new modules it is critical that the vessel is stationary, this is achieved through dynamic positioning (DP). The operational limit for DP systems is based on the current affecting the vessel. For retrieval and installation of modules that are fitted into other structures (such as a template or a cassette solution) it is especially important to have a good DP and AHC system to carefully guide the modules and prevent slamming. DNV-OS-H203 [32], Sec. 5B shows a DP equipment class selection, but DP for retrieval and installation of subsea modules is not listed. DNV-OS-H203 [32], Sec. 5B-102 then states that “Required equipment class for DP operations not stated in Table 5-1 should be decided based on a thorough evaluation of operational risks.” It is therefore important to look closely and individually at each module installation/retrieval and include current as a limiting factor for the operation.

3.4 SUBSEA LIFTING

A marine lifting operation shall be planned in accordance to the “safe fail” principle, which states that if any problem occurs the vessel and the lifted object shall stay in a stable controlled condition. There are two main criteria that are decisive for whether or not the lifting operation can be performed within the forecasted weather window:

- Static and dynamic loads on lifted object, cable, crane or other lifting equipment shall not exceed the capacity requirements.
- Snap loads due to slack slings or cable shall be avoided if possible.

Both these criteria must be fulfilled throughout the whole lifting operation according to DNV-RP-H103 [33] sec 4.4.

There are several critical phases that must be considered during the design and planning of operation for subsea structures. Each of these phases is discussed in the following chapters and three of the different stages can be seen in Figure 3.6.

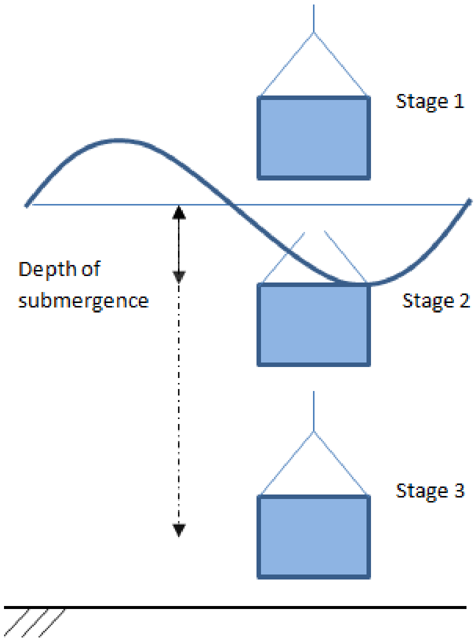

Figure 3.6: Different stages of a lifting operation

In stage 1 the object is mainly exposed to slamming forces, in stage 2 the object is fully submerged below the wave trough where it's still affected by the waves but also by drag and inertia forces. In stage 3 the object is close to the seabed where slamming forces and vertical resonant motion might be affecting the object [25].

3.4.1 LIFTING OFF FROM DECK

When the module is lifted from deck and maneuvered clear of the vessel the following aspects are according to DNV-RP-H103, sec 9.1.1.2 important to consider are:

- Clearance between lifted object and crane boom
- Clearance between crane boom and any other objects
- Clearance between the lifted object and any other object
- Clearance between the underside of the lifted object and grillage/seafastening of the vessel or barge
- Bottom clearance between crane vessel and seabed when lifting in shallow water.

When the lifting operation includes lifts from other vessels the task becomes even more demanding as both vessels will move differently in the waves. They tend to also affect each other's motions just by being in such close proximity, which makes calculating the motions and planning the operation harder.

3.4.2 SPLASH ZONE LIFTING

Lifting analysis of subsea structures determines the maximum allowable design seastate in which safe installation or retrieval can be achieved. Normally, the analysis shows that the larges governing forces are found when lifting the object through the air/water intersection. This is due to the large kinematic water particle forces in the splash zone. When the lifted object enters the water it will experience huge slamming wave forces which could damage the object or exceed the acceptance criteria for the lifting operation. The systems dynamic response is dependent on the hydrodynamic added mass and drag damping forces; these are increasing as the depth of submergence of the object increases. The irregular waves will also cause varying buoyancy so an object suspended in a wave crest will have a substantial uplift due to buoyancy while when the object is in a wave trough there is no buoyancy forces present [34].

Since subsea structures are fitted with different equipment and protective structures they seldom have a regular geometrical shape. It is therefore important to select relevant hydrodynamic coefficients. In this thesis DNV's recommended practice "Modeling and Analysis of Marine Operations" [33] have been used to determine these coefficients and are further discussed in chapter 4.1.3.

3.4.3 LOWERING TO SEABED

The lowering to seabed is considered a relative stable operation where the changes in dynamic forces are more predictable than when going through the splash zone. This is mainly since several hydrodynamic parameters such as added mass and drag force are constant. One scenario that might affect the dynamic forces when lowering a module to the seabed is vertical resonant motion in the lifting wire. Vertical resonant motions in the wire and the lifted object occur when the natural period of the system coincides with the natural period of the crane tip. The natural resonance period of the lifting system increases with the cable length, thus there is a water depth where the crane tip and the lifting system have the same frequency as is shown in Figure 3.7. At this water depth the dynamic forces will increase and might lead to slack line conditions and large dynamic forces in the wire [35].

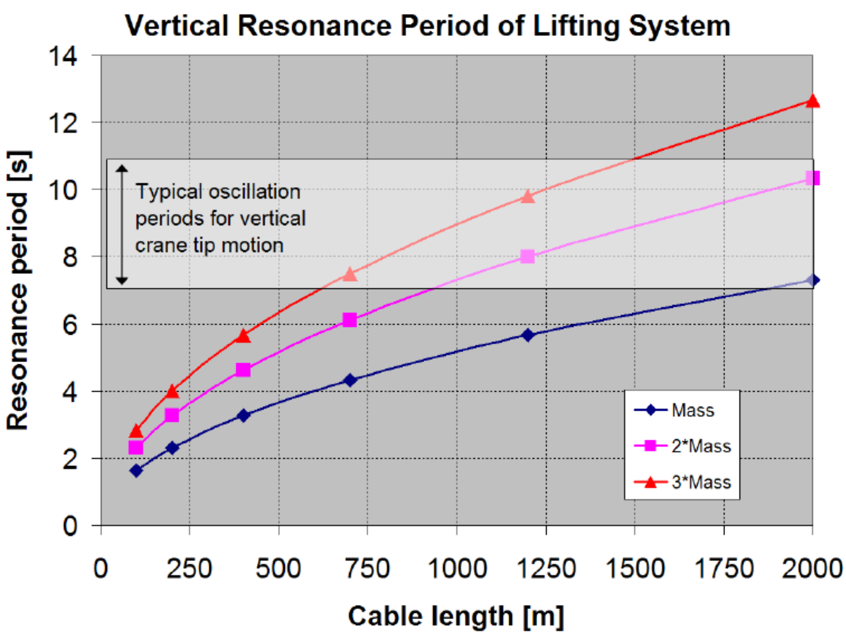

Figure 3.7: The water depth where typical resonant motion occurs [35]

When a standard steel cable is used in a deepwater lift, the weight of the cable becomes a critical parameter as the static weight at the crane tip increases linearly with the cable length. This means that the allowable dynamic load on the crane (and cable) decreases as the lowering depth of structure increases, thus the operation phase when the structure is close to the seabed may be the governing operation phase.

3.4.4 LANDING AND RETRIVING MODULES ON THE SEABED

Landing the module on the seabed is also seen as a critical phase of the subsea lifting operation. The oscillating motion of the module can cause slamming forces with the seabed or other structures causing damage to the subsea equipment. As the module lands on the seabed the lifting operation changes to a fixed end system, which again can cause oscillating motions. AHC is often used to prevent the motion of the lifted object and thus prevent it from excessive movements [33].

4 INPUT PARAMETER STUDY

This chapter gives an overview of how the needed parameters for both the analysis performed in chapter 5 are found. The chapter also includes the assumptions made- and acceptance criteria related to the analysis. The needed formulas are defined in this chapter, but the calculations can be found in Appendix A.7.

4.1 DYNAMIC ANALYSIS

During an IMR operation where a module has to be replaced, the largest dynamic forces will usually be experienced when lifting through the splash zone thus it is this stage of the operation that limits the weather criteria. Different vessels will have different limiting weather criteria depending on how they are affected by the weather, which is the essence of this analysis. Three different sized vessels are analyzed to evaluate the worst kind of weather they are able to complete the replacement of different modules used for Statoil's Subsea Factory.

The analysis is based on a gradual lowering of modules over the side of different vessels. First the vessel is rotated 22,5° to induce rolling motions as well as heave motions. Then the module and vessel is analyzed in ORCAFLEX at a specified height over the water for 1800 seconds (30 minutes) which is the requirement from DNV [33, sec3.4.3] with a specific T_z. The module is then lowered and the analysis is run again. This continues until the module is fully submerged and no slamming forces are present. Then the T_z is changed and the analysis is done again for the different period. When all the different wave periods have been analyzed the H_s is changed and the analysis is done for another wave height.

4.1.1 ACCEPTANCE CRITERIA

The acceptance criteria for a lifting analysis are there to guarantee that the hoisting system never experiences loads over the save working load (SWL) for the crane. The allowable maximum load in the crane wire is defined as a smaller value than the crane capacity and the lifted objects design capacity. The allowable minimum load should, if possible, be above zero to prevent slack sling. From DNV-RP-H103 the recommendation is that:

$$F_{hyd} \leq 0{,}9 * F_{static} \qquad [N]$$

Where

F_{hyd}	-	characteristic hydrodynamic force $[N]$
F_{static}	-	static weight of object $[N]$

This means there is a 10% safety margin and that the dynamic forces shall never be larger than 90% of the static forces. A second way of evaluating the slack sling criteria is by the use of a dynamic amplification factor (DAF) and making sure the DAF is below 2.0. The DAF can be defined as:

$$DAF = \frac{F_{static} + F_{hyd}}{F_{static}} < 2.0 \quad [-]$$

To limit the maximum load in the crane wire the following formulas are used:

$$F_{static} + F_{hyd} < SWL_{crane} * g \qquad [N]$$
$$F_{static} + F_{hyd} < SWL_{wire} * g \qquad [N]$$

Where

SWL_{crane}	-	safe work load for the crane	$[Te]$
SWL_{wire}	-	safe work load for the wire	$[Te]$
g	-	gravitational acceleration	$[m/s^2]$

4.1.2 ASSUMPTIONS MADE FOR THE ANALYSIS

To be able to perform the analysis some assumptions had to be made and are explained as follows:

- The crane tip is fixed and follows the motion of the vessel which makes the vertical motion the same for the crane and the vessel, furthermore the AHC of the crane is neglected.
- In accordance to DNV the mass coefficient C_M have been defined as $C_m = 1 + C_a$
- The stiffness of the whole crane system is assumed to be equal the stiffness of the crane wire.
- The crane is located close to the mass center in x-direction so the pitch motion becomes as small as possible and can be neglected.

4.1.3 HYDRODYNAMIC LOADS AND COEFFICIENTS

When performing splash a zone analysis in ORCAFLEX, there are several hydrodynamic loads and coefficients that have to be evaluated. The more realistic these coefficients and loads are the better the analysis becomes. The common trend in computerized splash zone analysis is that the computer programs tend to overestimate these coefficients compared to module analysis in e.g. a wave tank [34].

4.1.3.1 ADDED MASS

A conservative method of evaluating the added mass for a fully submerged object is by adding the volume of half a sphere on top and bottom of the object described in DNV-RP-H101 Appendix A, Table A-2. When lifting through the waterline only the bottom part of the added mass is present, but as the object becomes fully submerged the added mass on the top of the object must also be evaluated which can be seen in Figure 4.1. [33]

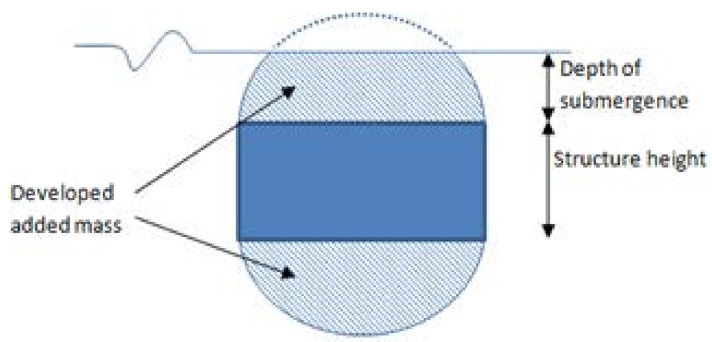

Figure 4.1: Estimated added mass depending on level of submergence

There will also be a contribution to the added mass from the sides of the object. DNV-RP-H101 sec. 4.6.3.3 gives the following formulas:

$$A_{33} \approx \left[1 + \sqrt{\frac{1-\lambda^2}{2(1-\lambda^2)}} \right] * A_{33o} \qquad [kg]$$

$$\lambda = \frac{\sqrt{A_p}}{h_A + \sqrt{A_p}} \qquad [-]$$

Where

A_{33o}	-	added mass for a flat rectangular plate with shape equal to the horizontal projected area of object (found in DNV-RP-H205, Table A-2) $[kg]$
h_A	-	height of submergence $[m]$
A_p	-	area of submerged part of object projected on a horizontal plane $[m^2]$

As the modules are lowered through the splash zone the λ value will decrease while the total added mass increases, as can be seen in Figure 4.2. After the entire module is below the water surface the added mass will start to rapidly increase due to a second half circle seen above the module in Figure 4.1. The formula [36] describing the added mass for the second circle is fairly complicated as part of the circle is missing, but it can be written as:

$$A_{33,top} = \rho_{sw} * C_A \frac{\pi}{2} * R^2 - A_C \qquad [kg]$$

And

$$A_C = R^2 * \cos^{-1}\left(\frac{R-h_c}{R}\right) - (R-h_c) * \sqrt{2Rh - h_c^2} \qquad [m^2]$$

Where

$A_{33,top}$	-	added mass on top of object $[kg]$
A_C	-	missing area of the half circle from Figure 4.1 $[m^2]$
ρ_{sw}	-	density of seawater $[kg/m^3]$
C_A	-	added mass coefficient $[-]$
R	-	half the width of the module $[m]$
h_c	-	height of the missing partial circle from Figure 4.1 $[m]$

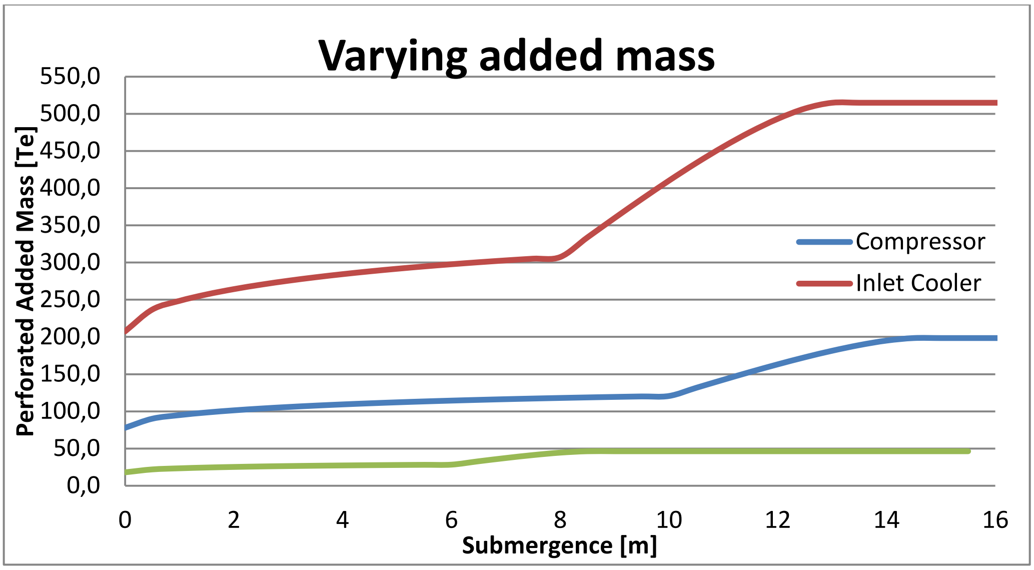

Figure 4.2: How the added mass for the different modules changes with submergence in the waterline

The increase in the added mass once the module is below the waterline is much larger than the increase while the side of the object is being submerged. The increase of the added mass after submergence might affect the splash zone lifting operation and is why the ORCAFLEX analysis checks the effective wire tension for the module when it is fully submerged with maximum added mass.

The subsea modules are not perfect prisms, but rather a combination of pipes and beams welded together. To account for this when applying added mass for the structures a perforation rate is used. To estimate how this perforation rate affects the added mass for the subsea module DNV-RP-H103 sec. 4.6.4.1 is used [33].

The perforation of the different modules is estimated based on visual inspection and shown in Table 4.4. The perforation will reduce the added mass since water can move freely through the object, but according to DNV-RP-H101 the recommended guidance given in sec. 4.6.4.1 will in most cases be overly conservative and overestimate the added mass [33].

4.1.3.2 DRAG AND INERTIA

When the object passes through the splash zone there are several hydrodynamic loads affecting the object. One way of describing the dynamic loads are by the use of the slamming force together with the Morison equation. The Morison equation is a sum of the inertia force and the drag force the lifted object experiences and is given as [25]:

$$F(t) = \rho_{sw} C_m V \dot{u} + \frac{1}{2} \rho_{sw} C_d A |u| u \qquad [N]$$

where

$F(t)$	-	total inline force on the object $[N]$
$\dot{u}$	-	flow acceleration, i.e. the time derivative of the flow velocity $u(t)$ $[m/s^2]$
ρ_{sw}	-	density of seawater$[kg/m^3]$
C_m	-	inertia coefficient given as $C_m = 1 + C_a$ $[-]$
V	-	volume of the body $[m^3]$
C_d	-	drag coefficient $[-]$
A	-	area of the body in the flow direction $[m^2]$
u	-	the flow velocity $[m/s]$

The first part of the formula above represents the inertia force on the object which are caused by the water that is forced away to make room for the object when it's lowered through the waterline. The inertia forces are dependent on the submerged volume of the object and will increase as the object is lowered through the splash zone.

The second part of the Morison equation represents the drag force. There are mainly two contributors to drag force; friction force between the object and the fluid, and the pressure differential upstream/downstream of the object. The drag coefficient of a complex subsea module may be fairly hard to estimate where the most accurate method for defining it is to carry out scaled model tests for the object. It is recommended in DNV-RP-H103 sec 4.6.2.4 that unless computerized fluid dynamic (CFD) studies or model tests have been performed the drag coefficient should be in the magnitude of at least 2.5.

4.1.3.3 SLAMMING

As previously mentioned the water entry is a crucial for any subsea lifting operation. Estimations and analysis of the hydrodynamic loadings in the splash zone is a tedious and complex task since several factors such added mass, drag forces, buoyance forces and water impact forces are constantly varying [34].

One way to estimate where the largest hydrodynamic forces are found, is to do a stepwise lowering analysis and simulation of the modules from air to fully submergence. The modules are kept in a fixed position while subjected to the estimated wave spectrum for the area of the operation. The module is kept at this elevation for 30 min as is requested by DNV sec. 3.4.3.7. The module is the subjected to different sea states where both the H_s and T_z are changed. The result of such an analysis can be found in chapter 5.1. The characteristic impact forces on the module are called slamming forces and can, according to DNV-RP-H103 sec 3.2.9, be defined as follows [33]:

$$F_s(t) = \frac{1}{2}\rho_{sw} C_s A_p v_s^2 \qquad [N]$$

Where

$F_s(t)$	-	total slamming force on the object $[N]$
ρ_{sw}	-	density of seawater $[kg/m^3]$
C_s	-	slam coefficient $[-]$
A_p	-	horizontal projected area of the object $[m^2]$
v_s	-	slamming impact velocity $[m/s]$

According to DNV-RP-H103 sec 3.4.2.20, the slam coefficient should not be taken as less than 5.0 for any of the complex structures defined in chapter 2.3. The slamming impact velocity is a combination of all the velocities the module experiences and can be written as [33]:

$$v_s = v_c + \sqrt{v_{ct}^2 + v_w^2} \qquad [{}^{m}/{}_{s}]$$

Where

v_c	-	module pay-out velocity $[m/s]$
v_{ct}	-	vertical velocity of the crane tip $[m/s]$
v_w	-	vertical water particle velocity $[m/s]$

In other words the slamming force is dependent on two things; the horizontal projected area of the object and the velocity this area hits the water with. The largest slamming forces are therefore expected to be present for short wave periods where the vertical wave particles have the largest velocities. The slamming forces are also expected to be largest when the module is close to or slightly submerged in the ocean.

4.1.4 ORCAFLEX MODEL

The finite element program ORCAFLEX has been used in this analysis both for modelling of the different components including the vessels, as well as computing the different forces on the modules. The vessels have been dimensioned and based on the vessel found in ORCAFLEX since no specific response amplitude operators (RAO) is known. ORCAFLEX is able to scale it’s built in RAO data to fit for different vessels and this is used for the modeling of the three vessels used in the simulations. The wire between the vessel crane and the module is simulated by the line function and for the different modules a 6D buoy is used.

4.1.4.1 IMR VESSELS

There is a standard vessel in ORCAFLEX, however the dimensions of this vessel doesn't fit the average dimensions of IMR vessels so these have been slightly modified. The standard vessel is originally based on a tank ship [14] with a length of 103m and a weight of 9018Te.

The dimensions of the vessels used in this thesis can be seen in Table 4.1 and a sketch of the vessel is seen in Figure 4.4 and Figure 4.4. The data used for vessel A, B and C is based on the IMR vessel portfolio found in Appendix A.4 and the scaling process defined in ORCAFLEX. From the ORCAFLEX manual the scaling process is defined as follows [14]: "The scaling is based on Froude scaling which scales each item of data by a factor that depends on the unit of that item. If R = ratio of vessel type length, then the scaling factor is applied as follows:

- All lengths are scaled by R
- All masses (and added masses) are scaled by R^3
- All times are scaled by $R^{0,5}$"

These scaling rules are the same as those used in deriving full scale ship performance from physical model tests and are correct if the vessel is a perfect scaled replica of the vessel type in all respects.

Table 4.1: Dimensions of the vessels used in the simulation

	Vessel A	Vessel B	Vessel C
Weight [Te]	6500	9500	16000
Length, L [m]	90	110	145
L1 [m]	46,3	56,6	74,6
L2 [m]	43,7	53,4	53,4
Width, W [m]	18	22	30
Height, H [m]	9	11	14

For the simulation a vessel heading of 22,5° is added in order to impose roll motions on the vessels, as is listed as a recommendation in DNV-RP-H103 sec. 7.3.9 for doing lifting analysis. The vessel primary motion is set to *none* while the vessels superimposed motion is set to "*displacement RAOs and harmonic motions*". The statics calculations are set to "*6 Degrees of freedom*" and the only included effect is "*added mass and damping*". No other settings are changed for any of the vessels.

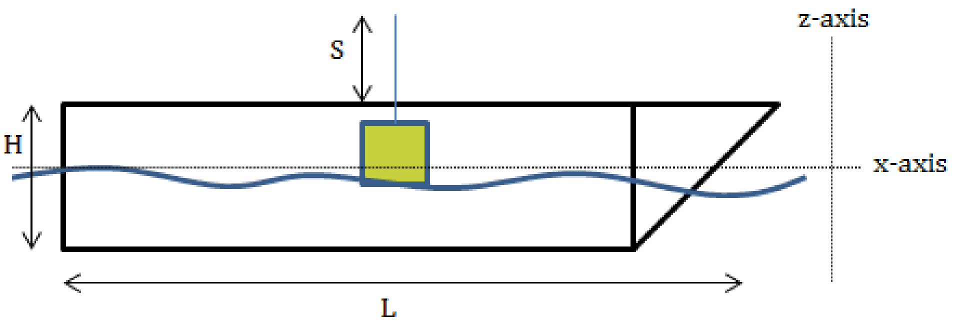

Figure 4.3: A sketch of the vessels used in the ORCAFLEX analysis as seen from the side

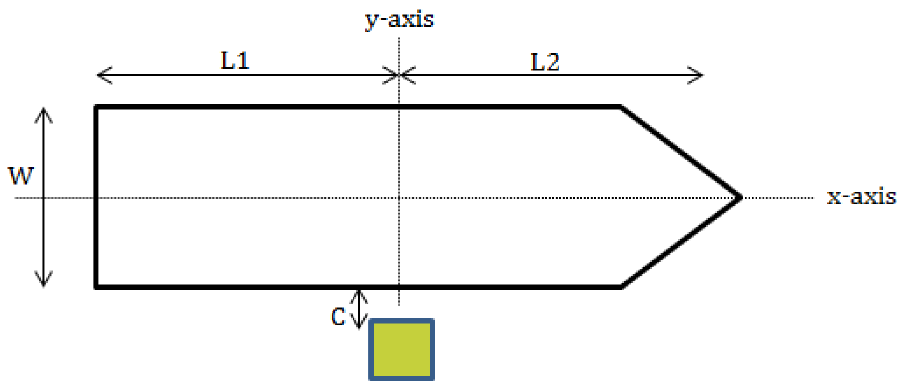

Figure 4.4: A sketch of the vessels used in the ORCAFLEX analysis as seen from above

From the sketches in Figure 4.4 and Figure 4.4 it's seen that there are two variables not defined in Table 4.1, this is the clearance, C and the origin of the crane tip, S. The clearance is defined as half the modules width, while the crane tip is defined as 12m above deck.

Since it is assumed that the crane tip has an infinite stiffness the crane tip motion will be equal to the motion of the vessel where the crane is located. In other words it will be solely based on the RAO curve of the vessel. As roll and pitch motion is dependent on the length from the centerline of the vessel the crane tip motion transfer function can be written as [25]:

$$Vertical\ crane\ tip\ motion = RAO_{heave} + y * \sin(RAO_{roll}) \quad [m/m]$$

Where

RAO_{heave}	-	vessel vertical displacement in heave $[m/m]$
y	-	distance from the point of origin for the roll motion to the crane tip $[m]$
RAO_{roll}	-	vessel roll angle $[deg/m]$

The analysis is based on the crane being located on the side of the vessel for an over the side lifting operation. The cranes position is so that the pitch motion can be neglected, i.e. it is located close to the centerline where the pitch motion has its origin. In ORCAFLEX this is simulated by the line function being located close to the center of the vessel.

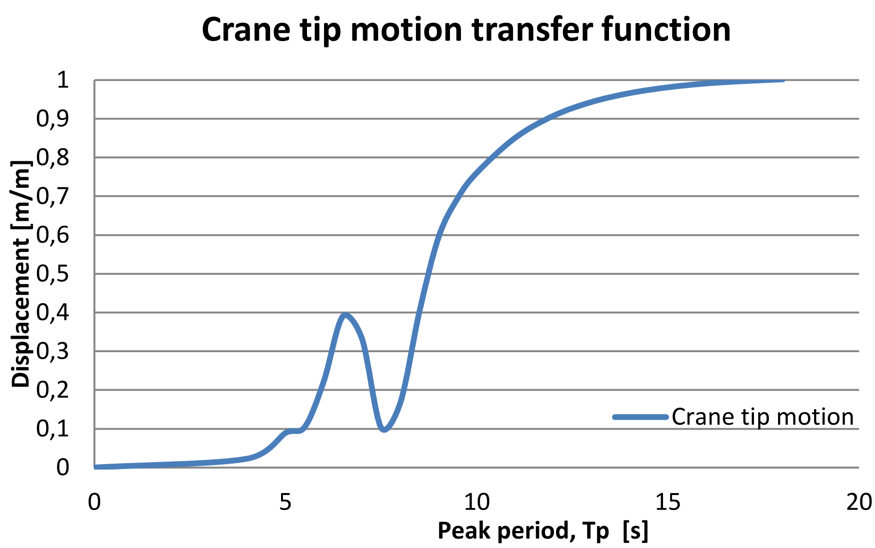

Figure 4.5: Crane tip motion for vessel A

The RAO data for the different vessels are collected from ORCAFLEX with a heading of 22,5°. The crane tip motion transfer function is calculated and shown in Figure 4.5 based on the tables shown in Appendix A.7. In the analysis the heading is chosen in such a way that the module is hit by the waves first and not the vessel, this prevents the vessel from damping the waves before the module is hit and it's the worst case scenario is analyzed.

The graph shown in Figure 4.5 shows how vessel A will react to wave motions and it can be seen that there is a peak period around 6 to 7 seconds, so it is expected that waves with these or shorter periods will cause large crane tip motions and thus large slamming forces for any module lifted in these waves. The analysis done in ORCAFLEX is based on the zero up-crossing periods, T_z and from ORCAFLEX it's possible to find the corresponding T_P values.

Table 4.2: Zero up crossing periods with corresponding peak periods from ORCAFLEX

T_z	T_P
4	5,15
5	6,43
6	7,72
7	9,00
8	10,29
9	11,58
10	12,86
11	14,15
12	15,44
13	16,72

From Table 4.2 one can see that the critical peak periods from 5-7 second corresponds to a zero up-crossing period around 4-6 seconds and these periods are the ones where the crane tip motion will be largest thus the slamming forces must be evaluated for these periods.

4.1.4.2 LIFTING WIRE

The hoisting system has a capacity of 400Te, so the wire that is used for this crane will have a minimum break load (MBL) close to the capacity of the crane. To find the wire with sufficient MBL the following formula is used:

$$F_{sling} < \frac{MBL_{sling}}{\gamma_{sf}} \qquad [N]$$

Where

F_{sling}	-	is the capacity of the sling $[N]$
MBL_{sling}	-	minimum break load of the sling $[N]$
γ_{sf}	-	safety factor $[-]$

The safety factor should always be larger than 3 [35] and for these calculations, γ_{sf} is chosen to 3.1. This gives a wire with the specifications found in Table 4.3.

Table 4.3: Wire specifications [37]

$\boldsymbol{N2}^{©}$ Crane Ropes Non-Rotating 55x K26WS Compacted	
Nominal diameter	$127\ [mm]$
Effective diameter	$107{,}9\ [mm]$
MBL	$13434\ [kN]$
Maximum lifting force	$4333\ [kN]$
Weight in air	$7{,}934 * 10^{-3}\ [Te/m]$
Metallic Area	$9139\ [mm^2]$

ORCAFLEX has two methods for connecting the buoy to the vessel; *"line function"* and *"winch function".* The reason for choosing the line function in this thesis is that this function models the weight of the wire, while the winch function connects the structures with a massless string. The line function can also be used to model a pipe, so it's possible to set inner and outer diameter. In the *"geometry and mass"* tab, the outer/inner diameter and weight per unit length is set according to Table 4.3, while under the *"structure"* tab the axial stiffness of the hoisting system is set. No other parameters were changed.

The line is connected to the IMR vessel to simulate a lifting over the side operation, and the buoys relative connection position is below the crane tip. The effective tension in the line near the crane tip is the parameter that is used for the analysis as it is here the largest forces are found.

4.1.4.3 SUBSEA MODULES

The modules that are being analyzed in this thesis are presented in chapter 2.3 and a summary of the data used in given in Table 4.4. The forces the modules are experiencing throughout the operation are described in this chapter. The forces are either calculated with Microsoft Excel or ORCAFLEX, while the formulas used are found in DNV-RP-H103 chapter 3 and 4.

Since the modules are not perfect cubes, the projected areas that are affected by the slamming forces will not be equal to area of the bottom of the modules. The effect of perforation is described in DNV-RP-H103 sec. 4.6.4.

Table 4.4: Data summery for specific modules [9]

	Compressor	Pump	Inlet and anti-surge cooler
Weight [kg]	289000	45000	235000
Dimension [m x m x m]	11 x 9 x 10	5 x 5 x 6	15 x 10 x 8
Perforation [%]	40	25	30
Added mass waterline [kg]	77800	18000	207400
Added mass submerged [kg]	198000	46300	514800
MTTF [years]	7	5	81

From OREDA 2009 the mean time to failure (MTTF) is estimated, but these estimates may be inaccurate because in 2009 the extent of subsea installations and corresponding failure data were scarce and for the compressor non-existing. The OREDA 2009 data may be seen in Appendix A.6. The data for the compressor is in this case based on topside critical failures where the elements not included in the subsea design were removed from the total critical failure rate.

The modeling of the three subsea modules is done in ORCAFLEX by using the *"6D buoy function"*. The 6D buoys dimensions are edited under the *"drawing"* section to resemble the different modules. The drawing in ORCAFLEX is based on nodes and the connection between the different nodes with a local coordinate system located in the center of the drawing. The different modules can be seen in Figure 4.6.

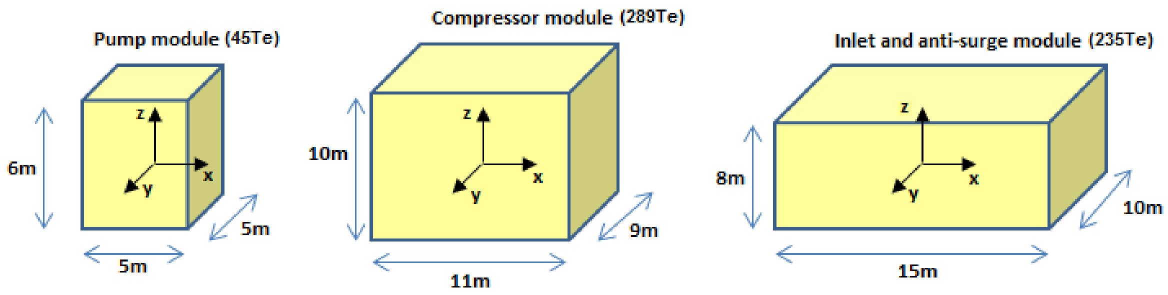

Figure 4.6: The modules that have been drawn and analyzed in ORCAFLEX

The total volume of displaced water and the center of this volume can be altered under the *"properties – geometry"* section, while the *"properties – translation motion"* gives the user the ability to alter the drag area, drag coefficients, hydrodynamic added mass, the added mass coefficients and the inertia coefficients on the 6D buoy. The *"properties – slam"* gives the user the chance to add a slam area for calculation of slamming forces as well as the slam coefficient, C_s.

The *"inertia section"* lets one decide the 6D buoys weight in air as well as the mass motion of inertia for all three directions. In this analysis it is assumed that the center of mass for the different modules is located in in the center of the module (x = 0, y = 0, z = 0).

4.1.4.4 ORCAFLEX INPUT PARAMETERS

In order to perform the ORCAFLEX splash zone analysis a set of parameters are required. The input parameters for the three different modules are presented in the tables below. The calculation of the parameters is given in Appendix A.7.

Table 4.5: Input parameters in ORCAFLEX for the three modules

Parameters	Pump module	Compressor module	Inlet and anti-surge cooler module
Height of object	6m	10m	8m
Mass of object in air	45Te	289Te	235Te
Mass moment of inertia x-direction,($\boldsymbol{I_x}$)	228,75Te*m^2	3212Te*m^2	3212Te*m^2
Mass moment of inertia y-direction, ($\boldsymbol{I_y}$)	228,75Te*m^2	5659Te*m^2	5659Te*m^2
Mass moment of inertia z-direction, ($\boldsymbol{I_z}$)	187,5Te*m^2	6365Te*m^2	6365Te*m^2
Volume of displaced water	5,769m^3	37,05m^3	30,13m^3
Drag area x-direction ($\boldsymbol{A_x}$)	30m^2	90m^2	80m^2
Drag area y-direction ($\boldsymbol{A_y}$)	30m^2	110m^2	120m^2
Drag area z-direction ($\boldsymbol{A_z}$)	25m^2	99m^2	150m^2
Drag coefficient x-, y- and z-direction	2,5	2,5	2,5
Hydrodynamic added mass coefficient x-direction ($\boldsymbol{C_{Ax}}$)	0,63	0,61	0,64
Hydrodynamic added mass coefficient y-direction ($\boldsymbol{C_{Ay}}$)	0,63	0,60	0,74
Hydrodynamic added mass coefficient z-direction ($\boldsymbol{C_{Az}}$)	0,61	0,63	0,69
Hydrodynamic added mass fully submerged x-direction ($\boldsymbol{A_{33x}}$)	84,7Te	180,1Te	262,2Te
Hydrodynamic added mass fully submerged y-direction ($\boldsymbol{A_{33y}}$)	84,7Te	209,9Te	402,1Te
Hydrodynamic added mass fully submerged z-direction($\boldsymbol{A_{33z}}$)	75,0Te	198,5Te	514,8Te
Hydrodynamic inertia coefficient ($\boldsymbol{C_{Mx}}$)	1,63	1,61	1,64
Hydrodynamic inertia coefficient ($\boldsymbol{C_{My}}$)	1,63	1,60	1,74
Hydrodynamic inertia coefficient ($\boldsymbol{C_{Mz}}$)	1,61	1,63	1,69
Hydrodynamic slam coefficient ($\boldsymbol{C_s}$)	5,0	5,0	5,0
Slam area z-direction	27m^2	60m^2	105m^2

4.2 ENVIRONMENTAL ANALYSIS

An environmental condition is defined as natural phenomena that occur randomly, sometimes without warning. The ones that can influence on any offshore operation include waves, wind, current, air and water temperatures, ice and snow falls [38].

4.2.1 ACCEPTANCE CRITERIA

The most common factors that influence any vessels operability is the significant wave height, H_S and wave peak period, T_P. It is therefore important to define the two following criteria when completing IMR operations:

- Defining a criteria for the maximum H_S and T_P
- Defining a minimum weather window for the operation

An IMR operation is classified as either weather restricted- or unrestricted operation based on the time it take to be completed. The difference between restricted and unrestricted operations is the acceptance criteria for the environmental loads. A weather restricted operation has a shorter planned operation time than 72 hours, which is the case for most IMR operations and the weather acceptance criteria is then based on the forecasted weather. According to DNV-OS-H101 (2011), Ch. 4, Sec 4B "The duration of marine operations shall be defined by an operation reference period, T_R:

$$T_R = T_{POP} + T_C \quad [h]$$

Where

T_R	-	operation reference period $[h]$
T_{Pop}	-	planned operation time $[h]$
T_C	-	estimated maximum contingency time $[h]$"

T_{Pop} is assessed based on a detailed schedule of operations while T_C is the maximum contingency time which is meant to cover any uncertainties in the T_{Pop} or other contingencies affecting the operation. A planned intervention where the objective is to retrieve and replace a subsea component could have a T_{Pop} of 12 hours. DNV-OS-H101, Ch. 4 recommends that the T_R shall be at least twice the planned operation period and a T_C of less than 6 hours is generally not accepted. A typical intervention operation will therefore have an operation reference period of.

$$T_R = 12 + 12 = 24 \quad [h]$$

Unrestricted marine operations will base its environmental criteria on extreme value statistics, it's important for these operations to have a way of discontinuing the operation should unfavorable weather occur.

There will be times when it's not possible to do IMR operations due to environmental conditions. To make sure that the safety is never compromised there must be a constant monitoring of the weather as well as some limiting operational environment criteria (OP_{LIM}). These criteria shall, according to DNV-OS-H101, section 4 B600, always be less than[38]:

- The environmental design criteria
- Maximum wind and waves for safe working
- Equipment specified restriction
- Limiting conditions for position keeping systems
- Any limitations defined in HAZID/HAZOP
- Limiting weather conditions for carrying out identified contingency plans.

The wave height is often the limiting environmental condition and is therefore used for the analysis in this thesis. Waves in the ocean are often a combination of different waves causing an irregular sea state.

One way to describe this irregular sea state is by the use of a wave spectrum. A wave spectrum is defined as "the power spectral density function of the vertical sea surface displacement" from DNV-RP-H101 Sec. 2.2.5.1. For the North Sea JONSWAP spectrum is the most accurate. JONSWAP spectrum is characterized by a parameter, γ, which for the North Sea varies from 1- 7 where the average has been found to be $\gamma = 3{,}3$. [33]

The wave peak period, T_P, is an important factor when determining the environmental conditions a vessel can operate in as the RAO curve of the vessels vary greatly with different wave periods. The $T_P$is defined from where the relevant wave energy spectrum's frequency has its maximum while the zero-up-crossing period, T_z, is defined as the average time interval between two successive up-crossings in relation to calm sea level from DNV RP-H101 sec. 2.2.5. [33]

When analyses are performed for different H_s it is important to include a range of different zero crossing periods. DNV-RP-H101 sec 3.4.2.22 gives the following formula:

$$8.9 * \sqrt{\frac{H_S}{g}} < T_z < 13 \qquad [s]$$

Where

H_s	-	Significant wave height
g	-	Gravitational acceleration
T_z	-	Zero-up-crossing wave period

The relevant wave heights that have been used in the analysis in this thesis can then be seen in Table 4.6.

Table 4.6: Relevant periods for JONSWAP spectrum

Significant wave height ($\boldsymbol{H_s}$) [m]	Zero crossing period ($\boldsymbol{T_z}$) [s]
2	4 – 13
2,5	4 – 13
3	5 – 13
3,5	5 – 13
4	6 – 13
4,5	6 – 13

4.2.2 ALPHA FACTOR

Due to uncertainties in both monitoring and forecasting of the environmental conditions a factor of uncertainty also known as a α-factor is chosen. DNV OS-H101 Sec. 4D divides different operations in weather forecast category A,B and C depending on their sensitivity to weather conditions. Typical IMR operations are classified as category B and α-factors for IMR operations can be found in Table 4.7 and it's shown that the α-factor is calculated based on the planned operation time only.

Table 4.7: Level B alpha factors for waves [38]

Operational period [h]	Design Wave Height [m]						
	$H_S = 1$	$1 < H_S < 2$	$H_S = 2$	$2 < H_S < 4$	$H_S = 4$	$4 < H_S < 6$	$H_S \geq 6$
$\boldsymbol{T_{POP}} \leq 12$	0,68	Linear Interpolation	0,80	Linear Interpolation	0,83	Linear Interpolation	0,88
$\boldsymbol{T_{POP}} \leq 24$	0,66		0,77		0,80		0,86
$\boldsymbol{T_{POP}} \leq 36$	0,65		0,75		0,77		0,84
$\boldsymbol{T_{POP}} \leq 48$	0,63		0,71		0,75		0,81
$\boldsymbol{T_{POP}} \leq 72$	0,58		0,66		0,69		0,79

For H_S values other than the ones found in Table 4.7 linear interpolation is used for finding the α-factors.

$$p(x) = f(x_0) + \frac{f(x_1) - f(x_0)}{x_1 - x_0}(x - x_0) \quad [-]$$

The operational limiting criteria will then be given as(DNV OS-H101 sec 4B):

$$OP_{WF} = OP_{LIM} \times \alpha \qquad [m]$$

where

OP_{WF} - Operational limiting criteria
OP_{LIM} - Operational environmental limiting criteria
α - Alpha factor

During operations in the arctic i.e. Barents Sea, the α-factor may be considerably lower due the occurrences of polar lows. Polar lows are very hard to forecast as they can occur rapidly with little to no warning. They have a tendency to occur during winter, but may occur all year, bringing heavy snow showers and icing with a constantly changing wind direction so it's hard to weathervane the vessel. The α-factors for The Barents Sea cannot be found in DNV and must be based on the seasonal uncertainties [39]. From the formula above one can see that if the α-factor is close to zero the OP_{WF} will also be close to zero.

4.2.3 WEATHER WINDOW

Another critical factor to consider for the operation is the duration of when the H_S is below the OP_{WF}, also called the "weather window". Data found in Appendix A.2 gives the durations in hours for when the sea state is below specified values for a part of the Norwegian Sea. The table is based on wave data gathered from historical weather maps from the years 1955 – 1995, where data is gathered every 6th hour and linear interpolation is used between the data points. The mean durations show the average expected durations, but one must be aware that there are variations from year to year [40].

It is possible to describe the wave situation in the northern North Sea from the wave data shown in Appendix A.2, Figure A 3, Table A 2. The values found here are inserted as parameters in a Weibull distribution. The Weibull distribution is given as [40]:

$$F_{WH}(d) = 1 - exp\left[-\left(\frac{d}{\alpha}\right)^{\beta}\right]; d \geq 0 \qquad [-]$$

Where

$F_{WH}(d)$ - probability of a given wave height $[-]$
d - the wave height $[m]$
α - scale parameter for the Weibull distribution $[-]$
β - shape parameter for the Weibull distribution $[-]$

The result of this approximation can be seen in Figure 4.7, the red line indicates that 50% of the waves are smaller than 2,1m and 90% of the waves are smaller than 5,1m [40].

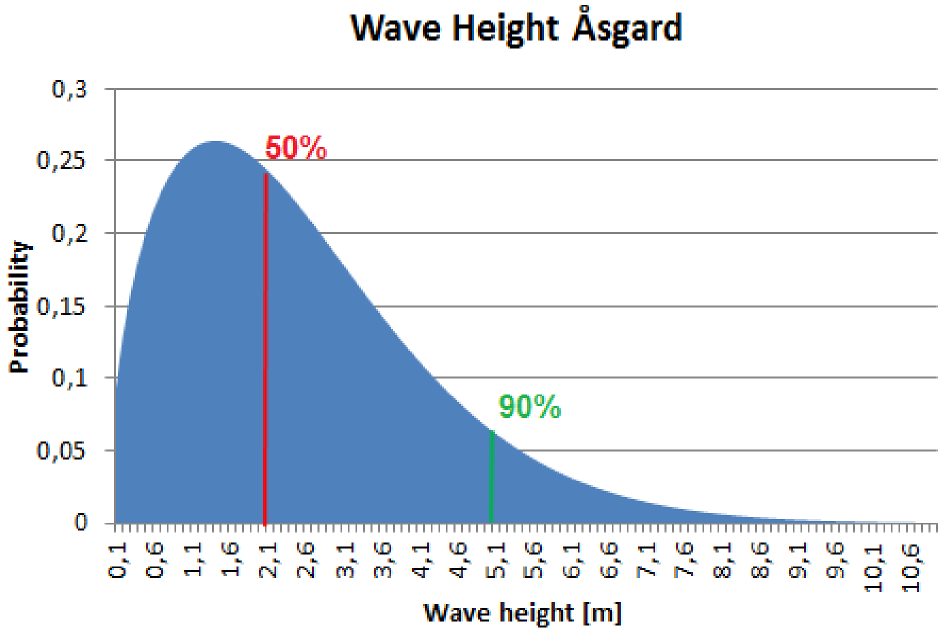

Figure 4.7: The average yearly wave height for the Åsgard area

To be able to describe the duration, D, for any given weather window the Weibull distribution can again be used. This approximation has been proven to represent the duration reasonably well. The Weibull distribution is given as [40]:

$$F_D\,(d) = 1 - exp\left[-\left(\frac{d}{\alpha_D}\right)^{\beta_D}\right]; d \geq 0 \qquad [-]$$

Where

$F_D\,(d)$	-	the probability that duration where the significant wave height is lower than the operational criteria$[-]$
d	-	mean duration of the weather window $[h]$
α	-	scale parameter for the Weibull distribution $[-]$
β	-	shape parameter for the Weibull distribution$[-]$

Using the data sheet from Appendix A.2 it's also possible to create an approximation of the durations when the weather is below a specific wave height and thus finding the weather window.

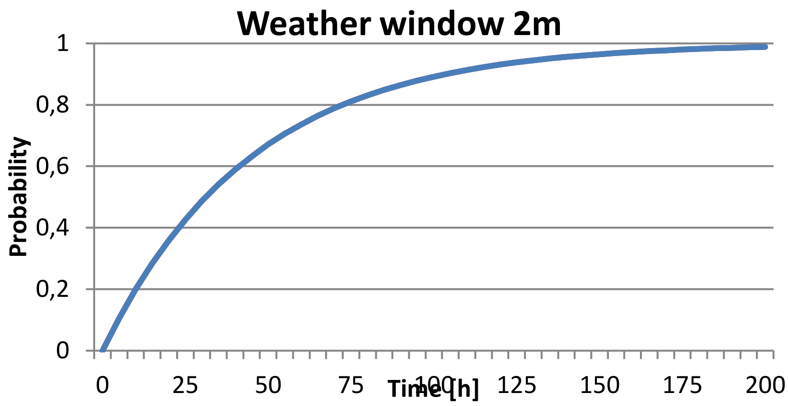

Figure 4.8: The duration of a weather window in February for the Åsgard field

Figure 4.8 shows the probability for a weather window on the Åsgard field in February when the wave height is below 2m. The calculations for the wave heights and the weather window can be seen in chapter 5.2.

5 ANALYSIS

This chapter will consist of a presentation of the results obtained from a lifting operation (chapter 5.1). The operation is performed by three vessels, where three different modules are lifted through the splash zone. Then an environmental study is prepared for the geographical area where the lifting operation takes place (chapter 5.2).

5.1 SPLASH ZONE ANALYSIS

From ORCAFLEX the maximum effective tension in the wire close to the crane tip together with the maximum slamming forces on the bottom of the lifted object is collected and evaluated. The reasoning behind choosing this data to collect is that should the forces in the top of the wire exceed 400Te or 3924 KN, the acceptance criteria from chapter 5.1.1 which states that the total force must be less than the crane capacity will be breached. The other acceptance criteria from the same chapter states that the wire tension shall not be less than 10% of the total static force of the lifted object. The lower minimum tension limit for the different modules is given in Table 5.1. The limiting criteria are represented in the graphs in this chapter as a red line. The maximum static force will vary slightly as the module is more and more submerged. The values used for the acceptance criteria are based on no submergence and therefore slightly conservative.

Table 5.1: Minimum allowable effective tension for the modules

Modules	Minimum effective tension criteria [KN]
Pump module	44,1
Compressor module	283,5
Inlet and anti-surge cooler module	230,5

Table 5.2 and Table 5.3 gives an overview of the results found for the lifting of the compressor module with vessel A when the $H_s = 2.0m$. The Z values represent the height of the bottom of the module, so $Z = -5$ will represent the submergence of half the compressor module. This is an example for the data received from ORCAFLEX that has been processed in Microsoft Excel. The cells in the table show the total tension in the top of the wire, close to the crane tip. The forces are given in KN.

Table 5.2: The maximum tension in the lifting line for vessel A, compressor module lifting in Hs = 2,0m

Vessel A	*Compressor Module Maximum Effective Tension [KN]*										
$H_s = 2,0m$	*Zero up crossing periods, T_z [s]*										
	4	5	6	7	8	9	10	11	12	13	**MAX**
Z = 3	2846	2897	2926	2942	2891	2883	2879	2878	2877	2883	**2942**
Z = 1	3024	2931	2911	2945	2900	2881	2891	2879	2878	2887	**3024**
Z = 0	3529	3224	3136	3094	3015	2905	2880	2877	2874	2885	**3529**
Z = -1	3744	3284	3193	3133	3000	2907	2876	2884	2847	2851	**3744**
Z = -2	3437	3188	3149	3033	2962	2868	2841	2849	2861	2829	**3437**
Z = -5	3041	3057	3009	2926	2905	2837	2784	2750	2755	2722	**3057**
Z = -10	2775	2701	2695	2657	2614	2688	2682	2675	2670	2671	**2775**
Z = -25	2502	2540	2577	2562	2554	2539	2528	2522	2524	2507	**2577**

For the maximum effective tension the cells with the highest values are marked red and gradually as the force diminish the cells changes from red to yellow and then to green for the cells with the lowest maximum tension. For the minimum effective tension the coloring of the cells is reversed where the lowest numerical value of tension is marked red.

Table 5.3: The minimum tension in the lifting line for vessel A, compressor module lifting in Hs = 2,0m

Vessel A			*Compressor Module Minimum Effective Tension [KN]*								
$H_s = 2,0m$			*Zero up crossing periods, T_z [s]*								
	4	5	6	7	8	9	10	11	12	13	**MIN**
Z = 3	2825	2798	2755	2759	2787	2789	2784	2892	2791	2796	**2755**
Z = 1	2523	2719	2778	2749	2773	2788	2786	2790	2790	2795	**2523**
Z = 0	2267	2325	2516	2508	2617	2732	2757	2761	2772	2779	**2267**
Z = -1	2113	1854	2471	2493	2516	2657	2688	2672	2721	2718	**1854**
Z = -2	2254	2131	2493	2525	2520	2629	2650	2647	2644	2645	**2131**
Z = -5	2334	2205	2284	2376	2392	2473	2514	2503	2531	2550	**2205**
Z = -10	2248	2181	2200	2226	2251	2268	2295	2312	2325	2355	**2181**
Z = -25	2413	2388	2351	2356	2365	2391	2404	2410	2403	2423	**2351**

From the tables for the three vessels in varying wave heights with the different modules the maximum and minimum values (the row furthest to the right in Table 5.2 and Table 5.3) are collected and put in Appendix A.7. The result from these tables can be seen in the different graphs and matrixes throughout chapter.

5.1.1 PUMP MODULE

The pump module's weight in air is 45Te thus the limiting criteria for this module will be based on slack sling. To prevent slack sling in the cable the lowest effective tension allowed in the cable is from Table 5.1 found to be 44.1 KN and is marked in Figure 5.1 by a red line. The calculations were first done for the vessel A since this is the smallest vessel. Once the limiting criteria were found for this vessel, the next smallest vessel, Vessel B was analyzed and the effective tension in the wires for both vessels can be seen in Figure 5.1. Since it was shown that vessel B was able to install the pump module in $H_s = 4{,}5m$, it was assumed that vessel C would also be able to do this installation based on the fact that a larger vessel will have less motion and thus less impact forces affecting the splash zone lifting operation.

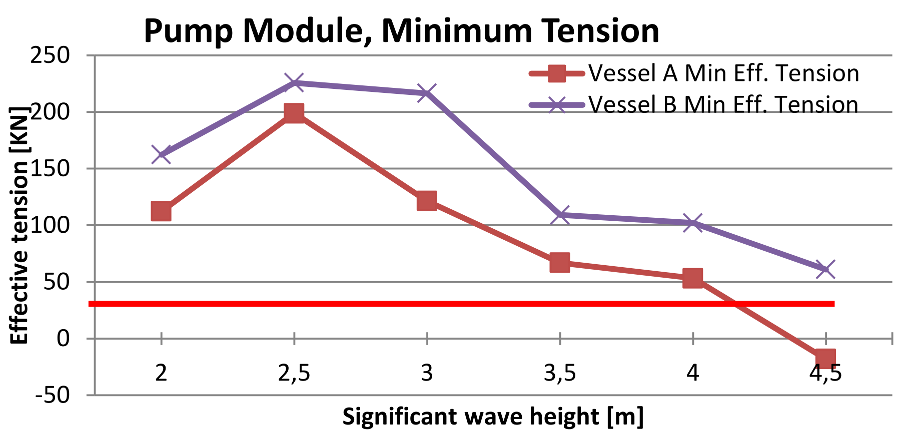

Figure 5.1: The limiting wave height and minimum effective tension when lifting the pump module.

The maximum effective tension the wire experiences when lifting the pump module has also been analyzed in order to find the smallest crane which can do the installation in the varying wave heights. A 120Te crane is represented in Figure 5.2 by a light blue stapled line while a 100Te crane can be seen as a stapled red line.

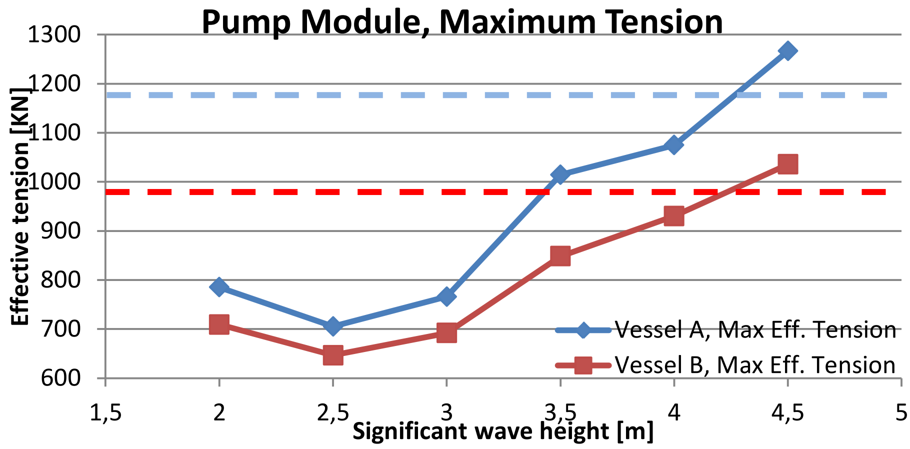

Figure 5.2: Maximum effective tension in the wire lifting the pump module in varying wave heights

5.1.2 COMPRESSOR MODULE

The compressor module weighs 289Te and is the heaviest module that is analyzed in this thesis. The lowest allowable effective tension in the wire is from Table 5.1 found to be 283,5KN and is marked in Figure 5.3 and as a red line. The other limiting criterion is the total capacity of the wire and crane which is taken as 400Te. This means that a dynamic force of 1089 KN is the absolute maximum the crane can handle. The minimum effective tension results from ORCAFLEX can be seen in Figure 5.3.

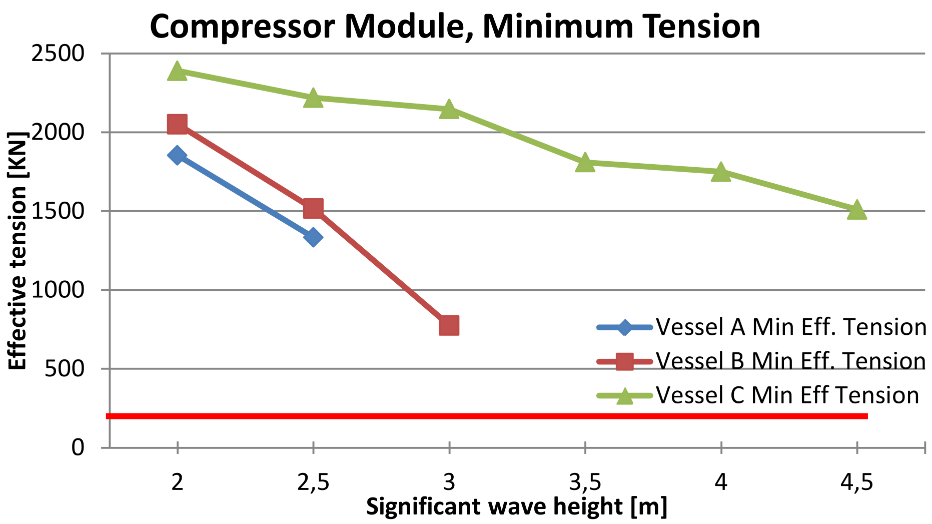

Figure 5.3: The minimum effective tension for the vessels lifting the compressor module

The maximum allowable effective tension in the wire when lifting the compressor module has also been analyzed in ORCAFLEX for the different wave heights and the results can be seen in Figure 5.4. Once the seastate where the maximum effective tension exceeds the capacity of the crane and wire is found, further analysis of this vessel is stopped. This is the reason why vessel A only has two data points in Figure 5.4 while for vessel C there are five data points.

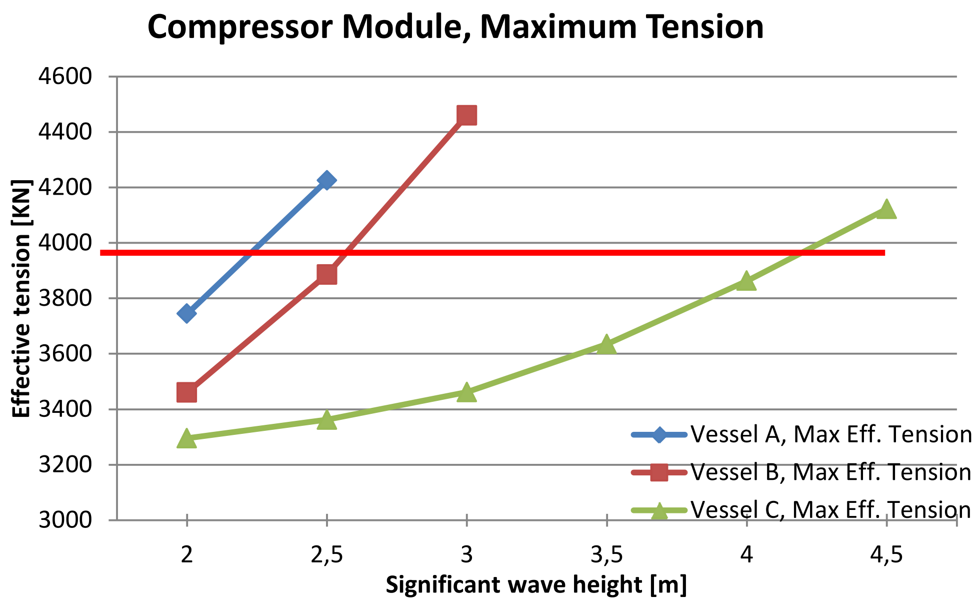

Figure 5.4: The maximum effective tension in the wire when lifting the compressor module in varying wave heights

5.1.3 INLET AND ANTI-SURGE COOLER MODULE

The inlet and anti-surge cooler module has a weight of 235Te and is the largest module analyzed in this thesis. Due to its large water plane area and thus large slamming forces it is a critical component to analyze in regards to splash zone lifting, even though it does not weight as much, or has as short MTTF as the compressor module. The minimum allowable effective tension in the lifting wire is from Table 5.1 defined as 230,5KN.

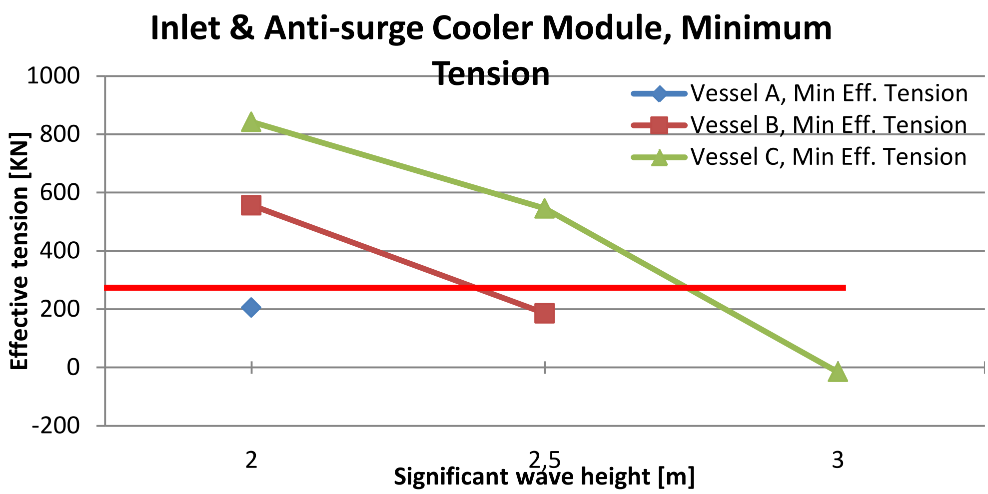

Figure 5.5: The minimum effective tension in the wire when lifting the anti-surge module through the splash zone

The minimum and maximum effective tension based on the calculations from ORCAFLEX can be found respectively in Figure 5.5 and Figure 5.6. Like the other modules, once the analysis shows that a vessel cannot lift the inlet and anti-surge cooler module in a specific wave height, the analysis for that vessel stops. This is the reason why there are so few data points in these graphs.

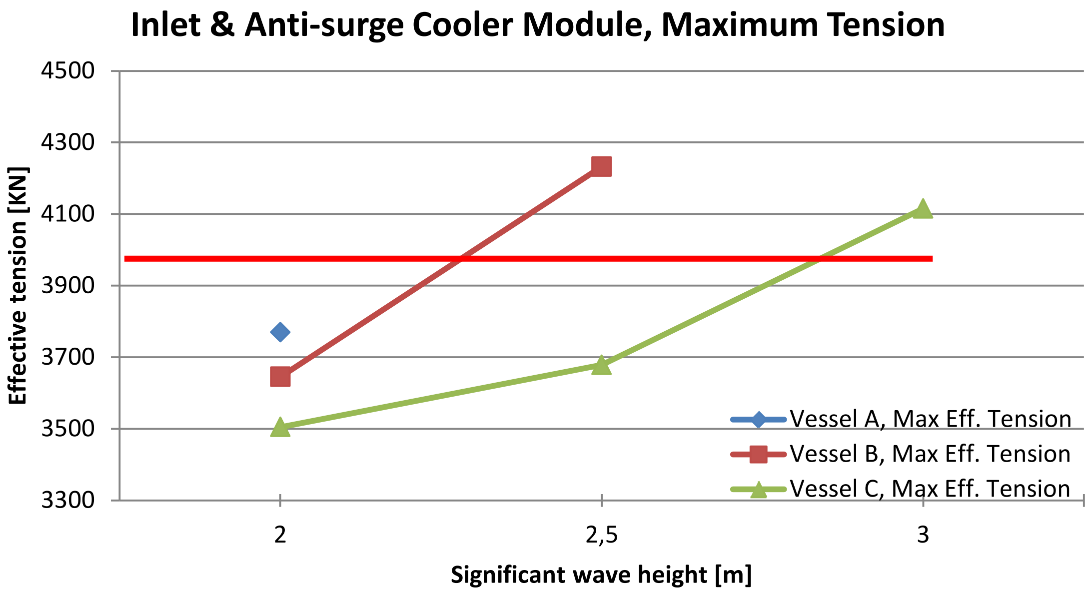

Figure 5.6: The maximum effective tension in the wire when lifting the anti-surge module through the splash zone

5.1.4 SUMMARY

From the results from graphs in this chapter it is possible to gather information about what sea states the three vessels are able to lift the three different modules through the splash zone. This information is based on results from ORCAFLEX together with the acceptance criteria defined in chapter 5.1.1. The results from the graphs are gathered together in a matrix shown as Table 5.4.

Table 5.4: An overview of the significant wave heights where the vessels can operate the different modules

Summary			
	Vessel A	*Vessel B*	*Vessel C*
Pump module	*4,0m*	*4,5m*	*4,5m*
Compressor module	*2,0m*	*2,5m*	*4,0m*
Inlet & anti-surge cooler module	*N/A*	*2,0m*	*2,5m*

These results are based on the wave period which causes the highest effective tension in the wire. These wave periods are critical since they coincide with the peak in the vessels RAO curves discussed in chapter 4.1.4.1.

5.2 EVALUATION OF WEATHER WINDOW

If it's assumed a module has failed and it needs to be retrieved and replaced. There are three available vessels that can complete this unplanned marine operation. Vessel A has an Operational limit(OP_{LIM}) of H_s = 2,5m, vessel B has an OP_{LIM} of H_s = 4,0m while vessel C has an OP_{LIM} of H_s = 4,5m. The operation has a T_{POP} of 12hours and the entire operation is assumed to take 24hours including contingencies. In the following example one of the Weibull parameters, β_D, is assumed to be set to 1, this then gives us the exponential distribution and α_D becomes the mean duration when H_S is lower than then operational criteria (OP_{WF}). The distribution is then given as:

$$P(x) = exp\left[-\frac{D}{d_{avg}}\right], d_{avg} > 0 \qquad [-]$$

Where

$P(x)$	-	The probability that the operation time is smaller than the weather window $[-]$
D	-	The length of the operation (Duration) $[h]$
d_{avg}	-	Mean duration when significant wave height is less than the relevant limit $[h]$

The α-factor is found by linear interpolation for the different vessels as shown in chapter 4.2. Vessel A has an OP_{WF} of H_s = 2,1m, vessel B can work in waves as high as H_s = 3,8m, while vessel C is limited to waves of H_s = 3,8m. Using the data from Appendix A.2 it's possible to calculate and compare the probabilities of when the two vessels would be able to commence the operation as well as figuring out if the weather window will be large enough to complete the operation.

Table 5.5: Probability of the weather window being large enough for a 24hour operation

	Vessel C $(\boldsymbol{OP_{WF} = 3,8m})$		Vessel B $(\boldsymbol{OP_{WF} = 3,2m})$		Vessel A $(\boldsymbol{OP_{WF} = 2,1m})$	
Month	Average [h]	Probability	Average [h]	Probability	Average [h]	Probability
Jan	104	0,794	81	0,743	45,1	0,587
Feb	130	0,832	93	0,772	48,5	0,610
Mar	166	0,866	116	0,813	58,5	0,664
Apr	362	0,936	208	0,891	77,7	0,734
May	672	0,965	384	0,939	122,4	0,822
Jun	898	0,974	506	0,954	138,4	0,841
Jul	875	0,943	557	0,958	167,5	0,867
Aug	548	0,957	382	0,939	150,6	0,853
Sep	227	0,900	151	0,853	76,5	0,731
Oct	122	0,822	87	0,758	54,3	0,643
Nov	101	0,789	73	0,719	40,5	0,552
Dec	76	0,728	62	0,681	40	0,549
Yearly	185	0,878	141	0,843	67,5	0,700

The probabilities in Table 5.5 are the probabilities that the operation will have a weather window that is large enough for a 24hour operation for the three vessels with different operational limits. The values marked with average are the average weather window available for the given month. The results are also combined in a graph shown in Figure 5.7.

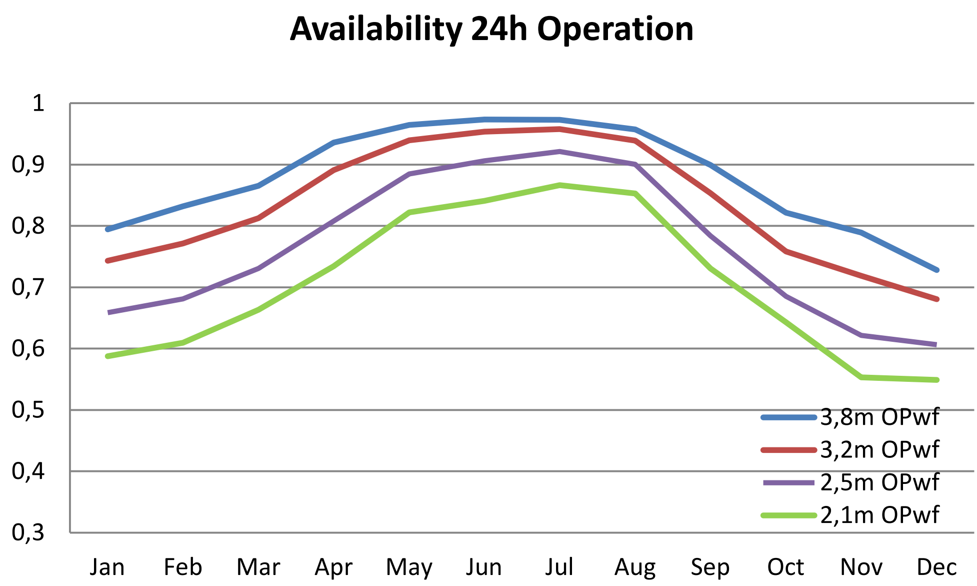

Figure 5.7: The availability for different operational criteria during a 24h operation

The probability of a random critical failure in the module discussed in the section above is evenly distributed throughout the year so the vessels must be able to complete the operation during the winter season. The distribution of the highest weather conditions, $X^{(n)}$, of n identically distributed variables, *X,* which in turn is statistically independent, can be written as[40]:

$$F_{X^{(n)}}(x) = \left[1 - \exp\left(-\left(\frac{OP_{time}}{d_{avg}}\right)\right)\right]^n \qquad [-]$$

Where

$F_{X^{(n)}}(x)$	-	The probability of failing the operation for a n number of events $[-]$
OP_{time}	-	The operation time $[h]$
d_{avg}	-	The average time the wave height is below the limiting wave height $[h]$
n	-	The number of events $[-]$

The number of events is found from the data sheet in appendix A.2 and together with the formula above it's possible to find the probability of completing the operation during a specific season as is shown in Table 5.6.

Table 5.6: The probability of not being able to complete the unplanned operation during the winter months

Month	Vessel A $(OP_{WF} = 2,1m)$ Probability of failure	Vessel B $(OP_{WF} = 3,2m)$ Probability of failure	Vessel C $(OP_{WF} = 3,8m)$ Probability of failure
Nov	0,034	0,0006	0,0001
Dec	0,065	0,0013	0,0005
Jan	0,042	0,0010	0,0003
Feb	0,024	0,0011	0,0003
Winter season	$2,21*10^{-6}$	$8,03*10^{-13}$	$4,52*10^{-15}$

No matter which vessel is used for the operation, all vessels will be able to complete the unplanned retrieval and installation during the winter season. Vessel C and vessel B is almost guaranteed to complete the operation during the first month.

6 CONCLUSIONS

This chapter will discuss the results from the analyses completed in the previous chapter and combine this with the information developed and described in this thesis. Conclusions will be drawn on the bases on the results achieved and the educated assumptions implemented throughout this thesis.

6.1 DISCUSSION

From the ORCAFLEX analysis one notices that the largest dynamic forces in the wire is found when the modules bottom is located from Z = 0 to Z = -2. The forces decrease until the object is fully submerged and then peaks again when the top of the module passes the waterline. This is true for all the three vessels analyzed in this thesis and can also be observed in Table 5.2 and Table 5.3 as well in Appendix A.7. The results from both the dynamic analysis and the weather conditions at Åsgard can be seen in Table 6.1. The significant wave heights that are discussed in this chapter will represent the highest seastate the vessel can operate in.

Since there are always uncertainties with weather forecasts an operation will not be started when the waves are too close to the maximum significant wave height, the alpha factor is multiplied with the significant wave height to give the operational criteria (OP_{WF}), this will be the limiting criteria for starting a lifting operation and an overview for all vessels and modules can be seen in Table 6.1.

The vessels used for the analysis cannot be called top of the notch IMR vessels and heave compensating and anti-roll systems have not been accounted for in this analysis. In reality the hull for IMR vessels is designed to withstand vessel motions, bilge keels, bulge bows and X-Bow®s are design examples mentioned in this thesis which reduces vessel motion. The ORCAFLEX ship model does not include these designs, so the results gained from the dynamic analysis may be conservative. In reality the modules would be connected to the special handling system (SHS) tower, where guide wires would keep the module in place, this is another aspect the analysis have not considered.

Pump module

The pump module can be handled by the smallest vessel (vessel A) in waves up to and including $H_s = 4{,}0m$ before there is a danger of slack sling. The other vessels have no problem lifting the pump module, even in $H_s = 4{,}5m$ there is enough effective tension in the line to prevent slack sling.

As seen in Appendix A.4 the typical safe work load (SWL) for the main crane on one of the smaller IMR vessels is around 100-120Te. So for a small vessel with a 100Te SWL crane the pump module can be moved through the splash zone when the $H_s = 3{,}0m$ and $H_s = 4{,}0m$ if the SWL is 120Te. For the medium sized IMR vessel (vessel B) a 100Te crane can work in sea states up to and including $H_s = 4{,}0m$ and a 120Te crane can lift the module in all the sea states evaluated in this thesis. The largest vessel (vessel C) has no problem installing the pump module in any of the evaluated sea states.

With a mean time to failure (MTTF) of 5 years, the pump module will be a component that will require much IMR work. It is also assumed that there will be a large quantity of pump modules on any subsea factory as all of the process components require pump modules either to boost, separate, compress or inject. The fact that even the smallest vessel is able to do complete installation and retrieval of the pump module in sea states as high as $H_s = 4{,}0m$, (with the α-factor included this gives an $OP_{LIM} = 3{,}2m$) this gives according to Table 5.5 a yearly average availability of 84,5%. Pumping systems are commonly used subsea and the condition monitoring of the pumps are constantly improving thus optimizing the maintenance.

Compressor module

The compressor module is the heaviest module analyzed in this thesis, the water plane area isn't especially large though and the analysis shows that there is never any danger of slack sling, with 153Te as the minimum effective tension in the line. The capacity of the crane is the limiting factor and due to the large vertical motions in the waves with the shortest periods, typically a T_z of 4-6s, vessel A and B have trouble lifting the module without causing large slamming forces. Vessel A is limited by $H_s = 2{,}0m$ while Vessel B can lift the compressor until the waves are larger than $H_s = 2{,}5m$. If the α-factor is included, vessel A will only be able to work in $H_s = 1{,}6m$ while vessel B can work in $H_s = 2{,}0m$. The response amplitude operator (RAO) for vessel C doesn't peak in the waves with the shortest periods thus there is no resonant motion for this vessel in these waves. This is important since it is the shortest waves are the waves where the vertical wave particles and the slam forces are largest. This is why in Figure 5.4 one can see how calmly vessel Cs effective tension rises with the increasing wave height. Vessel C can lift the compressor module in waves up to and including $H_s = 4{,}0m$ before the crane and wire capacity is breached.

The MTTF for the compressor module is estimated to be 7 years, but as subsea gas compression is a new technology it would be wise to anticipate that unplanned maintenance could be required. The MTTF for the compressor is also based on existing topside installations and the Ormen Lange test module located at Nyhamna, so the sample size for subsea gas compressors is small and therefore may be inaccurate.

To reduce the slamming forces on the compressor module a vertical configuration could have been chosen instead of a horizontal. A vertical arrangement reduces the module footprint, and with a smaller slam area less forces act on the module. A vertical configuration also lets the electrical motor be located on top of the compressor to give optimum protection against seawater or other liquids leaking into the motor, which further improves the reliability. The gas compressor that will be located at Ormen Lange will have this configuration.

Inlet and anti-surge cooler module

The Inlet and anti-surge cooler module have such huge dimensions that it is very impractical to lift through the splash zone. The large water-plane area induces large forces on the module once it hits the splash zone. Even though it is weighs over 200Te, there is a substantial risk of slack sling in the lifting wire. For vessel A lifting the cooler module in $H_s = 2{,}0m$ will according to Figure 5.5 cause slack sling. It is therefore assumed that vessel A won't be able to handle this module at all. For vessel B, $H_s = 2{,}0m$ will be the limiting wave height. Vessel C is also affected by the large footprint this module has, and even though the vessel doesn't experience any resonant motion the slam forces from the waves limits the operation to $H_s = 2{,}5m$.

As the Inlet and anti-surge cooler module doesn't have any rotating equipment, the MTTF is estimated to be 81 years. The module is in essence just an assembly of thin pipes with a collectively large surface that cools the incoming hydrocarbons which doesn't require much maintenance. One challenge the module might face is excessive biological growth on the pipes which will reduce the performance of the cooler, but the growth can in most cases be removed remotely without the need of replacing the module.

Table 6.1: The significant wave height for when the vessels can start the operation on Åsgard

Operational Criteria [24hour operation]			
	Vessel A	*Vessel B*	*Vessel C*
Pump module	*3,2m*	*3,8m*	*3,8m*
Compressor module	*1,6m*	*2,1m*	*3,2m*
Inlet and anti-surge cooler module	*N/A*	*1,5m*	*2,1m*

Today oil-companies are focusing on reducing cost, but the trend of “bigger is better” is still an approach that is used for development of IMR vessels. It would be cost efficient to utilize the existing IMR vessel fleet for the workover on the subsea factory instead of investing in new giant IMR vessels. From the previous analyses one can see that to ensure a good year around availability for the subsea plant the vessels must be able to work in high sea states. With good condition monitoring of the modules that are prone to critical failures (for the Åsgard compression train this is the pump and compressor modules), a smaller vessel can arrive at the location, wait for the right weather conditions and complete the operation before the critical failure occurs. From the analysis it is shown that both the smaller vessels are able to complete the splash zone lifting operation for the most vulnerable modules.

The Åsgard compression station will be the world’s first subsea compression station and a milestone is subsea processing. With the entire oil and gas industry is watching in anticipation, the pressure on Statoil to implement this new technology without significant failures, is immense. It is therefore understandable that the vessel responsible for the installation and IMR operations is the new technologically advanced North Sea Giant. Even though this may not be the most cost efficient solution in the long run for the IMR, it will improve the availability of the compression station in the start-up phase, by reducing the waiting on weather (WOW) the smaller vessels would have experienced.

It is difficult to predict a trend when it comes to subsea processing due to its novelty and the fact that each case will have their unique circumstances and thus individual needs, though it’s believed that future compression stations will have smaller modules compared to the ones seen at Åsgard. This would enable the smaller vessels in the existing IMR fleet handle the module replacement of all the rotating equipment on any future subsea gas compression station.

6.2 CONCLUSION

The vessel which in this thesis represents the largest and most advanced IMR vessel on the marked, vessel C, will have no problems completing the IMR required for the subsea factory. The only module that cannot be installed or retrieved in $H_S = 4{,}0m$ is the inlet and anti-surge cooler module because of its large water plane area. This vessel would have no problem completing year around IMR work on the Åsgard field.

Vessel B, which represents the average IMR vessel found on the marked today, will be able to the year around workover for the modules that are expected to have the lowest MTTF. The vessel is able to theoretically lift the inlet & anti-surge cooler module, but due to the operational criteria being so low another vessel would have been used for an unplanned retrieval of this module. If in the future the size and weight of the compressor module is reduced, then these medium sized IMR vessels will be able to replace and install all the equipment with low MTTF in the same waves as vessel C.

The last vessel represents a small IMR vessel. Vessel A will mainly be required to do ROV inspection and light maintenance and repair work. The vessel can also be used for retrieval of light modules such as the pump module or subsea control modules. From the analysis it’s concluded that there is possible to lift the compressor module, but the operational criteria is so low that in reality this vessel wouldn’t have been used for this operation.

7 SUGGESTIONS FOR FURTHER WORK

As mentioned previously in this thesis, the RAO data for the vessels are solely based on the generated RAO curves from ORCAFLEX, it would be recommended to do the analysis for vessels with more realistic RAO data and also to include active heave compensation and guide wires in the analysis. In this thesis the lifted modules have been fixed in place. When the module is hooked to the SHS tower and suspended over the side of the vessel there would be additional forces affecting the module.

The modules are also modelled as cubes with a given perforation percentage, this is a simplified method of estimating slam- and water plane area. The modules are structural elements are mainly pipes and beams. A lumped buoy could be created for each pipe or beam segment to give more accurate slam- and water plane areas.

Computational fluid dynamics (CFD) uses a numerical technique for solving and analyzing fluid flows. This method for solving complex fluid flows have become a standard tool in the oil industry, much because of the evolving speed of computer computation. The objective of CFD analysis is to be able to predict forces acting on the module and to improve the estimate of the coefficients used in the analysis. Comparing CFD testing to the numerical analysis from ORCAFLEX would be of interest. The next step would then be to make model tests and analyze the models in wave tanks to further improve the accuracy of the analysis.

It would also be of interest to redo the analysis with the improved ORCAFLEX input mentioned above for the smaller compressor module being designed by the JIP and see if the same conclusion for large vessels can be drawn.

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APPENDIX

A.1 THE ÅSGARD COMPRESSION STATION

Name:	Illustration:	Weight:	Dimension:
Template structure		1800t	74 x45 x 26m
Compressor module		289t	11 x 9 x 10m
Inlet/anti-surge cooler module		235t	15 x 10 x 8m
Discharge cooler module		107t	9 x 7 x 5m
Scrubber module		210t	8 x 8 x 12m
Pump module		45t	5 x 5 x 6m

Figure A 1: The modules that make up the Åsgard compression station [9]

The subsea compression facility at Åsgard consists of two parallel compressor trains that accelerate the gas from several production templates from the satellite fields Mikkel and Midgard and back to the floating production platform "Åsgard B". One compressor train is able to boost the gas pressure by 50 bar and in total the compression system will deliver over 21 million SCM gas per day. Each compressor train consists of six process modules and several support modules for control- and power-distribution. The compression process is described in Figure A 2. The inlet gas has been heated by mother earth and needs to be cooled down. The inlet cooler module uses seawater to cool the incoming gas down to a safe temperature of 10-16°C. The cooled gas then passes through the vertical scrubber module removing all condensate liquids from the production. The condensate liquids are then pumped directly into the export line to Åsgard B using the condensate pump module. The gas passes through the top of the scrubber and into the compressor to be compressed to the required pressure. Compression of gas creates heat and again the gas is cooled using seawater in the discharge cooler module [4]

The compression facility is installed to boost the gas velocity and prevent liquid gas to form in the flowlines. The faster decline in production than anticipated is due to higher pressure loss than expected combined with earlier water break-through in some of the wells. A minimum gas rate is essential to maintain flow assurance as liquid gas in the flowlines will create dynamic instabilities (slugging) and prevent a continuously MEG distribution preventing hydrate formations.

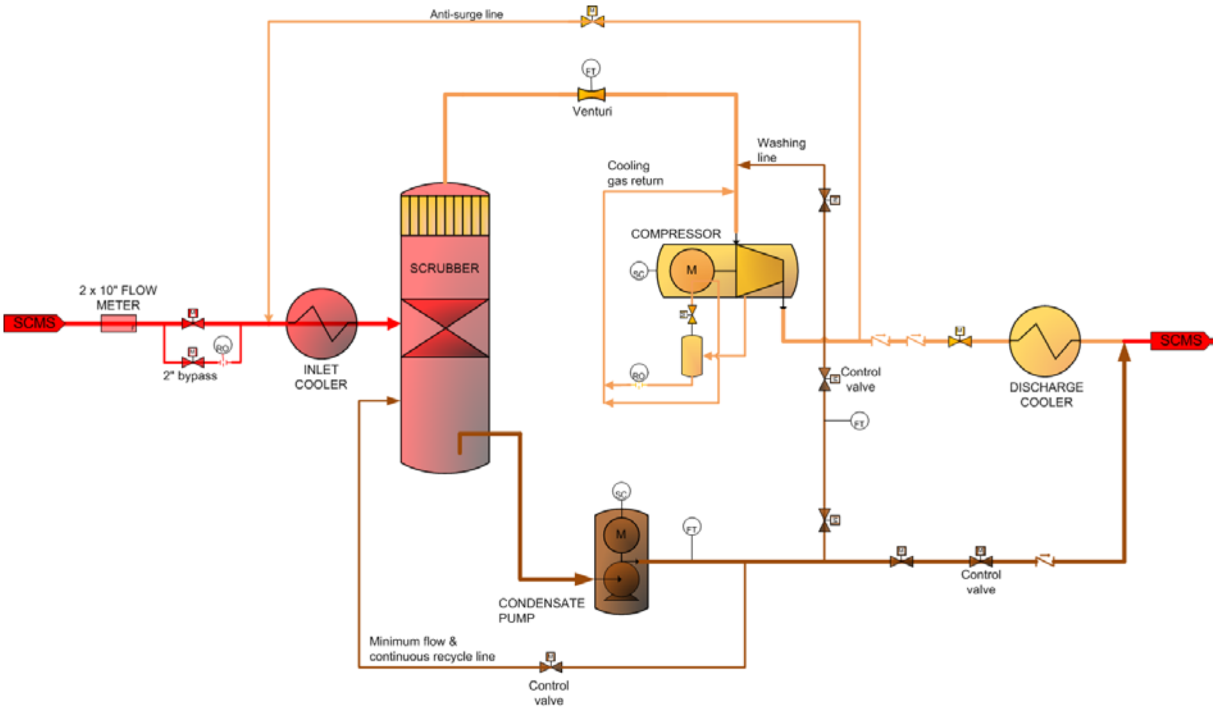


Figure A 2: The compression process on Åsgard compression system [9]

The compression system in itself is nothing new and is done every day on platforms and in process facilities, but this will be the first commercially full scale, all electric compression station installed on the seabed. To ensure good maintainability and supportability Statoil has ordered three separate compression trains, where two are installed in the compression station at Åsgard and a third is on standby onshore.

A.2 WAVE STATISTICS FOR ÅSGARD

Table A 1: Duration (hours) of sea states where the significant wave height are below specified values [40]

	2m				3m			
Month	Mean (h)	Stdev.(h)	Max (h)	# events	Mean (h)	Stdev.(h)	Max (h)	# events
1	42	44	247	3.4	73	84	425	5.1
2	45	44	323	3.9	80	96	610	4.7
3	54	59	430	4.2	99	124	976	5.1
4	69	60	321	4.9	156	221	1.841	4.7
5	104	106	642	5.3	288	301	1.536	3.5
6	112	122	1.034	5.0	376	414	1.790	2.9
7	136	142	745	4.7	451	451	1.852	2.4
8	131	127	823	4.4	327	274	1.224	2.5
9	71	71	441	4.9	126	138	867	4.9
10	52	55	341	4.2	75	93	771	5.9
11	38	38	255	4.0	63	68	400	6.0
12	38	36	243	3.2	58	65	443	5.6
Year	75	92	1.034	49.5	126	207	1.852	47.0
	4m				5m			
Month	Mean (h)	Stdev.(h)	Max (h)	# events	Mean (h)	Stdev.(h)	Max (h)	# events
1	112	137	759	5.2	178	400	4.984	4.8
2	143	216	2.104	4.2	229	453	4.487	4.0
3	183	282	2.089	4.7	392	717	4.795	3.8
4	414	656	3.852	3.5	996	1.230	4.559	2.2
5	768	909	3.746	2.3	1.614	1.303	3.839	1.6
6	1.028	886	3.137	1.7	1.770	1.036	3.147	1.3
7	981	701	2.579	1.5	1.653	626	2.757	1.1
8	603	430	1.835	1.7	963	516	2.013	1.3
9	252	258	1.595	3.3	403	371	1.784	2.4
10	134	184	1.488	5.3	237	267	1.491	4.0
11	111	130	761	5.4	200	219	1.086	4.0
12	80	93	555	6.7	142	190	1.361	5.5
Year	200	389	3.852	36.5	317	633	4.984	25.4
	6m				7m			
Month	Mean (h)	Stdev.(h)	Max (h)	# events	Mean (h)	Stdev.(h)	Max (h)	# events
1	307	683	5.757	4.0	566	1.144	6.698	2.9
2	549	1.073	5.616	2.9	960	1.614	6.326	2.2
3	882	1.534	5.601	3.0	1.994	2.405	7.759	2.1
4	2.263	1.860	5.246	1.6	3.567	2.029	7.480	1.2
5	2.822	1.298	4.713	1.1	3.299	1.714	6.760	1.2
6	2.531	930	3.969	1.1	3.204	1.093	6.016	1.0
7	2.066	634	3.249	1.0	2.707	825	5.296	1.0
8	1.353	556	2.505	1.1	1.963	825	4.552	1.0
9	653	490	1.804	1.8	1.166	789	3.808	1.3
10	435	376	1.494	2.5	747	623	3.088	1.8
11	328	349	2.175	3.0	566	525	2.344	2.1
12	275	339	1.865	3.8	439	515	2.283	2.7
Year	544	1.030	5.757	15.5	961	1.594	7.759	9.0

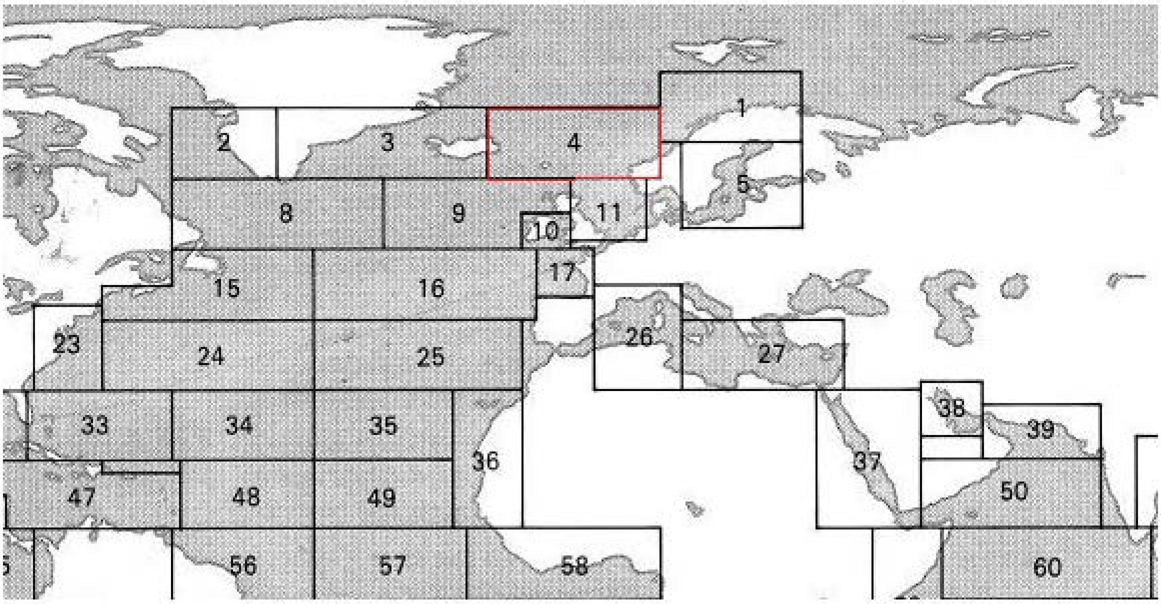


Figure A 3: Definition of nautical zones for estimation of long term wave distribution parameters for the Norwegian Sea [38]

Table A 2: Weibull parameters α and β for the long-term probability distribution of the significant wave heights in the Norwegian Sea [38]

Area	α	β
1	2.33	1.33
2	1.96	1.34
3	2.74	1.35
4	2.84	1.53
5	1.76	1.59
6	2.76	1.45

A.3 WEIBULL DISTRIBUTION

The cumulative distribution function (CDF) of the Weibull distribution is defined as:

$$F_X(x) = 1 - e^{-\left(\frac{x}{\alpha}\right)^{\beta}}, x > 0 \tag{D.1}$$

Where

α	-	scale parameter (α > 0)	[−]
β	-	shape parameter (β > 0)	[−]

The expected mean value is

$$\mu = \frac{\alpha}{\beta} \Gamma\left(\frac{1}{\beta}\right) \tag{D.2}$$

and the standard deviation is

$$\sigma = \sqrt{\frac{\alpha^2}{\beta}\left[2\Gamma\left(\frac{2}{\beta}\right) - \frac{1}{\beta}\Gamma^2\left(\frac{1}{\beta}\right)\right]} \tag{D.3}$$

Where Γ is the gamma function defined by

$$\Gamma(x) = \int_0^{\infty} e^{-t} t^{x-1} dt \tag{D.4}$$

Integration by parts gives the functional relation of the gamma function

$$\Gamma(x+1) = x\Gamma(x) \tag{D.5}$$

With the mean and standard deviation values known the Weibull parameters can be calculated as follows. From Eq. (D.2) and (D.3) it's possible to deduce that

$$\left(\frac{\sigma}{\mu}\right)^2 = \frac{\frac{2}{\beta}\Gamma\left(\frac{2}{\beta}\right)}{\left(\frac{1}{\beta}\right)^2 \Gamma^2\left(\frac{1}{\beta}\right)} \tag{D.6}$$

Which gives the value for the shape parameter and next the scale parameter is calculated from Eq. (D.2)

$$\alpha = \frac{\mu}{\frac{1}{\beta}\Gamma\left(\frac{1}{\beta}\right)} \tag{D.7}$$

Figure A 4: Subsea 7 IMR portfolio [41]

MEASUREMENTS		
Lenght o.a		160,9 m
Lenght b.p.p		144,0 m
Breadth		30,0 m
Dept Main deck		10,7 m
Max. Draft		7,5 m
Net		5446T
Gross tonnage		18151T
Deck load	Apr.	1000T

CLASS NOTATION

DnV +1A1, HELDK, DYNPOS-AUTRO, EO, DK(+), COMF-(V3), NAUT-OSV, CLEAN DESIGN

DECK CAPACITIES	
Main deck open area	2900 m2
Main deck closed area	650 m2
Main deck strenght	10 T/M2
Fuel Oil	2000 m3
FW	1000 m3

LOADING COMPUTER

Shipload

DP SYSTEM AUTRO DP3	
Kongsberg	K-POS DP-21
Kongsberg	K-POS DP-11BU
Kongsberg	K-POS DP-11 / cJoy

Reference systems:
DPS 700
DPS 200
Dual HIPAP 500

ENTERTAINING SYSTEM

Seatel TVRO Sat.system
INFOTAINMENT in all cabins and rec.rooms

LIFE SAVING EQUIPMENT	
Life boats	2x 102
Life rafts	2x 25
Rescue boats	2

Figure A 5: North Sea Giant specifications [42]

VESSEL SPECIFICATION SHEET

SUBSEA IMR AND ROV SUPPORT VESSEL

The State-of-the-Art IMR vessel Edda Fauna came into operation in 2008 as the flagship in the IMR fleet. Edda Fauna was designed with special emphasize on providing excellent safety and work conditions on deck with a closed deck hangar. Edda Fauna accommodation and office facilities are of very high standards, creating a good working environment for offshore crew and clients.

MAIN CHARACTERISTICS

LENGTH O.A	108,70 m
LENGTH B.P	96,00 m
BREADTH MLD	23,00 m
DRAFT, MAX	7,80 m
AIR DRAFT	42,00 m from base line
DEPTH MLD TO MAIN DECK	9,60 m
DRAFT , MIN	5,3 m
DRAFT DESIGN	6,50 m
SERVICE SPEED	Approx. 15 knots
MAX SPEED	Approx. 16.5 knots

KEY VESSEL FEATURES

- SUBSEA IMR AND ROV SUPPORT VESSEL.
- DE-ICE NOTATION, WITH COVERED LIFEBOATS, MOB BOATS, BOW AREA AND HANGAR AREA.
- 60 TE MODULE HANDLING SYSTEM (MHS)
- MHS OPERATIONS IN ENCLOSED HANGAR AREAS
- SKIDDING SYSTEM FOR 60 TE MODULES ON MAIN DECK
- 1 OBS-ROV & 2 WORK-ROV'S
- LARS HANDLING SYSTEM FOR OBS-ROV AND WORK-ROV
- THREE MOONPOOLS FOR MHS AND ROV OPERATIONS
- 100 TE AHC OFFSHORE CRANE
- FIXED INSTALLED SCALE SQUEEZE SYSTEM ONBOARD
- SCR CATALYTIC REACTORS FOR REDUCED NOX EMITION TO AIR
- OUTSIDE DECKAREA 610M^2, INSIDE DECK AREA (HANGAR) 650M^2
- ACCOMODATION FOR 90 PERSONS TOTAL
- CLASSIFICATION: DNV+ 1A1,SF,E0,ICE-C,DYNPOS AUTR (ERN 99/98/97) CLEAN,DK(+),
- HL(2,8),LFL*COMF-C(3)-V(3)NAUT OSV(A),HELDK-SH,PMS,ISM
- HELIDECK FOR SIKORSKY S-92

Figure A 6: Edda Fauna specifications [43]

A.5 INTERVENTION COST CALCULATIONS

The assumptions this quick analysis is based on are as follows:

- The intervention operation in itself takes 24 hours
- It takes 1 month to mobilize the IMR vessel, crew and module
- When a pump or compressor module is shut down, the production is reduced. The lost revenue for a subsea pump is 3MNOK/day and for a compressor 18MNOK/day
- The intervention operation costs all in all 20MNOK
- The lifetime of a pump or compressor is assumed to be equal. The MTTF for both components is 6 years.

For strategy 1; "run until failure" there is a 30 day shutdown for every failure. The cost of a pump breakdown will be:

$30days * \frac{3MNOK}{day} + 20MNOK = 110MNOK$ with yearly expenses: $\frac{110MNOK}{6years} = \mathbf{18,33MNOK/year}$

While the cost for a compressor breakdown will be:

$30days * \frac{18MNOK}{day} + 20MNOK = 560MNOK$ where the yearly expenses are: $\frac{560MNOK}{6years} = \mathbf{93,33MNOK/year}$

If instead strategy 2 is used; "Replace after certain limit is reached" and this limit is 3 years of operation. The cost of this strategy for the pump module will then be as follows:

$$1day * \frac{3MNOK}{day} + 20MNOK = 23MNOK\ and\ \frac{23MNOK}{3years} = \mathbf{7,67MNOK/year}$$

While for the compressor the cost will be:

$$1day * \frac{18MNOK}{day} + 20MNOK = 38MNOK\ and\ \frac{38MNOK}{3years} = \mathbf{12,67MNOK/year}$$

The 3rd strategy; "Condition based maintenance" is based on the assumption that it is possible to detect a failure at least one month before a critical failure occurs in either the pump or the compressor module. The average cost for the pump module then becomes:

$$1day * \frac{3MNOK}{day} + 20MNOK = 23MNOK\ and\ \frac{23MNOK}{5,917years} = \mathbf{3,89MNOK/year}$$

The cost of the intervention for the compressor will be:

$$1day * \frac{18MNOK}{day} + 20MNOK = 38MNOK\ and\ \frac{38MNOK}{5,917years} = \mathbf{6,42MNOK/year}$$

A.6 FAILURE DATA ÅSGARD COMPONENTS

Component Name	MTTF	Data Source	OREDA Filter	Critical Failure Modes	Comments
Scrubber	50	OREDA-2009 Volume 1 p. 369-376	Scrubber, Topside	External leakage Structural deficiency Instrumental failure	No relevant data available for subsea scrubbers. It is assumed that the subsea design will be more redundant than the topside equivalent and that the requirements for separation quality/performance will be lower, thus reducing the amount of calibration valves necessary. The contribution to failure from these valves is removed. Structural deficiency failures are also reduced due to the increased quality- assurance and control requirements for subsea manufacturing.
Compressor	7	OREDA-2009 Volume 1 p. 75	Compressors, Centrifugal, Electric	External leakage Fail to start on demand Internal leakage Low output Spurious stop Vibration	No relevant data available for subsea compressors. The subsea design differs from the topside design as there are no power transmission elements, no lubrication systems and no shaft seal system. The contribution from elements not included in the subsea design were removed and not accounted for in the total critical failure rate. The critical failure rate was also reduced since magnetic bearings are used instead of traditional bearings.
Inlet cooler	81	OREDA-2009 Volume 1 p. 303-310	Heat exchangers – shell and tube	External leakage Structural deficiency	No relevant data available for subsea coolers. As the passive cooler can be seen as segments of pipes, topside data for shell and tube coolers were studied. The critical failure rate was based on the piping of a shell and tube cooler. The subsea design was also assumed to be more robust than its topside equivalent, except in regards to vibrations.
Condensate pump module	5	OREDA-2009 Volume 1 p. 154	Pumps, Centrifugal, Condensate processing	Fail to start on demand Internal leakage Low output Spurious stop Vibration	No relevant data available for subsea pumps. External leakage is not considered a critical failure as the lube oil and the slight overpressure is designed to leak into the production.

A.7 ORCAFLEX ANALYSIS

The input data for the ORCAFLEX analysis shown in Table 4.5 are calculated by the formulas shown in this appendix.

Mass moment of inertia in x, y and z direction:

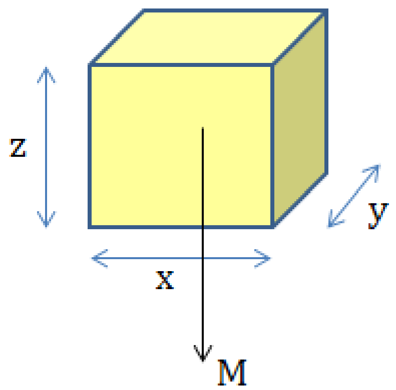
$$I_x = \frac{1}{12} * M * (y^2 + z^2) \quad [m^4]$$

$$I_y = \frac{1}{12} * M * (x^2 + z^2) \quad [m^4]$$

$$I_z = \frac{1}{12} * M * (x^2 + y^2) \quad [m^4]$$

Where

M	-	mass of module $[kg]$
x	-	length of module $[m]$
y	-	width of module $[m]$
z	-	height of module $[m]$



Volume of displaced water:

$$\nabla_{Steel} = \frac{M}{\rho_{steel}} \quad \left[\frac{kg}{m^3}\right]$$

Where

∇_{Steel}	-	volume of steel $[m^3]$
ρ_{steel}	-	density of steel $[kg/m^3]$

It is assumed that the module will be flooded with water before submergence as this is standard procedure to prevent implosion. The volume of displaced water is therefore assumed to be the steel volume of the module. It is also assumed that the module is made up entirely of steel with a uniform density of 7800$\frac{kg}{m^3}$.

The drag area in x, y and z direction:

$$A_x = y * z$$

$$A_y = x * z$$

$$A_z = x * y$$

Hydrodynamic added mass coefficients (C_A) in all directions is found from DNV-RP-H101 Appendix A, Table A-2, a relevant part of the table is shown below as it is assumed that the modules can be represented as rectangular plates with the added mass for the sided included as seen in chapter 4.2.3. The different C_A are found through linear interpolation.

Table A 3: Analytical added mass coefficients for three-dimensional bodies [33]

Body shape		*Direction of motion*	C_A				V_R
Flat plates	Circular disc	Vertical	$2/\pi$				$\frac{4}{3}\pi a^3$
	Elliptical disc	Vertical	b/a	C_A	b/a	C_A	$\frac{\pi}{6} a^2 b$
			∞	1.000	5.0	0.952	
			14.3	0.991	4.0	0.933	
			12.8	0.989	3.0	0.900	
			10.0	0.984	2.0	0.826	
			7.0	0.972	1.5	0.758	
			6.0	0.964	1.0	0.637	
	Rectangular plates	Vertical	b/a	C_A	b/a	C_A	$\frac{\pi}{4} a^2 b$
			1.00	0.579	3.17	0.840	
			1.25	0.642	4.00	0.872	
			1.50	0.690	5.00	0.897	
			1.59	0.704	6.25	0.917	
			2.00	0.757	8.00	0.934	
			2.50	0.801	10.00	0.947	
			3.00	0.830	∞	1.000	

Compressor module C_A in z-direction:

$$\frac{b}{a} = \frac{11}{9} = 1{,}22$$

$$C_A = 0{,}579 + \frac{0{,}642 - 0{,}579}{1{,}25 - 1{,}00} * (1{,}22 - 1{,}00) = \mathbf{0{,}634}$$

The added mass for the fully submerged compressor in z-direction is then:

$$A_{33,z} = \rho_{sw} * C_A * V_R \quad [kg]$$

Where

$$V_R = \frac{\pi}{8} * a^2 * b \qquad [m^3]$$

V_R represents the added mass volume of one submerged side, in the case the module is not fully submerged only the bottom of the module will experience added mass. The added mass on the bottom of the module is found by:

$$A_{33,z} = 1025 \frac{kg}{m^3} * 0{,}634 * \frac{\pi}{8} * 9^2 m^2 * 11 m^2 = \mathbf{227{,}9\,Te}$$

The fully submerged module will have twice this added mass since the top of the module will be submerged. The sides of the module must also be accounted for.

To account for the added mass due to the sides of the module the following formulas from 4.2.3 are used:

$$\lambda = \frac{\sqrt{A_p}}{h + \sqrt{A_p}} = \frac{\sqrt{9 * 11}}{10 * \sqrt{9 * 11}} = 0{,}4987 \quad [-]$$

and

$$A_{33s,z} \approx \left[1 + \sqrt{\frac{1-\lambda^2}{2(1-\lambda^2)}}\right] * A_{33o,z} = \left[1 + \sqrt{\frac{1-0{,}4987^2}{2*(1-0{,}4987^2)}}\right] * 227{,}9Te = \mathbf{352{,}9}\boldsymbol{Te}$$

When the module is fully submerged the second V_R will have to be accounted for as the top of the module is now submerged. The total hydrodynamic added mass for a non-perforated compressor module will then be:

$$352{,}9Te + 227{,}9Te = \mathbf{580{,}8}\boldsymbol{Te}$$

It is assumed that the compressor module will not be a solid cube, but an assembly of pipes and beams welded together. The module will therefore have several areas where water can flow freely through the module. To account for this it is assumed that the module has a perforation percentage. This percentage is assumed to be 40%. From DNV-RP-H103 sec 4.6.4.1 the perforation is given as:

$$A_{33} = A_{33S} * e^{\frac{10-p}{28}}, \quad if\ 34 < p < 50$$

The hydrodynamic added mass for the fully submerged compressor module will then be:

$$A_{33} = 580{,}8Te * e^{\frac{10-40}{28}} = \mathbf{198{,}9}\boldsymbol{Te}$$

The same calculation has been done for the other directions and the two other modules.

The hydrodynamic inertia coefficient is defined as:

$$C_M = C_A + 1$$

And the hydrodynamic slam coefficient is defined as:

$$C_S = 5$$

The slam area is found as:

$$A_w * Perforation\ rate$$

Where

A_w - the water plane area $[m^2]$

The maximum and minimum values from the ORCAFLEX which are used in the chapter 5.1, these are the maximum and minimum values for all the different zero crossing periods for each wave height. The tables also show the forces for each submergence level.

Vessel A

Table A 4: The max and min values for different submergence levels from the ORCAFLEX splash zone analysis for vessel A

Pump maximum values						
	Hs = 2	Hs = 2.5	Hs = 3	Hs = 3.5	Hs = 4	Hs = 4.5
Z = 1	530,9	519,3	627,4	636,7	727	864,8
Z = 0	785,1	705,1	765,9	968,9	914,8	1229
Z = -1.5	739	657,5	745,4	1014,5	1074,6	1266,3
Z = -3	617	616	684,3	793,7	930,1	1081,8
Z = -6	470,7	507,5	562,2	629,1	630,8	733,7
Z = -9	459,6	475,6	480,2	494,1	499,2	521,7

Pump minimum values						
	Hs = 2	Hs = 2.5	Hs = 3	Hs = 3.5	Hs = 4	Hs = 4.5
Z = 1	273,2	383,7	244,1	239,2	188,2	130,3
Z = 0	112,3	198,7	166,7	66,8	57,8	-2
Z = -1.5	206,9	225,8	121,3	75,8	53,1	-18,1
Z = -3	246,4	256,1	195,2	135,1	123,4	82,7
Z = -6	317,3	289,6	255,7	217,2	201,7	167,4
Z = -9	339	324,3	315	300,2	309,3	298,1

Compressor maximum values		
	Hs = 2	Hs = 2.5
Z = 3	2942,4	2988,7
Z = 1	3024,8	3525,7
Z = 0	3529,1	4082
Z = -2	3744,6	4225,1
Z = -5	3437,7	3342,2
Z = -25	2775,8	2641,1

Inlet and anti-surge cooler maximum values	
	Hs = 2
Z = 1	3557,5
Z = 0	3769,7
Z = -1	3441,2
Z = -3	3658,1
Z = -8	3347
Z = -13	2516,5

Compressor minimum values		
	Hs = 2	Hs = 2.5
Z = 3	2755,1	2733,2
Z = 1	2523,1	1992,7
Z = 0	2267,5	1333,3
Z = -2	1854	1553,5
Z = -5	2131,9	1655,8
Z = -25	2181,5	2282,8

Inlet and anti-surge cooler minimum values	
	Hs = 2
Z = 1	1012,9
Z = 0	205,5
Z = -1	281,9
Z = -3	996,6
Z = -8	1044,4
Z = -13	1605

Vessel B

Table A 5: The max and min values for different submergence levels from the ORCAFLEX splash zone analysis for vessel B

Pump maximum values						
	Hs = 2	Hs = 2.5	Hs = 3	Hs = 3.5	Hs = 4	Hs = 4.5
Z = 1	507,3	514,4	606,1	618	520,6	764,8
Z = 0	615,5	633,6	691,9	848,5	682,6	1035,6
Z = -1.5	709	646,4	581,6	823,9	930,1	1001,7
Z = -3	617	591,7	619,2	790,4	914,8	963,2
Z = -6	470,7	485,4	525,1	561,3	566,4	597
Z = -9	459,6	466,4	472,8	486	496,4	515,5

Pump minimum values						
	Hs = 2	Hs = 2.5	Hs = 3	Hs = 3.5	Hs = 4	Hs = 4.5
Z = 1	273,2	385,9	251,7	269,1	208,2	314,2
Z = 0	162,3	228,7	219,7	119,2	101,9	60,9
Z = -1.5	206,9	225,8	216,4	109	277,3	81,1
Z = -3	246,4	303,8	250,7	189,5	161,8	97,7
Z = -6	317,3	319,1	280,6	241,3	214,2	167,4
Z = -9	339	342,4	331,5	320,4	314,9	298,1

Compressor maximum values				
	Hs = 2	Hs = 2.5	Hs = 3	Hs = 3.5
Z = 1	2966	3287	3437	3609,8
Z = 0	3460,5	3885,4	4355,8	4410
Z = -2	3376	3824,1	4459,7	4491,5
Z = -5	3103	3319,1	3606,9	3748,8
Z = -10	2763,6	2821,2	2896,2	2977,9
Z = -25	2575,2	2638,5	2673,6	2633,4

Compressor minimum values				
	Hs = 2	Hs = 2.5	Hs = 3	Hs = 3.5
Z = 1	2643,3	2332,4	2209,6	2156,6
Z = 0	2074,5	1514,8	774,7	347,9
Z = -2	2050,6	1575,3	1057,9	1164,6
Z = -5	2334,4	2159,1	1946,2	1826,8
Z = -10	2246,4	2133,3	2000	1937,3
Z = -25	2349,5	2278,3	2242	2335,8

Vessel B

Table A 6: The max and min values for different submergence levels from the ORCAFLEX splash zone analysis for vessel B

Inlet and anti-surge cooler maximum values		
	Hs = 2	Hs = 2.5
Z = 1	3159,1	4227,8
Z = 0	3396,6	4232,3
Z = -1	3645	4112,9
Z = -3	3471,8	3531,2
Z = -8	2833	2969,4
Z = -13	2515,8	2591,1

Inlet and anti-surge cooler minimum values		
	Hs = 2	Hs = 2.5
Z = 1	1206,8	603
Z = 0	556,3	365,9
Z = -1	657,3	185,7
Z = -3	628,9	1127,4
Z = -8	1462,8	1367,9
Z = -13	1604,5	1481,5

Vessel C

Table A 7: The max and min values for different submergence levels from the ORCAFLEX splash zone analysis for vessel C

Compressor maximum values					
	Hs = 2	Hs = 3	Hs = 3.5	Hs = 4	Hs = 4.5
Z = 1	2919,8	3161,4	3330,8	3510	3901,6
Z = 0	3362,6	3461,7	3521,3	3863,2	4122,5
Z = -2	3360,3	3386,5	3635,2	3826,8	4110,1
Z = -5	2968,1	3195	3364,1	3539,3	3337,2
Z = -10	2702	2763,4	2833,4	2958,3	3043,5
Z = -25	2556,5	2621,8	2619,4	2667,3	2690,1

Compressor minimum values					
	Hs = 2	Hs = 3	Hs = 3.5	Hs = 4	Hs = 4.5
Z = 1	2687,2	2641,6	2287,3	1914,2	1510,3
Z = 0	2331,6	2154,3	1989,7	1750,5	1618,8
Z = -2	2219,1	2147,6	1809,5	1859,5	1616,4
Z = -5	2404,2	2227,7	2067,6	2083,4	2017,2
Z = -10	2296,5	2184,9	2114,8	2044,2	1958,3
Z = -25	2362,2	2336,2	2287,5	2286,1	2261

Inlet and anti-surge cooler maximum values			
	Hs = 2	Hs = 2.5	Hs = 3
Z = 1	2849,3	3275,2	3750,1
Z = 0	3504,7	3675,8	4115,3
Z = -1	3240,7	3678,2	3911,7
Z = -3	2939,2	3168,4	3936,6
Z = -8	2891,3	2873	2920
Z = -13	2552,6	2619,1	2398,9

Inlet and anti-surge cooler minimum values			
	Hs = 2	Hs = 2.5	Hs = 3
Z = 1	1368,5	797,3	442,1
Z = 0	842,4	606,4	-15,6
Z = -1	943,7	545,5	151,2
Z = -3	1603,8	816,5	801,8
Z = -8	1292,5	1287,3	1150,9
Z = -13	1610,4	1481,5	1668,1

The crane tip motion graph seen in Figure 4.5 is based on the RAO data found in ORCAFLEX. The table where the data is acquired from ORCAFLEX is shown below.

Table A 8: The RAO data found in ORCAFLEX

0° | 22,5° | 45° | 67,5° | 90° | 112,5° | 135° | 157,5° | 180°

Periods: 22

Period	Surge		Sway		Heave		Roll		Pitch		Yaw	
	Ampl.	Phase	Ampl.	Phase	Ampl.	Phase	Ampl.	Phase	Ampl.	Phase	Ampl.	Phase
(s)	(m/m)	(deg)	(m/m)	(deg)	(m/m)	(deg)	(deg/m)	(deg)	(deg/m)	(deg)	(deg/m)	(deg)
0,00	0,000	0,0	0,000	360,0	0,000	360,0	0,000	0,0	0,000	0,0	0,000	360,0
4,00	0,0081	68,3	0,00082	206,8	0,014	282,6	0,065	106,7	0,056	-97,1	0,035	247,0
5,00	0,012	-88,5	0,016	264,8	0,065	133,4	0,175	-63,7	0,265	-220,4	0,032	214,2
5,50	0,017	66,2	0,018	126,2	0,091	190,1	0,099	63,4	0,678	-177,4	0,122	182,6
6,00	0,033	99,9	0,044	89,6	0,166	237,5	0,410	80,6	0,887	-165,6	0,087	134,3
6,50	0,029	104,8	0,044	65,1	0,306	239,5	0,607	79,1	0,660	-135,3	0,140	40,8
7,00	0,024	-55,5	0,030	12,7	0,244	211,8	0,625	73,2	1,079	-93,8	0,270	18,2
7,50	0,108	-70,6	0,046	-49,3	0,042	169,3	0,433	68,2	1,558	-89,5	0,372	11,3
8,00	0,203	-75,4	0,078	-69,1	0,149	22,9	0,129	135,0	1,804	-89,7	0,441	8,2
8,50	0,295	-78,6	0,113	-74,2	0,299	15,6	0,747	176,2	1,892	-90,6	0,479	6,7
9,00	0,378	-80,8	0,150	-77,2	0,422	12,4	1,233	145,5	1,880	-91,2	0,498	6,1
9,50	0,449	-82,3	0,182	-80,6	0,525	10,3	1,222	124,5	1,814	-91,6	0,504	5,6
10,00	0,510	-83,4	0,208	-82,7	0,608	8,8	1,093	114,0	1,725	-91,7	0,500	4,9
11,00	0,607	-84,8	0,248	-84,8	0,730	6,5	0,864	104,9	1,526	-91,6	0,470	4,0
12,00	0,678	-85,7	0,277	-85,9	0,809	4,9	0,704	101,1	1,338	-91,3	0,429	3,3
13,00	0,730	-86,3	0,299	-86,6	0,861	3,9	0,588	99,1	1,172	-91,0	0,387	2,7
14,00	0,768	-86,9	0,315	-87,1	0,897	3,2	0,498	97,8	1,030	-90,8	0,347	2,3
15,00	0,797	-87,3	0,328	-87,5	0,921	2,7	0,428	96,8	0,910	-90,7	0,312	2,0
16,00	0,819	-87,6	0,337	-87,8	0,939	2,3	0,372	96,2	0,808	-90,5	0,280	1,8
17,00	0,836	-87,9	0,344	-88,1	0,952	2,0	0,326	95,6	0,721	-90,4	0,252	1,6
18,00	0,849	-88,1	0,350	-88,3	0,961	1,7	0,289	95,2	0,646	-90,4	0,228	1,4
19,00	0,860	-88,3	0,355	-88,5	0,969	1,5	0,257	94,8	0,582	-90,3	0,207	1,3
20,00	0,869	-88,5	0,358	-88,6	0,975	1,4	0,231	94,5	0,527	-90,3	0,188	1,1
21,00	0,877	-88,6	0,362	-88,8	0,979	1,3	0,208	94,2	0,480	-90,3	0,172	1,0
22,00	0,883	-88,7	0,365	-88,9	0,982	1,1	0,189	94,0	0,438	-90,2	0,158	0,95
Infinity	0,924	90,0	0,383	90,0	1,000	0,0	0,000	0,0	0,000	0,0	0,000	0,0