FACULTY OF SCIENCE AND TECHNOLOGY

MASTER'S THESIS

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Abstract

Customized production and storage vessels, known as FPSO (Floating Production, Storage & Offloading vessel), are increasingly used in offshore oil and gas production due to their flexibility and ability to produce in deep water while exposed to severe weather conditions. With help from a dynamic positioning system and mooring system, the vessels can more or less keep their position through harsh storms. However, the vessels will have considerably larger movement than a rigidly fixed oil platform. One of many challenges is to avoid environmental loads being transferred to vulnerable equipment. Even smaller positioning offsets can cause serious consequences to the risers.

On most FPSOs, bend stiffeners are used to reduce bending forces at the interference point where the risers protrude from the turret on the way down to the seabed. The bend stiffener has a potential to fail and cause serious damage to the riser as they experience large bending forces in unfavorable weather conditions.

This study has its background from industry incidents where the bend stiffener has loosened without any real time knowledge of the failure. Thus, the purpose of this thesis has been to evaluate the possibility of an online monitoring device to provide a real time image of the riser positions. By doing so, the riser movement pattern can be recorded. Consequently, if an abnormal movement is recorded, the bend stiffener has most likely failed.

The main focus for this master thesis was to come up with the design of a deployment system to meet the given requirements of providing an online riser monitoring solution for BP’s Skarv FPSO. The thesis will evaluate different design alternatives and investigate the environmental loads the system will experience. Structural response and capacity analysis will be carried out for the important components to make sure the deployment system is suitable for further development.

Preface

This master thesis provides the design process and final result of the *Online Riser Monitoring Solution for Skarv FPSO* by Sveinung Fuglseth Rasmussen. The thesis has been conducted at the University of Stavanger (UiS) at the Department of Offshore Technology in the period of January to mid June 2012 and represents a workload of 30 ECTS points.

I would like to express my gratitude to my university supervisor Eiliv Janssen for great support and competent guidance throughout the development of this thesis. My greatest appreciation also goes to Thomas Brown, Martin Dove and the rest of the subsea team at BP Norway. Excellent support and guidance has been much appreciated while they have given me the opportunity to work at BPs office in Stavanger.

As this thesis is the final work of my master study at the University of Stavanger, I would like to express my appreciation to fellow students for good cooperation and teamwork. This appreciation goes especially to Espen Slettebø and Erlend Revheim. A special gratitude goes to Malin Toftesund Økland for helping out with spelling and grammar review.

Drawings and illustrations without references are designed and produced by myself, Sveinung Fuglseth Rasmussen.

Stavanger – 13th of June, 2012
Sveinung Fuglseth Rasmussen

Content

Abbreviations

BSCS	Bend Stiffener Connection System
BS	Bend Stiffener
BP	British Petroleum P.L.C
CP	Corrosion Protection
DNV	Det Norske Veritas
FPSO	Floating Production, Storage and Offloading unit
ID	Inner Diameter
IMP	Integrity Management Procedure
OD	Outer Diameter
ROV	Remote Operating Vehicle
SCU	Surface Control Unit
TSA	Thermally Sprayed Aluminum (coating)
WT	Wall Thickness

Codes and standards

Eurocode 3:1993	Basis of structural design
DNV-Ship rules Pt.3	Vessels accelerations
DNV-RP-C205	Environmental conditions and environmental loads
DNV-RP-F109	On-bottom stability design of submarine pipelines
DNV-RP-H103	Modeling and analysis of marine operations

1. Introduction

The Skarv FPSO is located in an area with harsh weather conditions. The risers are therefore exposed to severe loadings throughout their lifetime. Since a failure to the risers (flexible pipe) can have a catastrophic outcome to the platform and personnel onboard, bend stiffener components are installed at the riser interface with the FPSO hull. These components are meant to reduce the bending forces imparted to the risers. By monitoring the riser deflections/positions, one can provide a real time feedback of the bend stiffeners condition. The goal of this chapter is to provide an understanding of the purpose of this thesis.

In recent years within the oil and gas industry, the use of a marine vessel connected to a subsea network has been a satisfying solution for field production. A large vessel, containing production, storage and offloading modules is becoming more frequently used in harsh weather conditions, as an alternative to a rigidly fixed platform.

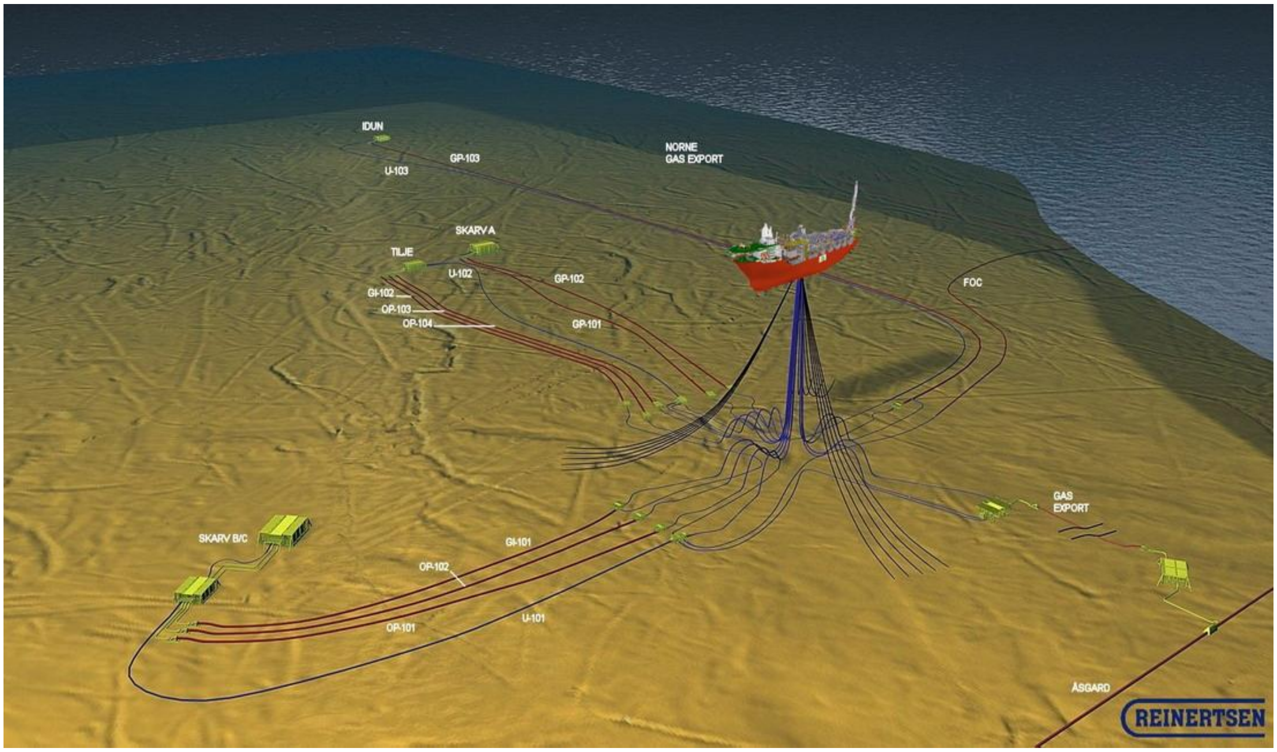

Figure 1-1: Skarv FPSO and its subsea system. (BP drawing archive, 2007-2012)

The FPSO is fastened to the seabed through mooring lines connected to the turret. On the FPSO, the turret is the center point of rotation, which allows the whole vessel to rotate around the connection point, while risers (flexible pipes) and umbilicals can stay in preferred position. This way, the FPSO can face the waves at all time, handle harsh weather conditions and still keep continuous production.

1.1 The Need for Monitoring

While the FPSO is producing, many important components and areas are exposed to rough weather conditions. Therefore, it is important to make sure the equipment maintains its integrity at all times. In recent years, safety and integrity management has gained an increasing focus within the industry. The need for monitoring and surveillance is therefore growing as a part of the process. Today, methods like measuring bending angle and tension in the mooring lines, as well as visual inspections of submerged equipment are frequently executed to prevent shutdowns and unwanted situations. In this task we will consider the connection area of the interface between risers and the turret.

The turret is the connection point between the subsea system and process unit. The risers, which bring the oil and gas to the surface, have a critical area at the point where the risers protrudes from the turret. Because of the bending moment generated by movement onto the risers, bending stiffeners are installed at the interface point where the risers protrude from the turret. The bend stiffeners are installed to prevent severe loadings on the risers. In a risk assessment performed on the Skarv FPSO, the consequence of failure to the bend stiffeners is considered as high (BP Norway, 2009), as they are preventing the risers from overbending. A damage to the risers cause critical situations due to their containment of hydrocarbons. An annual inspection is therefore required to ensure the integrity of the bend stiffeners and their connection system.

Due to the importance of these stiffeners to stay intact and the high failure consequence stated by the risk assessment (BP Norway, 2009), surveillance is needed. Cameras are planned to perform routine checks of risers and bend stiffeners at the Skarv FPSO (Roland Barr, 2011). The problem with cameras is that if a failure occurs between a routine surveillance check, it will not be noticed until next routine inspection. Therefore, there is a preference in the offshore business for real time surveillance solution for monitoring risers. This has been tested for the first time when BP installed a Riser and anchor monitoring system (RAMS) at Foinaven FPSO in 2007. The monitoring itself was successful and detected an incident where a bend stiffener had loosened from its position. An alarm was triggered as one of the risers was out of preferred position. As a response to the alarm, visual inspection showed at an early stage that one of the bend stiffeners had fallen down several meters. (Kaye, 2008)

Figure 1-2: Left: The bend stiffeners protrude down from the I-tube at the hull of the Skarv FPSO. Right: A historical illustration of a loosened bend stiffener on an in-service FPSO (Subsea7, 2001 and 2011).

1.2 Skarv FPSO

At the Foinaven FPSO, which is discussed in section 1.1, the deployment system was designed as temporary equipment to support a field trial of a monitoring device. It was deployed through an unoccupied I-tube, which is a pipe where the risers are pulled through the turret. This thesis will investigate and design a similar, but a permanent solution that could be deployed through the I-tubes on the Skarv FPSO.

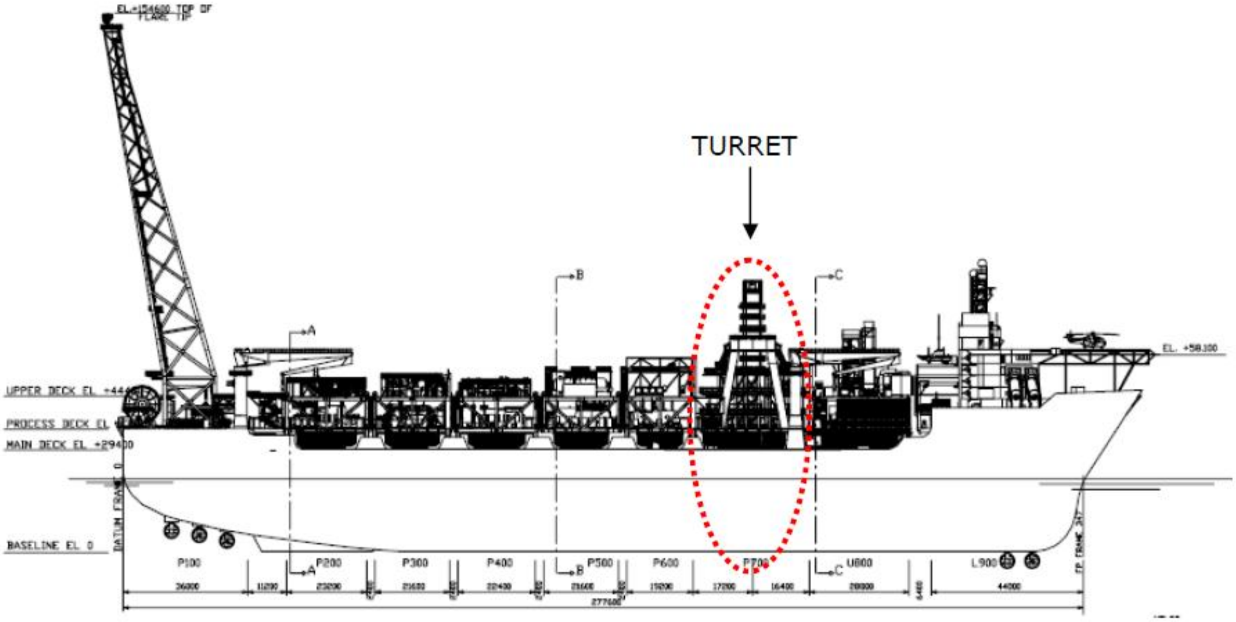

Figure 1-3: The monitoring system can be installed close to where the risers protrude from the Skarv FPSO turret (BP Norway, 2009).

The Skarv FPSO (illustrated in Figure 1-3) is a turret-moored FPSO, connected via flexible risers and flowlines to five templates at a water depth ranging from 325 to 375 meters. The field, which includes Skarv A, Skarv B&C. Tilje and Idun drill centers has an anticipated field life of 25 years with a startup in 2012. It is located in Norway, west of Sandnessjøen and is the newest field operated by BP Norway. The field is going to export oil and condensate with tankers and gas through an export pipeline to the Åsgård transport system (Subsea7 Norway, 2011).

In this thesis, we are looking into a monitoring system intended to fit dimensions and requirements on Skarv FPSO. The riser, turret and I-tube arrangement for the Skarv FPSO is illustrated in Figure 1-4.

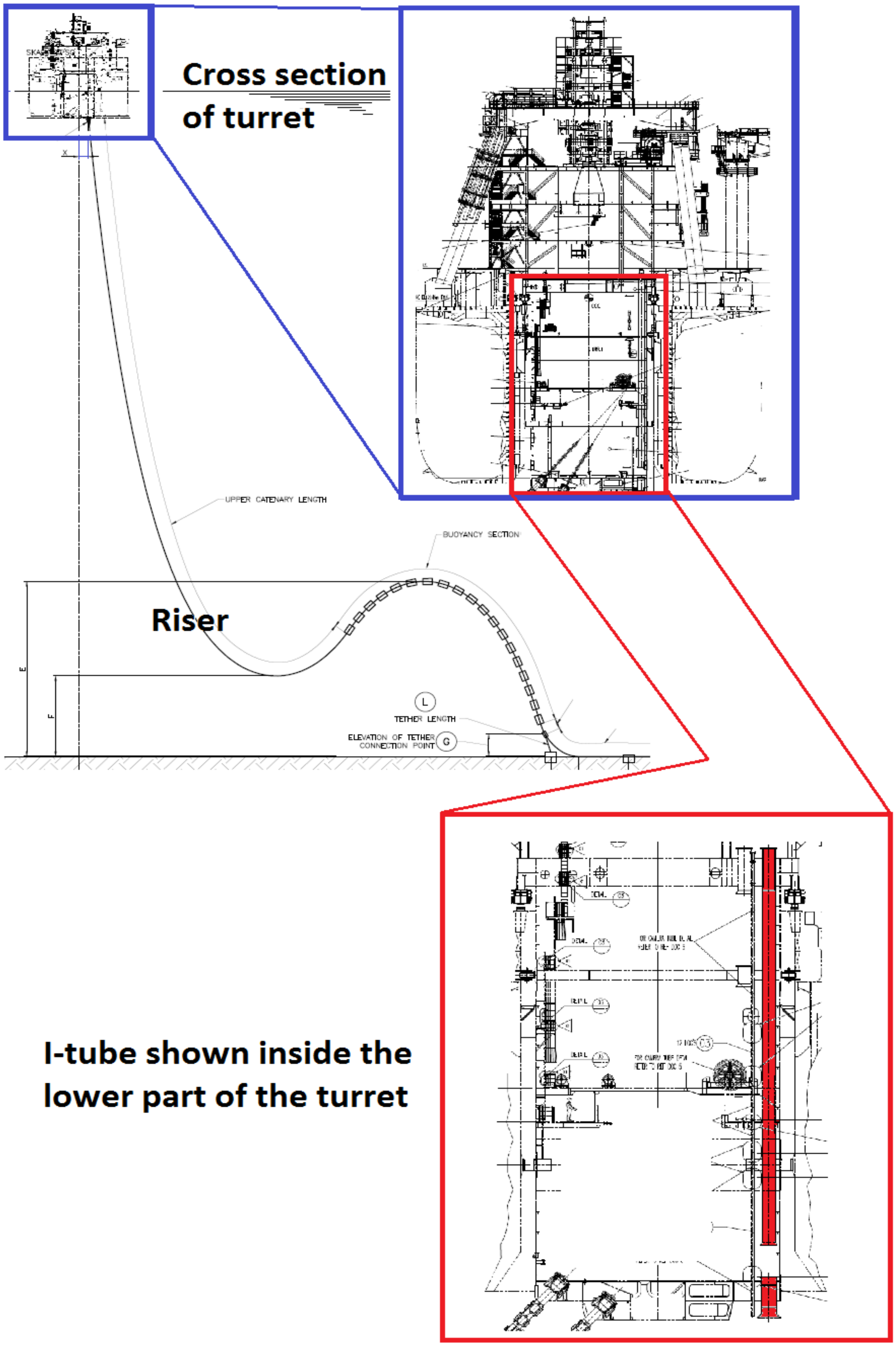

Figure 1-4: The I-tube Position inside the Skarv FPSO Turret (BP drawing archive, 2007-2012).

1.3 Problems and Objectives

The main objective of this report is to develop a solution for a riser monitoring deployment system. The system needs to meet given requirements and criteria for monitoring and operation. The main objectives for this report are as following:

1. Discuss and evaluate different solutions for riser monitoring.
2. Discuss and evaluate different solutions and designs for a deployment system.
3. Evaluate available installation methods and locations.
4. Analyze environmental loads applied to the preferred design.
5. Analyze the structure capacity for given requirements.
6. Discuss the calculation results and identify improvements that could be done.

Since this report is worked out with a given time frame, every aspect regarding product development is not included. With this in mind, the following limitations will give a better understanding of what is expected of this thesis.

1. Drawings are not intended to be fabrication drawings. It is only an early stage proposal of how the equipment could be designed.
2. Detailed capacity analysis of welds and joints are not a part of this thesis.
3. Analysis regarding the hydraulic components and system is not a part of this thesis.

The goal of this report is therefore to evaluate and discuss different solution to create an idea that could, with further work, be fabricated and used at Skarv FPSO.

1.4 The Report Structure

The thesis considers an actual problem, and then finds a solution as a primary goal. The report structure will be reflected by this. It is built up systematically by the different report phases, which is illustrated on Figure 1-5.

The monitoring system is divided into two parts. The first part looks into the sonar device and how the to monitor the risers position. The second design part is covering the deployment system that is holding the sonar head. Both parts will cover different solutions and an evaluation of the intended design. Next, the theory chapter provides an overview and a description of the challenges and loads that are experienced by the monitoring system. In chapter 3, the results and analysis regarding the calculations of environmental loads and the structure response are presented before the last two chapters cover a discussion and conclusion.

Each of the chapters and main chapters/sections, have a short introduction to give the reader an overview of what content can be expected. The main chapters/sections also provide a short summary at the end to highlight the most important content.

The process plan use for this thesis can be found in Appendix E.

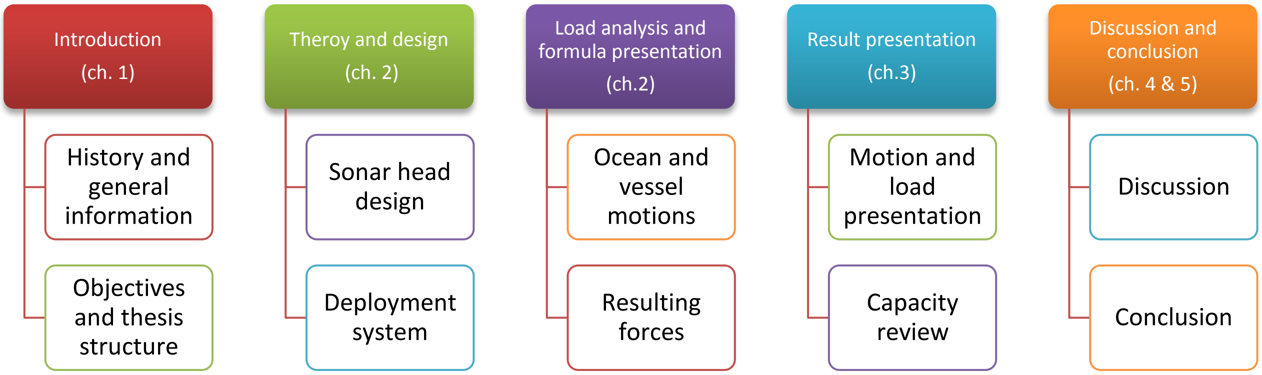

Figure 1-5: The report structure.

1.4.1 Source criticism

In the early stages of this thesis, a literature search on sonar head devices was researched through internet and by discussion with engineers in BP, who have experience with monitoring equipment. Most of the sources around the deployment system are based on drawings and reports created internally in BP, and therefore not published in any way. This contains information of BPs Skarv FPSO as well as reports from the previous attempt of a riser monitoring system, tested on Foinaven FPSO. On the Foinaven trial, BP claims that it was the first time this type of online position system was tested. Therefore, there is very small amount of literature around topic.

Since the thesis also uses BPs FPSO as references to dimensions and behaviors, it could result in a subjective judgment in relation to competitors and give a competitive advantage.

1.4.2 Method

This work done in this thesis has been carried out through the spring of 2012. Information gathering and report research has been done before various ideas and solutions have been evaluated. Discussions and meetings with supervisors or experts for different areas has been an important asset in gathering enough information to write this thesis.

1.5 Chapter Summary

In chapter 1, we have been introduced to background knowledge and information of the usage of FPSOs and why riser monitoring is needed. Information of the Skarv development field and the Skarv FPSO has also been given. The main problems and objectives have been presented on the background of the need for riser monitoring. At the end, the structure and method of this thesis is given to create a better overview of the thesis.

2. Theory and Design

The need for monitoring of the bend stiffener integrity is the background for this thesis. In this chapter, we are looking at the theory and design for how the monitoring device could be developed. This chapter is divided into three sections where the *monitoring device*, *deployment system* and *design loads* theory are discussed. The main section is the deployment system and where the most workload is done.

2.1 Monitoring Device

Before designing a new device, we look at different concepts of monitoring systems. This will be done to find or eliminate already existing technologies on the market. In this chapter, we will look at different types of monitoring systems that have been used in earlier projects to find out if the technology is suitable for this case.

2.1.1 Design criteria

The transducer head is the actual monitoring equipment. This will be located on the lower end of the deployment system. Several types of different subsea monitoring systems are available on the market. The challenge for this thesis is to find the equipment and supplier who will give the best results. To choose the monitoring equipment that fits the purpose best, it is important to evaluate following criteria and requirements:

1. The system needs to give a live feedback to control center.
2. The system needs to be sensitive to movement and give accurate results
3. The radius of surveillance needs to cover all risers at a certain depth below the turret
4. To ensure the system shall stay intact, it needs to be robust and easy to maintain.
5. The size of the device needs to be suitable for installation and retrieval

2.1.2 Background and alternatives

Riser monitoring provides the operator with valuable information to confirm the integrity of the risers, assist with operational decisions, optimize inspection, maintenance and repair schedules and procedures and calibrate design tools. The riser monitoring tools can be classified into two broad categories: Condition monitoring and structural response monitoring (Chezhian & S Meling, 2007).

Structural response monitoring is connected to dynamic response of the riser, such as vortex-induced vibrations and wave loads. In the output from such monitoring system, loads and stresses applied to the risers can be controlled at all time. These types of systems are often more complex than condition

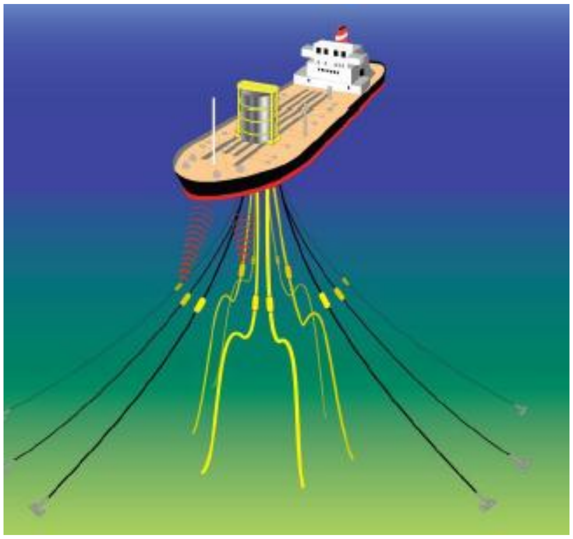

Figure 2-1: Optima-Wireless sensors mounted on risers. (WFS, 2012)

monitoring, and often involve several components placed all along the risers (Chezhian & S Meling, 2007).

Condition monitoring, which is more applicable for this thesis, consists often of one or few components to monitor temperature, pressure, position, top tension and so on. From the introduction we know that the primary objective is to monitor the bend stiffeners to ensure their integrity at all times. This will lead us into the next objective for this chapter; what possible solution can be considered?

According to Muthu Chezhian, DNV project manager, “A significant number of riser monitoring campaigns have been carried out in the last decade, and there is a plethora of experience that can be used for the benefit of future campaigns and assessments.” (Chezhian & S Meling, 2007). With this information in mind, it should be possible to select a device that serves the purpose. As we mentioned in the introduction, a similar device has already been tested at the Foinaven FPSO. We will further look into this equipment and compare it to other alternatives.

The mounting position is also important to evaluate. Two different alternatives could be relevant for this purpose:

1. Acoustic sensors mounted on each riser that gives relative distance to a main control unit.
2. Sonar head that measure positions and movement relative to the vessel and are directly connected to the control center.

From chapter Design criteria 2.1.1, we stated the first criteria as live feedback to the control center. By using the alternative 1, it would be harder to establish a real time link to output screen. In order to measure position, multiple acoustics sensors need to be fitted and put on the right position on each member as illustrated on Figure 2-1. They need to be fitted before deployment, by divers or ROVs. The communication to the surface is achieved by acoustic telemetry. In case of an FPSO with many risers and mooring lines, the complexity of this can be very expensive. Other downsides to this type of acoustic equipment are slow communication compared to real time equipment directly connected to the control center, Interventions by ROV or divers, which is risky for riser integrity or to the diver himself (Tritech International, 2012).

Real time targeting monitoring equipment, deployed through one of the I-tubes inside the turret, can

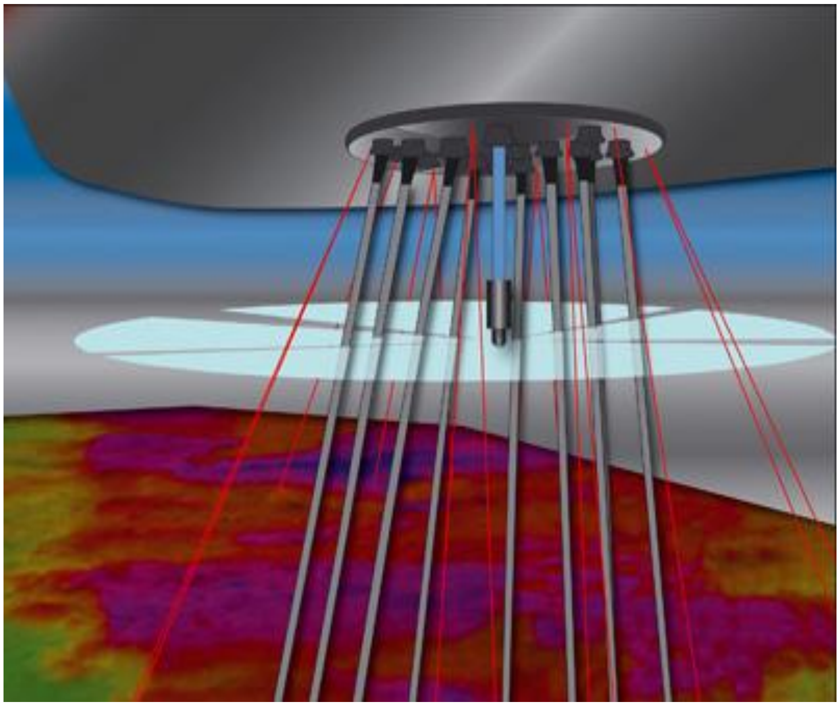

Figure 2-2: Real time monitoring system deployed thorugh the I-tubes (Tritech International, 2012).

be a supplement or an alternative to other acoustic sensors. Alterative 2 seems to be less expensive and more reliable. Further in this thesis, we are looking at the real-time monitoring system connected directly to the control center.

2.1.2.1 Tritechs sonar head used on Foinaven FPSO

On Teekay's Petrojarl Foinaven FPSO, BP has paid significant attention to monitor and maintaining riser integrity to the FPSO. (Kaye, 2008) Their requirement was to have an automated system to monitor bend stiffeners, risers, anchor lines and umbilicals. The system was designed to register movement in the members, relative to the FPSO turret.

This was done by designing a transducer head which could provide a 360˚ view and the ability to detect multiple targets close to each other. The Transducer head was controlled by software, which runs on a dedicated SCU. The software provided a real-time image of all riser positions and would set out an alarm if a riser moved out of a specific target area. The technology proved its value when BP recognized that one of the bend stiffeners had loosened from its position and resulted in larger movement of one of the risers.

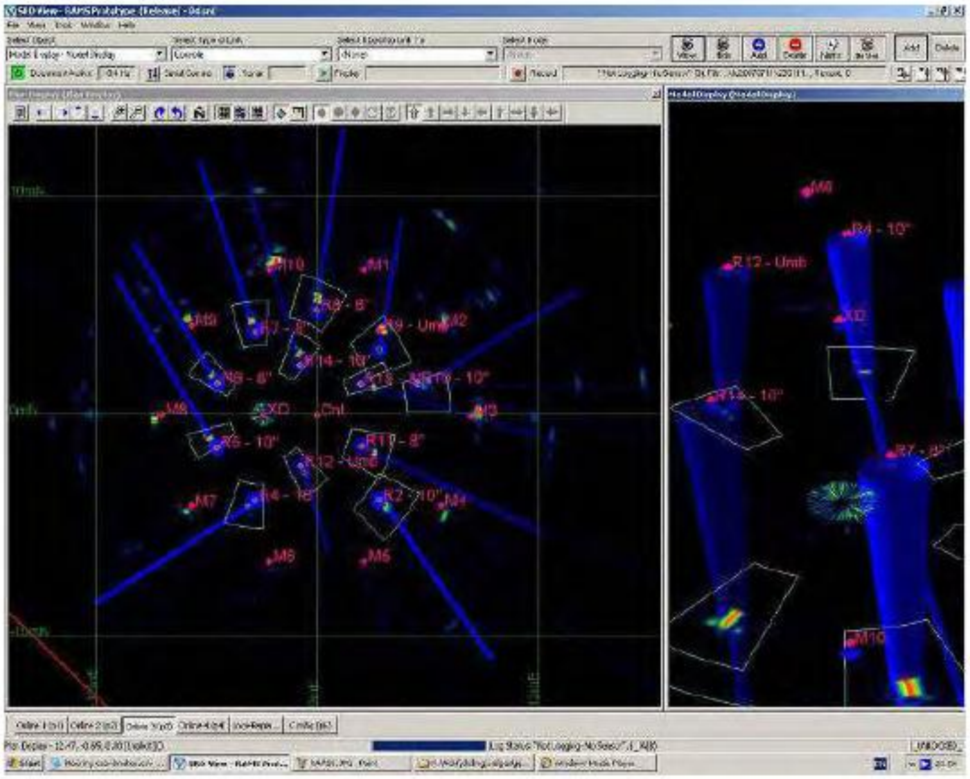

Figure 2-3 Position and movement limitations (Kaye, 2008)

The Transducer head in this case was designed by Tritech International, whom in their brochure introduce the sonar head as (Tritech International, 2012):

"Riser Anchor & Monitoring System (RAMS) is a 360° riser and anchor chain monitoring system for Floating Production Storage and Offloading Units (FPSOs); it is deployed beneath the vessel and monitors the presence, integrity and position of mooring lines and risers 24/7 from a single sonar head.

- *Deployed through the FPSO turret (ideally in the center of the risers and mooring chains), the RAMS sonar provides simultaneous real-time feedback on the status of all lines.*
- *RAMS is a dual-function system, monitoring the presence and integrity of mooring lines and the presence and position of risers from a single sonar head* deployed beneath the vessel.*
- *RAMS incorporates a unique Beam Steerable Transmitter that allows the system to be configured on installation to ensure the optimum sonar return from the mooring lines and risers to ensure 100% target detection and reliability.*
- *Unlike other monitoring systems for mooring lines the Tritech RAMS system is suitable for long-term deployment capability as it has no mechanical moving parts.*
- *Continuous data recording allows for detailed data export for offline trend analysis."*

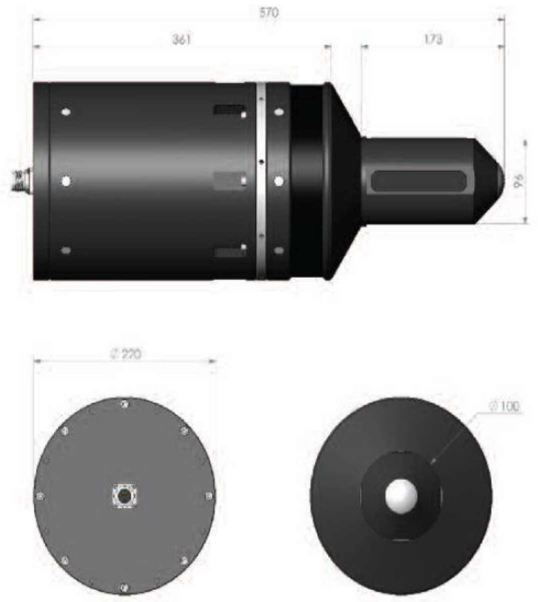

Figure 2-4: Tritech Sonar head (Tritech International, 2012).

2.1.2.2 Sentinel sonar head

The other alternative is the Sentinel sonar head produced by Sonardyne. This sonar head have similar specifications as the Tritech, but with a larger range. Sentinel sonar systems have been used to detect divers or items under the surface of a harbor. Sonardyne describe the system as (Sonardyne, 2012) :

"The transmitters themselves are fully programmable and supplied with a number of frequency modulated Doppler tolerant pulses that can be selected via the Sentinel system configuration file.

The compact 1:3 piezo-composite transducer array has 128 separately wired elements, which are used to form 256 equally spaced, receive beams – each with a 1.4° horizontal

Figure 2-5: Sentinel Sonar head (Sonardyne, 2012)

beam width. Software further interpolates these beams to provide highly accurate bearing estimation for the target.

The sonar head also contains the electronics to digitize, baseband, multiplex and transfer the signals received by the transducer, along with control and monitoring software that performs periodic built-in-testing to verify the health of the transducer elements and front-end electronics."

2.1.2.3 Comparing alternatives

For monitoring of the riser positions, the sonar transducer head is to be mounted to a deployment system, approximately four meters below the hull due to riser spreading. This equipment will need to fulfill requirements set for deployment and operational conditions. In this section, we will evaluate the two different sonar transducer heads that are already on the market.

When a specific sonar transducer head is chosen from a manufactorer, a set of requirements to the manufacture is normally needed to carry out a safe installation and make sure the equipment has the functions as intended over time. A given number of field trials with sufficient test results and reports are normally expected by the suppliers. To provide a better overview of the two alternatives, the most important parameters are gathered for comparison in Table 2-1.

Another option is a permanently deployed camera system, which would give excellent visualization. The problem with this solution is the requirement of a human to evaluate the result; Not an automatic alarm.

Specifications	Sentinel Sonar Head	Tritech Sonar head
Largest body diameter	330 mm	220 mm
Length	432 mm	570 mm
Weight in air	45.5 kg	25 kg
Weight in water	18 kg	9 kg
Operating depth	<50 m	<30 m
Detection range	900 m radius	30 m radius
Acoustic cover	360˚	360˚
Effective range resolution	<0.14˚	0.5˚
Target position	1 m at 150 m radius	10 mm

Table 2-1: Comparison of sonar head alternatives (Tritech International, 2012) (Sonardyne, 2012).

2.1.3 Chosen sonar head and specifications

In this chapter, we will be presented with the Sonar heads functionalities and an explanation on why the device is seen as best suited for this task. The main aspect is to receive an accurate and clear picture of the riser positions. This will ensure that any unregularly movements will be caught and trigger the alarm. In the comparison between Tritech and Sentinel's sonar heads, we can see that most of the specifications are very similar except for target sensitivity and range. Tritech's head has a range of 30 m radius, which will cover all the risers. It also has a sensitivity of 10 mm, which is by far better than Sentinel's 1 meter (at 150 m radius).

In the Foinaven RAMS investigation, it is confirmed that the Tritech sonar head is meeting all its expectations (See Appendix A).With the conclusion that Tritech's sonar head will be best fitted for this purpose, we will continue by presenting more information on how this device works.

The Tritech's Sonar head has a unique electronic steerable transmitter, which BP's Thomas Brown has described in the report from Foinaven field trial. (Brown, 2007)

"Some acoustic energy will always be reflected back towards the receiver in the case of a target that has a perpendicular edge assuming all the energy has not been absorbed by the target" (Brown, 2007). See Figure 2-6 *.*

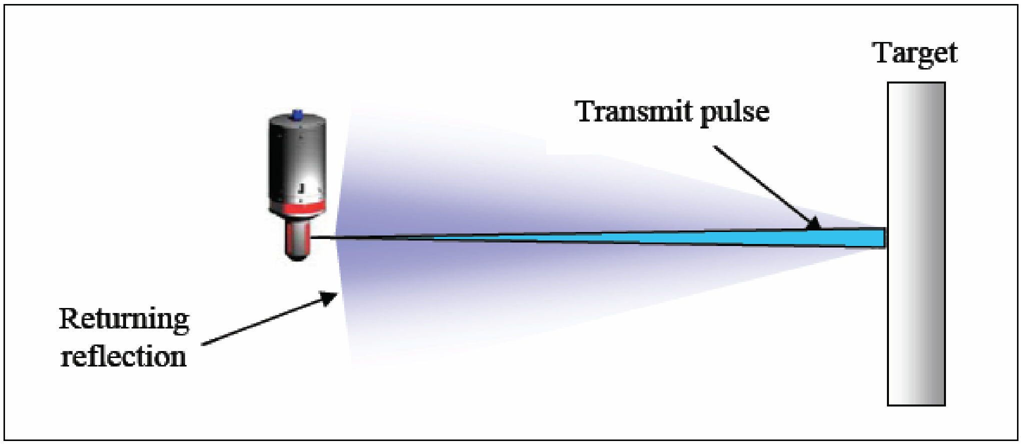

Figure 2-6: Tritech sonar head - Perpendicular targets being acquired (Brown, 2007).

"Generally in cases where the target is not perpendicular to the transmitter/receiver, the majority of energy is reflected away from the receiver but some energy will still be present (see Figure 2-7*). In these cases a higher gain level is required in order to minimize the target accurately. This effect can be minimized using SRD's beam steerable transmitter as discussed in the following section"* (Brown, 2007*).*

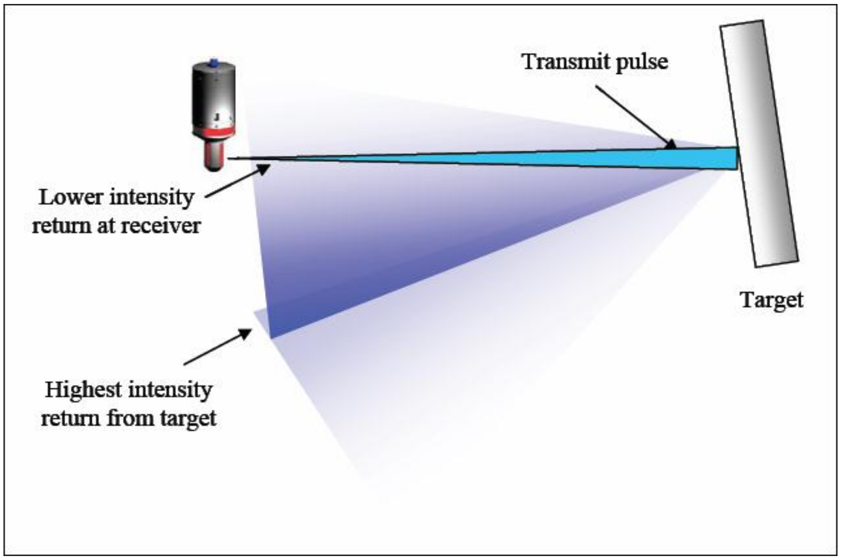

Figure 2-7: Tritech sonar head – Non perpendicular targets being acquired (Brown, 2007).

"To compensate for the reduction in echo strength when the vertical angle of the target increases, the RAMS design includes SRD's unique electronic beam steerable transmitters. These transmitters are capable of steering the transmit beam in the vertical direction to provide a perpendicular reflection from the riser as illustrated in Figure 2-8, thereby using all the available return energy" (Brown, 2007).

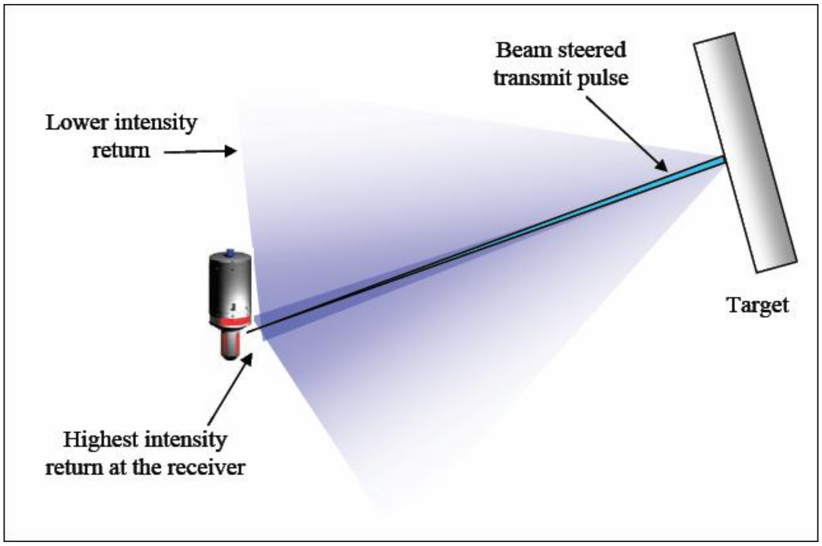

Figure 2-8: Tritech sonar head – Beam steerable transmitter in operation (Brown, 2007).

"Using the beam steerable transmitters not only gives the position of the riser but will also give information relating to the vertical angle of deflection of the riser at the point of measurement, and can be used to determine a "best fit" radius of curvature" (Brown, 2007).

"It is also expected that reflections will be present from other positions on the catenary as some energy is expected to return to the transmitter/receiver position. The intensity of the reflected echo is likely to diminish as the angle of incidence reduces from the orthogonal" (Brown, 2007).

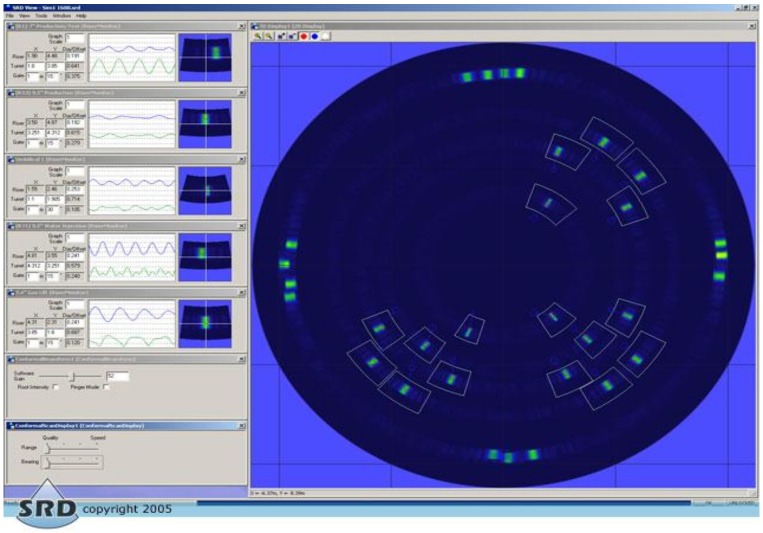

Figure 2-9: Tritech software display of the RAMS GUI installed on Foinaven. The risers position can be seen inside the alarm points (Brown, 2007).

2.1.4 Sonar head deployment system

After a sonar head fitted for the purpose is selected, we need to look at how the equipment is going to be deployed and stay functional over time. The Sonar head needs to be located approximately 4 meters below the hull, in as good view to the risers as possible. To manage this requirement, a deployment system needs to be designed. This is evaluated in chapter 3.

The reason to deploy the system down to 4 meters has its background from the Foinaven RAMS investigation meeting in appendix A. At 4 meters, the risers have been spreading out enough to get a clear view of all risers. The installation position is looked at in more into detail in chapter 2.2.7.3, where installation position of the deployment system is evaluated.

2.1.5 Section Summary

In this chapter, the monitoring device has been evaluated. Theory and information about monitoring technology, as well as an evaluation of alternatives has been presented. The recommended use of Tritechs Sonar head and its specifications has been examined.

2.2 Deployment System

The Tritech transducer head will need to be located in a position that generates the best feedback results. To achieve this, a deployment system is to be designed so that the transducer head will meet its requirements. In this chapter, we are going into detail of the different designs that are evaluated. Design criteria and requirements, material selection, installation properties, safety and maintenance issues will be presented to provide important information for further analysis of the system. The deployment is intended as the main work of this thesis.

2.2.1 Design Criteria and Limitations

In this chapter, a first stage idea of a deployment system is presented. The structure is going to be examined to evaluate if the proposed deployment tool can meet the given design criteria. Before looking at the different alternatives, we need to have a clear understanding of what the design criteria are:

1. Based on previous field trial on Foinaven FPSO, the Tritech transducer head had its best results at 4 m below the hull. For better estimation, field trial on Skarv FPSO is needed and therefore not included in this thesis.
2. Because of sensitivity reasons, the transducer head should not have a deflection of more than 10 mm relative to the turret. Since the transducer head is intended to be located at the lower end of the deployment system, the main pipe structure should be ridgd enough to meet this criteria.
3. The deployment system is to be designed to be functional through the lifetime of the field. Skarv development field has a lifetime of 25 years. Normal procedure is to design for 10 or 100 year condition, which in this case is most conservable to design for the 100 year condition.
4. The structure should be designed in a simple way. This means that it should not contain too advanced or expensive material/components.
5. The fabrication material needs to be robust enough to withstand both the design life of 25 years and the 100 year sea conditions.
6. The deployment system is to be designed such that it can be deployed inside the I-tube without causing any harm to other components or items already positioned at the location.
7. The deployment system is to be designed such that it can easily be retrieved for maintenance or relocated.
8. For safety reasons, the deployment system should not contain any material, fluid or sharp edges, which could harm people through installation or maintenance.

2.2.1.1 Limitations and simplifications

Due to limited time and experience for this thesis, the thesis only covers the development of an early stage idea. Therefore, the design and analysis will not cover all aspects that are normally included in a fully developed design report. Following limitations are presented to give the reader an overview of what level of detailed design and analysis that can be expected. Contents and areas that are only partially or fully excluded are:

1. A normal design report for a new type of equipment would include all detailed analysis from a service company and detailed fabrication drawings. Due to limited time, this thesis will not include detailed analysis, such as bolt and welds capacity. The analysis in this thesis will only provide enough information to decide if the deployment system meets its design criteria and could be developed further.
2. Drawings that are made in this thesis are only for the presentational purpose. Fabrication drawings with exact dimensions are therefore not part of the thesis.
3. Information and calculations of the hydraulic components and system are limited to an overview of what type of components could be used and what pressure is needed. Pressure loss and other hydraulic related problem are not included in this thesis.
4. This thesis will not include transportation, installation and lifting procedures. A simplified presentation containing relevant theory will be presented only to give an overview.
5. Crane and winches for installation and transportation is normally analyzed to ensure the installation process is carried out safely. This is not done in this thesis
6. Corrosion protective measurements such as anodes should be installed at the system but is not to be evaluated in this thesis.

2.2.2 Design principle

To be able to choose the most suitable design, different alternatives need to be reviewed. Since the deployment system is going to be installed inside one of the I-tubes, the basic design is already given. The previous solution on the Foinaven FPSO showed a functional deployment system regarding output and surveillance results (Appendix A, RAMS investigation meeting), but not in the case of the installation equipment (Kaye, 2008).The difference between Foinaven FPSO and Skarv FPSO is the I-tube design. At Skarv FPSO, the I-tube is divided into a lower and upper I-tube, which gives the possibility to use the top end of the lower I-tube as a hang-off section.

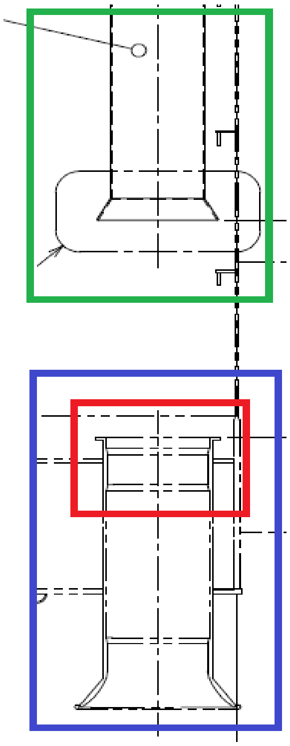

Figure 2-10: Possible mounting positions

2.2.2.1Mounting position

Before the design alternatives are presented, the mounting position needs to be evaluated. The mounting position determines how the deployment system should function and be designed. In addition, the deployment system should be designed for easy release. This way, it can be taken out for maintenance in required time intervals.

The system is to be mounted somewhere inside the I-tube. On Figure 2-10 to the left, the lower part of the upper I-tube is illustrated inside the green box while the lower I-tube is illustrated inside the blue box. The red box illustrates a vulnerable area where the bend stiffener is supposed to be connected, which is described more in detail under section 2.2.2.3. The different mounting positions considered in this thesis are:

1. Mounted inside both upper and lower I-tube
2. Mounted inside lower I-tube
3. Mounted on top and inside lower I-tube

Several aspects need to be considered when choosing the mounting position. Tight and stable connections, in addition to making sure the equipment does not harm any other components are the most important factors. To avoid the gap between upper and lower I-tube, only the lower I-tube could be used as deployment support. This requires checking the capacity of the bend stiffener connection system (BSCS) area. A short capacity analysis is covered in chapter 3.2.2.

By mounting the system in the upper I-tube, it will require a larger deployment system than the other alternatives. On the other hand, the mounting areas are divided on a larger area. The forces onto the lower deployment system will generate movement. For stability reasons, it is necessary to connect the deployment tool on two or more places with some distance in between.

To make the deployment tool smaller and avoid the gap of 2.5 m between upper and lower I-tube, the deployment system can be mounted inside the lower I-tube, below the BSCS. This solution requires larger precision under installation and does not have any boundaries if the centralizers should fail.

The last and preferred solution is to mount the device on top of the BSCS as illustrated in Figure 2-11. This allows the system to rest the self-weight under installation and operation. This solution has therefore an advantage compared to mounting the system below the bend stiffener connection system. To support the resting point, a hang off plate is used, which from now on is referred to as top-hat, some sort of centralizers are needed inside the lower I-tube. Since the diameter of the bend stiffener connection system is considerably smaller than the rest of the I-tube, an expandable solution is needed. The mechanism of the expandable centralizers is supposed to take care of both vertical and horizontal loads. Later in this chapter, we will go more into details on these items.

2.2.2.2 Review of mounting position

By choosing alternative three and mount the system partly on top and inside the lower I-tube, a closer review of the position is required. From the fabrication drawings, we can retrieve the basic information to form a sketch of the outcome. In Table 2-2, the most important properties of the I-tube are gathered for further analysis.

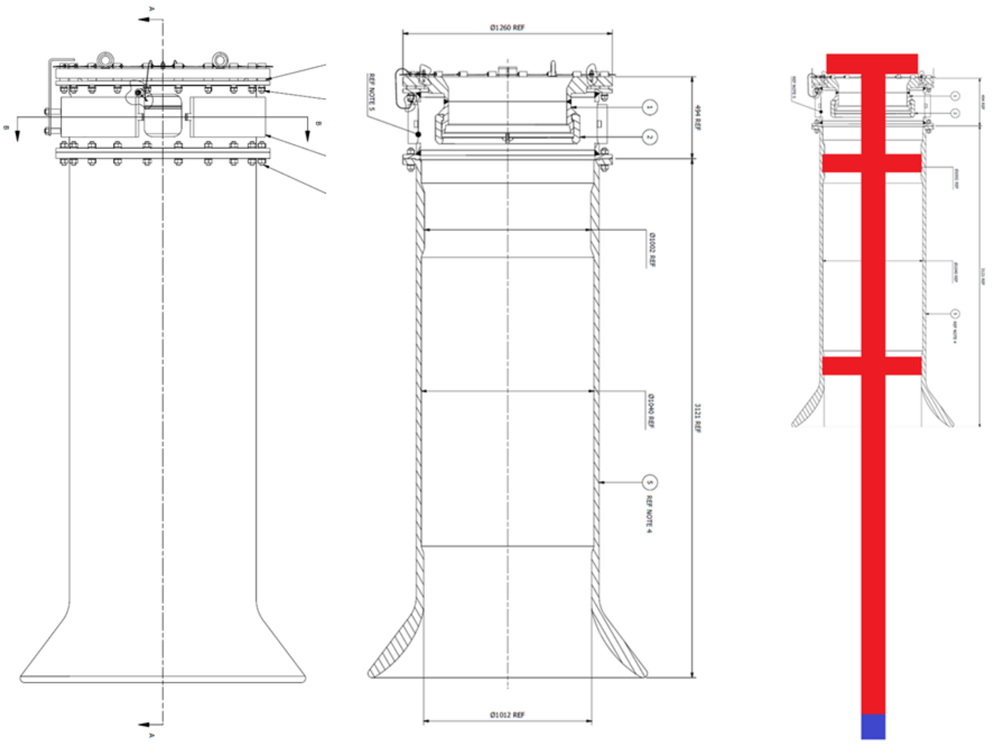

Figure 2-11: Lower I-tube illustrated by fabrication drawings, cross section and intended design position (red). The transducer head (blue) will be located 4 meters below the I-tube (BP drawing archive, 2007-2012).

Component	Dimensions
I-tube	ID: 1002/1040 mm
	OD: 273 mm
	WT: 30 mm
	Length: 3121 mm
Bend stiffener Connection system (BSCS)	ID: 702 mm
	Length: 494 mm
Total	**Length: 3615 mm**

Table 2-2: I-tube and BSCS properties (BP drawing archive, 2007-2012) .

Figure 2-12 gives an overview of the intended use of the I-tube and the BSCS. The drawing illustrates how the bend stiffener consists of the lower and upper bend stiffener.

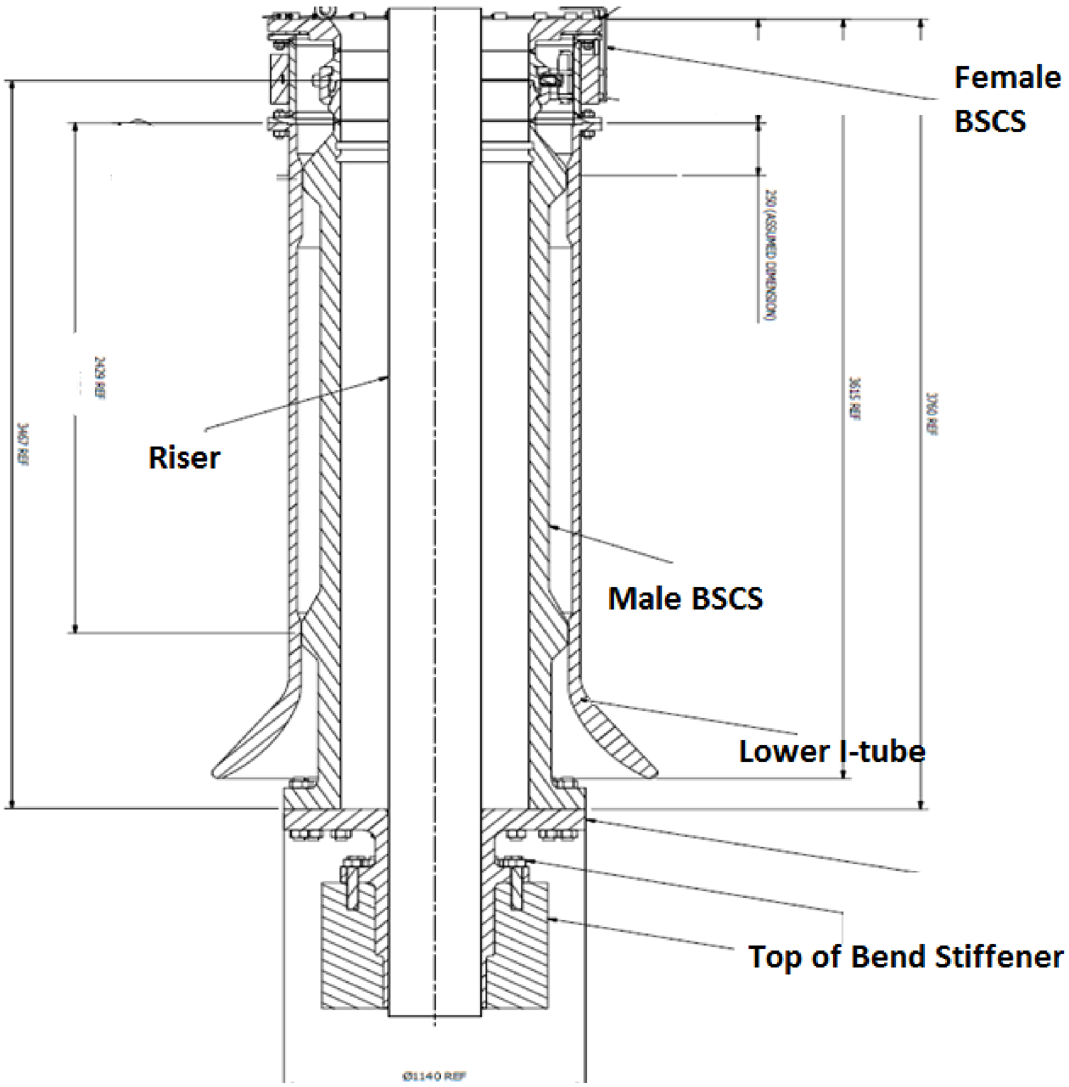

Figure 2-12: Bend stiffener connected inside an I-tube. (BP drawing archive, 2007-2012).

2.2.2.3 I-tube and Lynx Connector review

Regarding the installation of the deployment system, critical areas need to be identified for a closer look. The I-tube, which consists of hard steel, tested and designed for riser pull-in will not be inspected in further details. The bend stiffener connection area, which is used to lock the bend stiffener and keep it in place, has smaller and more vulnerable components. This area is referred as the Lynx Connector.

The Lynx connector is where the bend stiffeners are locked-in during the tie in of the risers. It is the most vulnerable component of the lower I-tube. It also contains the smallest diameter to be considered for the deployment tool.

The illustration on Figure 2-12 shows an occupied lower I-tube with the BSCS mounted inside. The top part is the female lynx BSCS, while the male BSCS and the riser is located inside the I-tube.

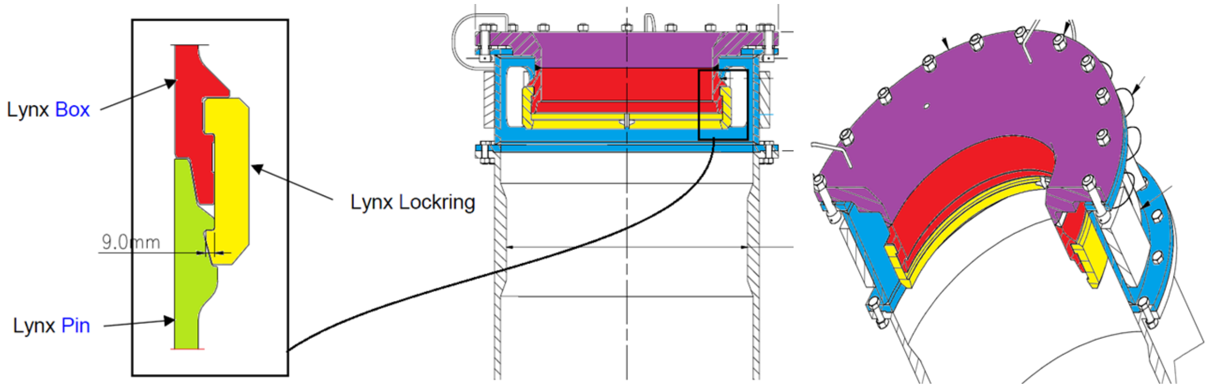

Figure 2-13: The upper part of the bend stiffener connection system is called "lynx connection system" and is the most vulnerable part of the lower I-tube (BP drawing archive, 2007-2012).

The Lynx connection system consists of:

- Lynx box (red)
- Hang off plate (purple)
- Support Stool(blue)
- Lynx lock ring (yellow)
- Lynx pin (green)

The package consists of Lynx box, hang off plate and lynx ring, which is mounted on top of the lower I-tube. The lynx box is welded to the hang-off plate, which is in return installed on the top of the Support Stool with shoulder bolts. To secure movement in all directions, the hang-off plate rests on a nylon washer between the support stool and the hang-off plate. (BP Norway, Technip, Oil States, 2008)

The lockring is the piece of equipment that locks the Bend Stiffeners. For release, the system use hydraulic force to push the Lynx lockring away from the Lynx pin. This releases the vertical force from the Bend Stiffeners. The support stool is where the downwards forces are held. This is a steel structure, installed on top of the lower I-tube. When a riser is installed, this is meant to be the support of the hang-off plate (BP Norway, 2007). A short capacity review is done in chapter 5 to ensure its reliability.

2.2.3 Different design alternatives

Throughout the process of finding a suitable design, different ideas and alternatives have been investigated. Since the deployment system needs to be lowered through the Lynx connection system, all of the design alternatives are based on a pipe structure with expanding devices to ensure fastening inside the lower I-tube.

Since we consider the Lynx connections system as a safe component to use, all of the following alternatives includes a top-hat to carry the structure weight. The following ideas are alternatives to force the equipment to fixed state by using pressure to lock the equipment inside the lower I-tube.

2.2.3.1 *Alternative 1 of 3 – Pressure balloon*

The first alternative has its ground principle form pressure. The idea is to use gas/air pressure or hydraulic fluid to expand a rubber balloon after it has passed the lynx system. The advantage with this is that you will have a large contact area, and therefore less pressure needed to create enough friction to meet the requirements. Figure 2-14 shows two illustrations of different ideas to an expanding balloon, using hydraulic pressure to expand the balloon inside the I-tube. The idea to the left illustrates a bendable steel frame, while the illustration to the right is an idea of pumping air or fluid into a rubber balloon to expand it.

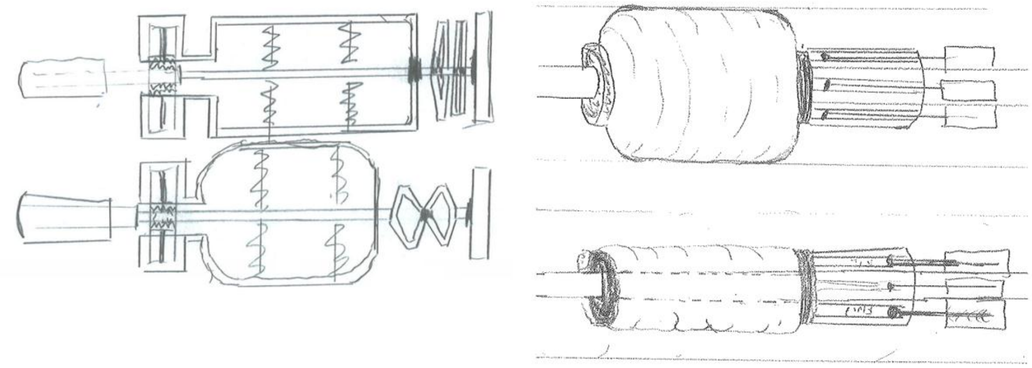

Figure 2-14: Ideas of different pressure balloon designs.

The downside is that the rubber balloon is will not be as robust as steel structures. The solution would most likely fail after some time due to higher complexity and lower material quality. The chances of leakage and failure of pressure form fluid or air is also considered as large. The bendable steel frame will most likely enter a plastic zone and most likely fail over time. The complexity of many small components needs to be considered in relation to cost and maintenance.

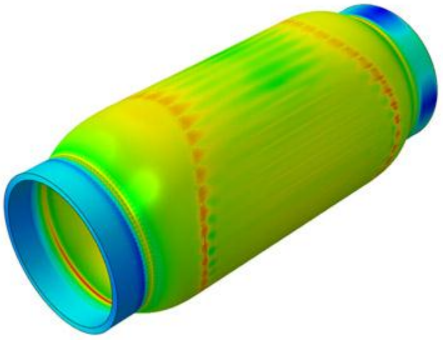

Figure 2-15: The principle of a pressure balloon is to expand it inside the I-tube (readwellservices, 2012).

2.2.3.2 *Alternative 2 of 3 - Mechanical*

The second principle is based on a mechanical where a screw mechanism expands several components or arms onto the inner wall of the I-tube. Several similar techniques are used in the industry with great success. The alternative downsides are that the screw mechanism has its limitations of how much it can expand, and that a considerable amount of force is needed to expand the mechanical component. A screwing mechanism could be manual or machine provided, but the complexity of this compared to a hydraulic system is larger. With a great chance of corrosion, this alternative could easily be stuck due to the steel against steel screwing mechanism.

Figure 2-16: A small version of a mechanical expansion principal (Megaduck, 2012).

2.2.3.3 Alternative 3 of 3– Expanding centralizers

The third and last alternative is based on expanding centralizers. By using hydraulic force, two pistons are pulled towards each other and forcing the connected arms to expand onto the inner wall of the I-tube. A Top-hat will let the device rest its full weight on top of the Lynx connector system. The arms can with this method be expended while the deployment tool is resting at installation position.

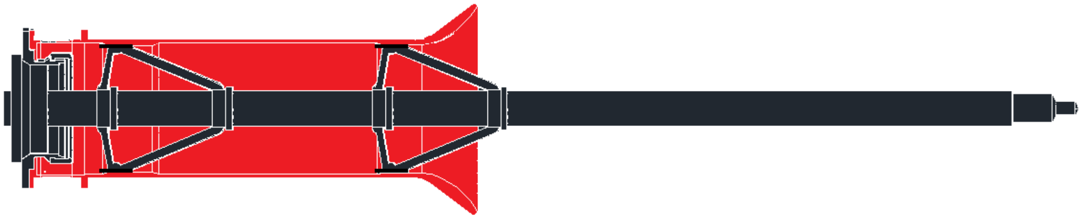

Figure 2-17: The deployment system located inside the I-tube with hydraulic expandable arms.

2.2.3.4 Alternatives summary

When examining the three alternatives, functionality, reliability and maintainability needs to be in focus. All alternatives have an expandable principle which creates pressure to the inner I-tube. The third alternative uses hydraulic pressure to expand the centralizers, something that could be considered as a more stable method than manual power or air pressure. In Table 2-3, a rating grade from 1 to 3 is given to the different alternatives in three different categories.

Alternative	Functionality	Maintainability	Reliability	Total grade
1. Expanding Balloon	2	2	1	5
2. Mechanical	2	1	2	5
3. Expanding centralizers	3	3	2	8

Table 2-3: Expanding method rating

From a brief analysis, Alternative 3 is having most advantages. Due to the hydraulic system and steel arms, it can be seen on as the easiest to maintain and largest reliability. As long as the hydraulic component can provide sufficient power to maintain its position at all time, the functionality is also considered to be high. Alternative 3 also has a great advantage when it comes to lesson learned. The same principle was used on the Foinaven field trial but with no top-hat. From the RAMS investigation meeting (Appendix A) the expanding centralizers were considered as successful. For further analysis, this thesis is considering alternative 3 as the best solution.

The top-hat design, which was included in all alternatives, can be seen as a fail-safe solution. The tool will never fall out of the bottom of the turret while this part is installed.

2.2.4 Components and material properties for preferred solution

The design needs to be evaluated to be used in calculations of loads and capacity. First of all, the material properties and dimensions of each of the most important components will be evaluated. Due to limitations stated in chapter 2.2.1.1, this section will only include a brief overview of the most important components. This is executed in a conservative way to ensure that the calculations around the capacity of the structure can be reliable. A fully developed deployment system will consist of many smaller parts such as bolts and welds. Following presentation is given to ensure a clear understanding of which components that are included.

The structure itself consists of a pipe to provide necessary length to the transducer head. The pipe will be resting on the top of the lower I-tube, using a top-hat to prevent any movement downwards. Two sets of centralizer clusters are added to the construction. This is done to take care of upwards movement and horizontal loads and moment. On Figure 2-18, a rough design proposal is made in order to present a clear understanding of the different components that are taken into account for this thesis.

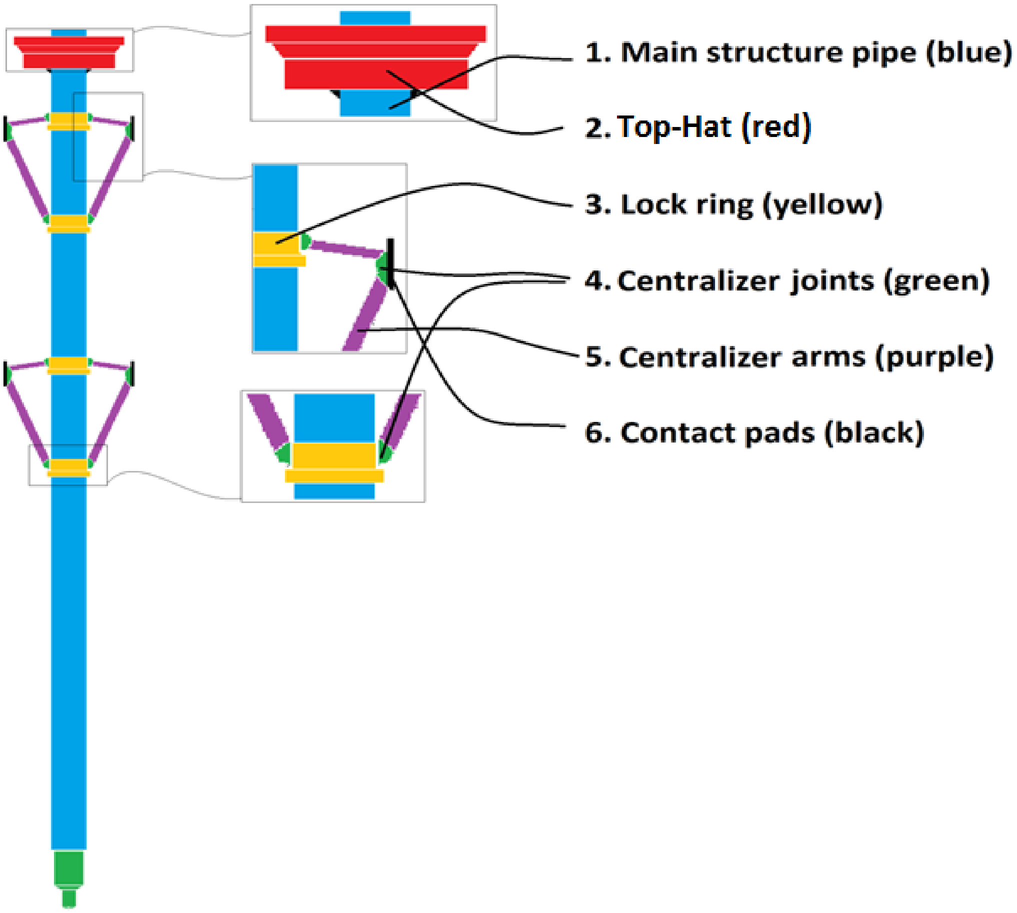

Figure 2-18: The main components of the deployment tool to be evaluated

The deployment system is divided into 6 main components. In the following sub-chapters, it is presented an explanation of each part including material properties is presented for later for analysis.

2.2.4.1 Main structure pipe

The main structure pipe is designed to be a 10 ¾ inch casing pipe. The calculations and analysis will later show whether the pipe needs to be larger or whether we can select a smaller pipe. This pipe is to be seen as the main component of the deployment tool. It will carry the transducer head on the lower end as well as the centralizers and top-hat. In Figure 2-19, a 3D model is made to give the reader a visualization of the system. The main structure pipe is colored blue for this purpose.

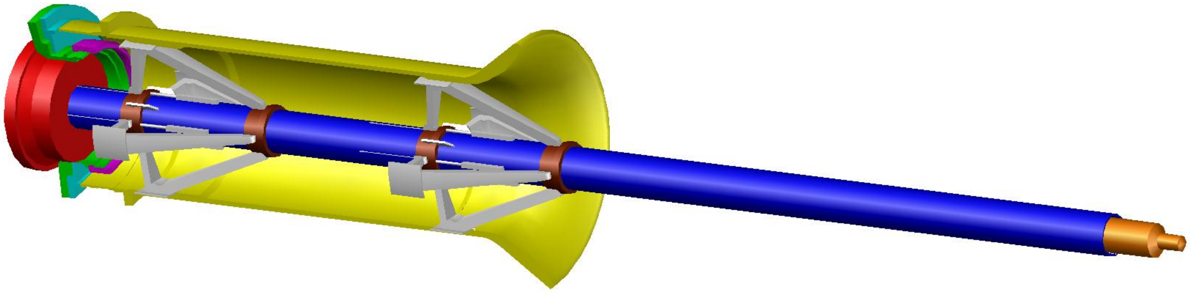

Figure 2-19: The deployment system shown inside the I-tube

The reason for choosing an ordinary 10 ¾ casing pipe is because of the design criteria 4 from chapter 2.2.1. The material properties and dimensions of the pipe are described in Table 2-4 below. The weight of the pipe is listed as 67.71 kg/m^3 for a 10 mm WT (canamservices.com, 2012).

Component	Sizes	Volume	Weight
10 ¾ casing pipe	ID: 263 mm	Steel volume: 0.032 m^3	515 kg
	OD: 273 mm	Geometry volume: 0.445 m^3	
	Length: 7615 mm		

Table 2-4: Material properties of the deployment pipe.

2.2.4.2 Centralizers

The centralizer contains four of the main components this thesis is evaluating. The expanding mechanism, which is driven by hydraulic components, has its purpose to lock the deployment system into fixed position. The centralizers are designed to keep the deployment tool stable against forces in all direction. This part of the component can be seen as the most complex one, and has therefore some limitations to be considered. In this thesis we will not design the centralizers in detail, but rather describe which criteria it needs to meet. These include:

1. Sufficient hydraulic power to provide design pressure
2. Maximum collapsed outer diameter to fit through the lynx bend stiffener connection area
3. Dimensions and materials of the arms strong enough to meet the required capacity.

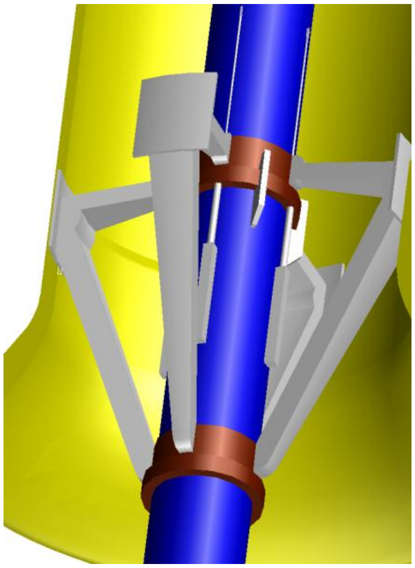

Figure 2-20: Expandable centralizers inside the I-tube.

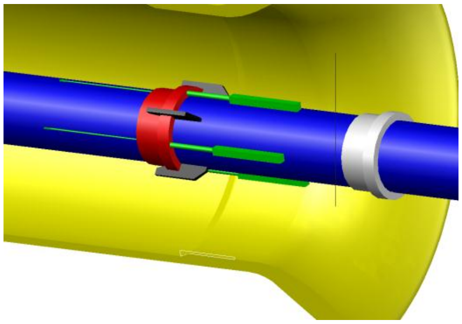

Figure 2-21: The hydraulic components (green) pushes and pulls the ring joint(red) up and down the specific tracks.

The mechanism is provided by hydraulic pressure. On Figure 2-21 the green elements are an illustration of hydraulic components, which are used to force the red element up and down the pipe, guided by welded tracks. This way, the arms (Illustrated in white Figure 2-20) can be expended to create enough force to meet the required stability of the deployment tool.

Centralizers properties

The centralizer consists of many smaller parts. In this subchapter we are going more into details about the dimensions and properties of these. Generally the components are made out of S355 steel with a density of 7850 kg/m^3. (Geocentrix Ltd, 2004). The different parts are:

1. Upper arm
2. Lower arm
3. Upper and lower connection point
4. Middle connection point including contact plate

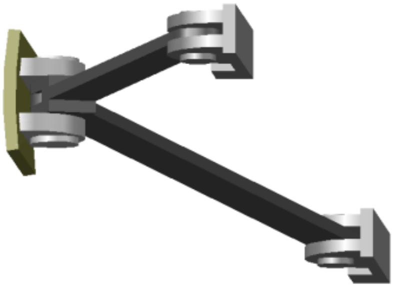

Figure 2-22: Detailed centralizer arm.

Upper arm

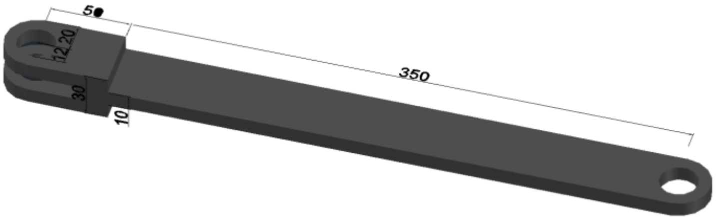

Figure 2-23: Upper centralizer arm.

The upper arm will withstand most of the pressure after the centralizer is fully expanded

Component	Length	Volume	Weight	Material	Yield stress
Upper Arm	420 mm	1.8×10^{-4} m^3	1.41 kg	S355	355 MPa

Table 2-5: Material properties of upper arm

Lower arm

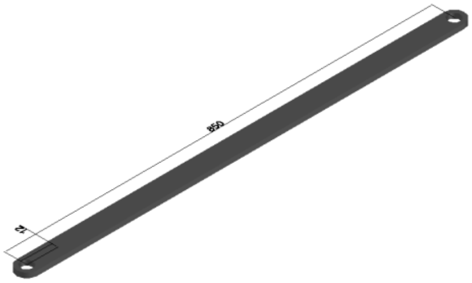

Figure 2-24: Lower centralizer arm.

The lower arm will is rotating on the lower connection point, which has a fixed position

Component	Length	Volume	Weight	Material	Yield stress
Lower Arm	870 mm	3.4×10^{-4} m^3	2.67 kg	S355	355 MPa

Table 2-6: Material properties of lower Arm

Upper and lower connection points

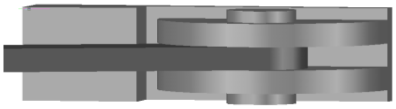

Figure 2-25: centralizer connection points.

The upper and lower connection points are welded to a base ring as illustrated in red in Figure 2-21. The upper connection base ring will slide down towards the lower base ring and force the centralizers to expand

Component	Length	Volume	Weight	Material	Yield stress
2x Connection points	N/A	0.78×10^{-4} m^3	0.61 kg	S355	355 MPa

Table 2-7: Material properties of upper and lower connection points

Middle connection point and contact plate

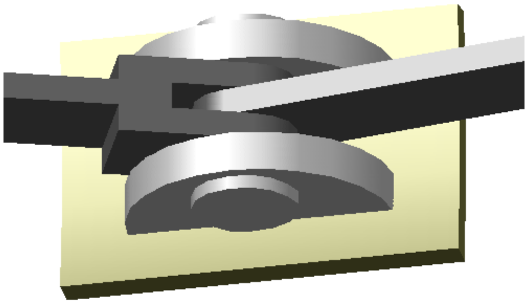

Figure 2-26: Centralizer mid joint and contact plate.

The middle connection point is welded to the contact plate. Both components made out of different materials. The contact plate has a TSA (thermally sprayed aluminum) coating material, which is a good coating material for steel against steel functionality (see Appendix A).

Component	Area	Volume	Weight	Material	Yield stress
Mid Connection point	N/A	0.92×10^{-4} m^3	0.72 kg	S355	355 MPa
Contact plate	0.18 m^2	2.16×10^{-4} m^3	1.69 kg	S355	355 MPa

Table 2-8: Material properties of connection middle point and contact plate

Centralizer looked as one piece

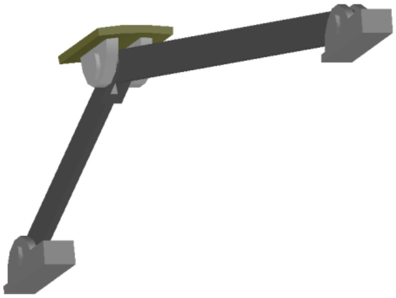

Figure 2-27: Centralizer arm:

A centralizer arm can be looked upon as one component with the following properties

Component	Comment	Volume	Weight	Material	Yield stress
One Centralizer arm		9.1×10^{-4} m^3	7.11 kg	S355	355 Mpa
All of the Arms	In total 8 pieces	72.8×10^{-4} m^3	57.15 kg	S355	355 Mpa

Table 2-9: Total overview of centralizer properties

2.2.4.3 Top-hat

The top-hat will be welded to the 10 ¾ inch casing pipe as the upper fundament and is designed to carry the whole structure weight in addition to downward force. In the installation phase, this component will hold the structure in place while the centralizer arms are expanding.

The top-hat as show in red in Figure 2-28 should be lowered down and installed on top of the lynx bend stiffener connection.

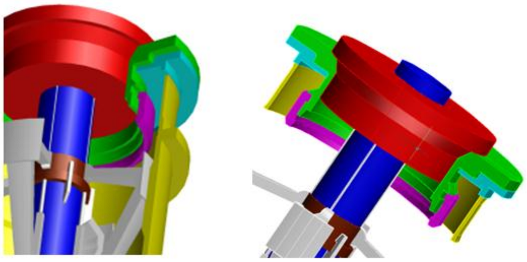

Figure 2-28: Top-hat mounted to the BSCS.

The Top-hat has the same “steel against steel” coating as the contact plates. A TSA layer will ensure a good contact between the BSCS and the top-hat. According to discussion from appendix A, TSA is a coating which is already used on the BSCS as a “steel against steel” material.

Component	Sizes	Volume	Weight	Material	Yield stress
Top-hat	ID: 273 mm	0.007 m^3	54 kg	S355	355 MPa
	OD:702 mm				
	WT:10 mm				
	Height: 180mm				

Table 2-10: Material properties of top-hat

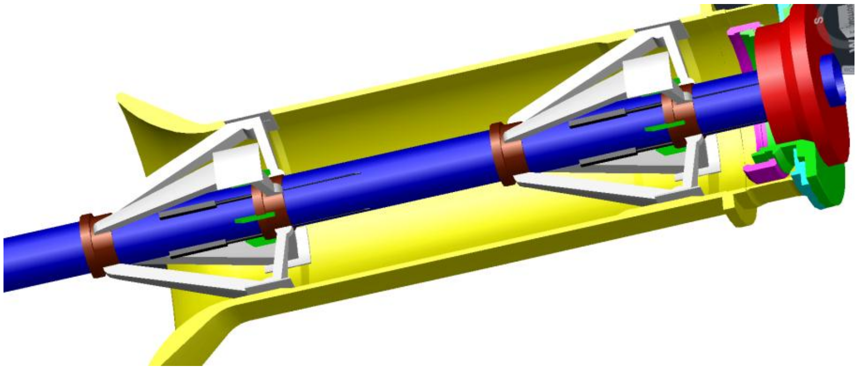

Figure 2-29: Top hat is the upper point of the deployment system (red).

2.2.4.4 Transducer head

The transducer head is described in detail under chapter 2.1.3. The Tritech sonar device is chosen on background of its previous success and sensitivity properties.

Component	Sizes	Weight
Transducer head	OD: 220 mm	25 kg

Table 2-11: Transducer head properties (Tritech International, 2012).

2.2.5 Hydraulics for preferred solution

As stated under limitations in chapter 2.2.1, the hydraulic system will not be analyzed or evaluated in detail. To get a more precise estimate of the deployment systems total weight, a study on witch type of hydraulic component that could be suitable for the intended task, is done.

Figure 2-30: Enerpac RRH hydraulic cylinder series (Enerpac, 2011)

The intended solution is a Enerpac RRH hydraulic cylinder device, which uses hydraulic fluid in both expanding and contracting. A solenoid valve is used to maintain the needed pressure after the hydraulics has reached the preferred position. Today, BP is using hydraulic components on subsea equipment (appendix A) and should therefore be considered as a solid solution. (Enerpac, 2011) The principle of the hydraulic flow is as illustrated in Figure 2-31.

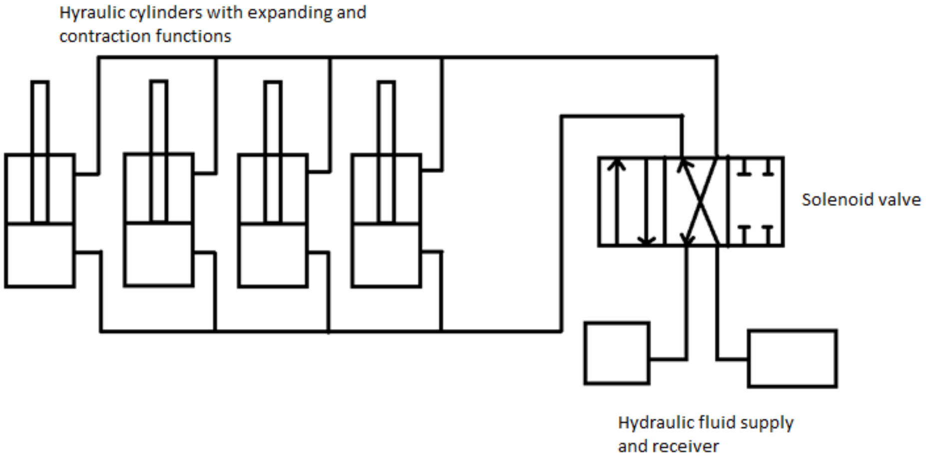

Figure 2-31: Hydraulic flow principle.

Calculations from Appendix B show that a set of 30 kN hydraulic components are needed. The total weight is shown in the table below

Component	Capacity	Stroke	Weight
Hydraulics - Enerpac RRH-3010	30 kN	258 mm	27 kg
Total for all hydraulic components (8x)			216 kg

Table 2-12: Total weight of hydraulic components.

2.2.6 Total weight estimation for preferred solution

As the stability and behavior analysis of the deployment tool requires the total weight, a total overview is given below. Due to the exclusion of smaller parts such as bolts, a conservative estimate of 10% of the total weight is added to take care of extra weight.

Component	Material	Weight
Main Structure pipe	Steel (10 ¾'' casing pipe)	515 kg
Top-hat	Steel (S355)	54 kg
Transducer head	Steel (S355)	25 kg
Centralizers	Steel (S355)	57 kg
Hydraulics	Steel (S355)	216 kg
Total		**867 kg**
10 % weight incensement		86 kg
Total estimated weight		**953 kg**

Table 2-13: Deployment system total material and weight overview.

2.2.7 Installation and safety

The installation process should be carried out in a safest possible way. Thus, a study should also contain procedures on how to install the equipment in the best and safest way. Safety rules are a considerable part of the industry and lots of effort and money are spent on carrying out the safest possible operations.

In this section, following sections are evaluated:

- Requirements for installation and mounting
- Available lifting equipment
- Position of equipment

2.2.7.1 General requirements for installation and mounting

To ensure a fully functional system, we need to have clear requirements for the installation and mounting operation. Therefore, different examples and notes that could be important for the operations are presented:

- The system should be designed in such way that it is compatible with space restrictions inside the turret area. This means that procedure for mounting and installation should take place before a final design. If the equipment is too large or difficult to handle, other alternatives need to be evaluated.
- The deployment system should be able to deploy the system all the way from I-tube entrance deck, down to the lower I tube (Subsea7 Norway, 2011):
    - The I-tube consists of two pipes, upper and lower I-tube. The deployment system should be installed in the lower I-tube, through the upper I-tube.
    - The gap between the lower and the upper I-tube is approximately 2.5 m and should be taken into account as the device could change angle more easily in this area.
    - The deployment tool should fit with good clearance through the upper I-tube, which has a diameter of 1.012 m. The centralizers should pass the BSCS, which has an inner diameter of 0.7 m
- The deployment system should be designed in such way that the on-board installation could be carried out in a safe manner. The manual handling shall follow given regulations and codes regarding handling and lifting.
- Sharp edges and possible threats should be marked with caution signs or most preferable, be redesigned.
- All persons involved in the installation process shall be involved in the planning process and be aware of possible safety issues.
- Inherently safe design.

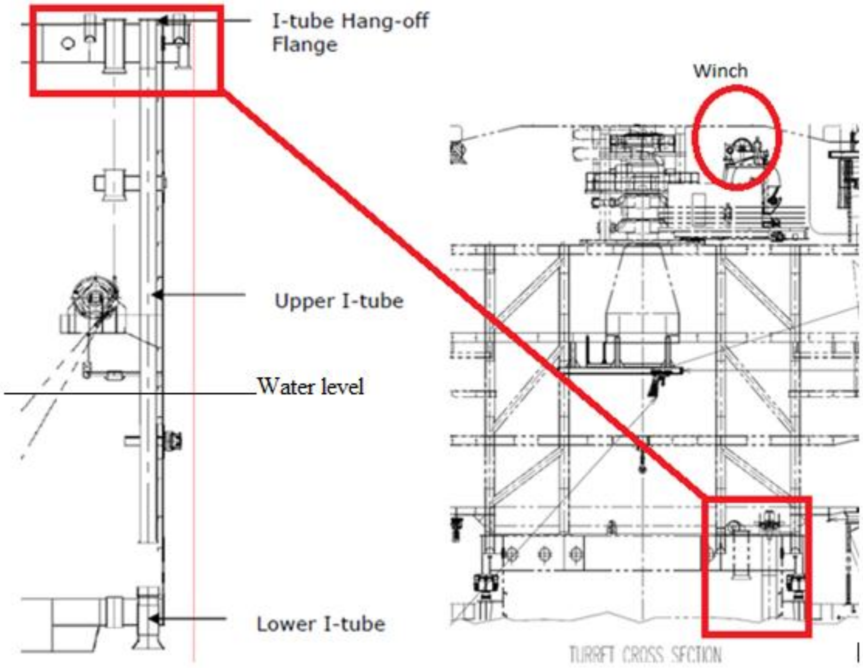

Figure 2-32: Overview of upper/lower I-tube inside the turret (Subsea7 Norway, 2011).

2.2.7.2 Available lifting equipment

Inside the turret, on board Skarv FPSO, there are a number of available winches and power supplies to use during the installation of the deployment tool. For this operation, a 40 ton winch on the top of the turret could be used to support the installation. This winch is located on rotating tracks so it can reach all of the I-rubes (Subsea7 Norway, 2011). Another alternative is to build a special movable A-frame that suits the installation more easily. The lifting and installation procedures will not be covered in detail in this thesis.

2.2.7.3 Position of the equipment

At the time of production start-up on the Skarv FPSO, 13 out of 21 I-tubes will be occupied by risers. In Figure 2-34, the occupied I-tubes are illustrated as red. Even though an occupied I-tube slot is in the shadow of another slot, the riser will be visible for the monitoring system due to riser spreading.

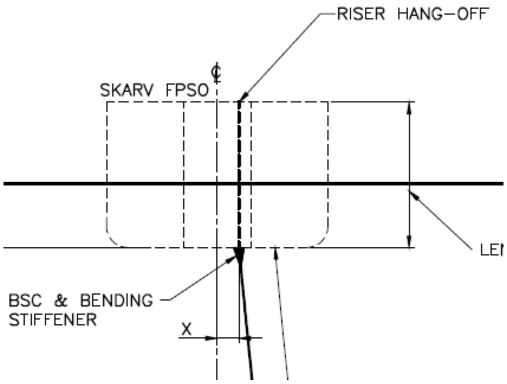

Figure 2-33:The riser will bend and separate from each other illustrated by X on figure (BP drawing archive, 2007-2012).

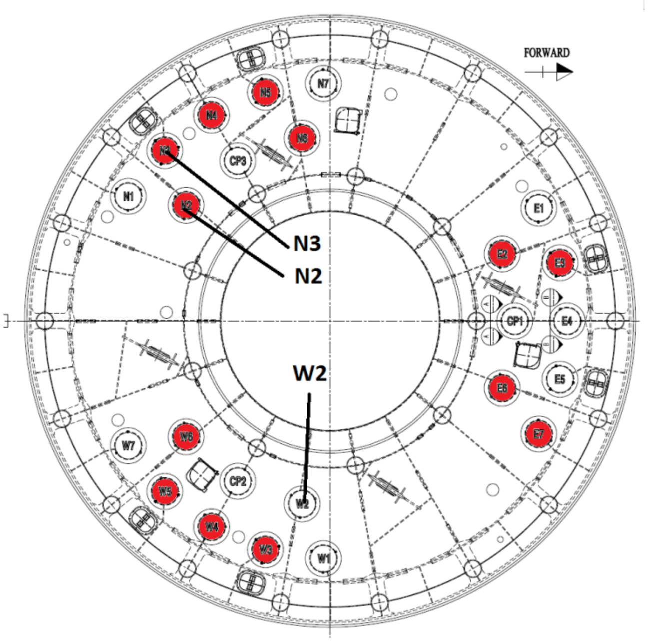

Figure 2-34: Collar deck and I-tube positions (BP drawing archive, 2007-2012).

The main objective when choosing slot is to make sure the transducer head has as clear view as possible to the risers. Slot W2 can be seen as the best alternative, even though N3 is lying in the shadow of N2 as we can see on Figure 2-35.

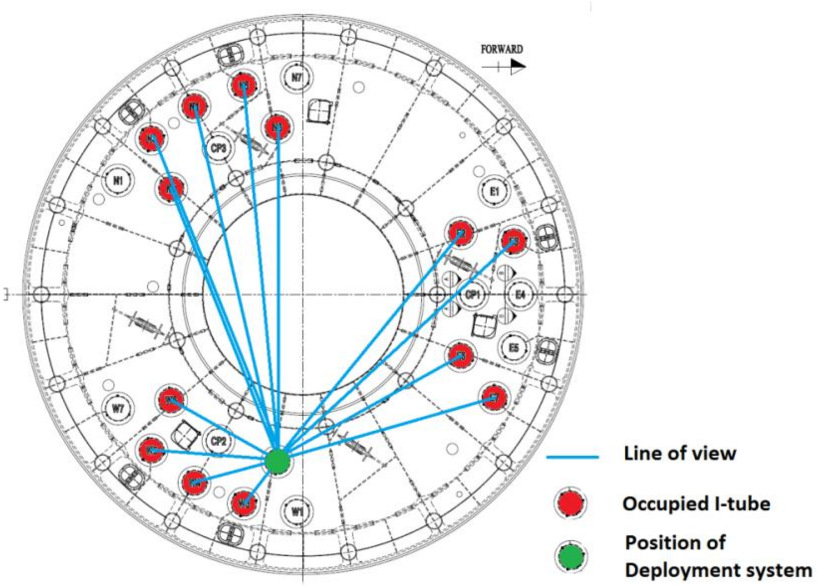

Figure 2-35: Position of the deployment system with line of view to occupied risers (BP drawing archive, 2007-2012).

2.2.8 Integrity management

To ensure the integrity of the system, normal integrity management procedures should be implemented in every part of the deployment systems lifetime.

2.2.8.1 Maintenance

To ensure the functionality of the deployment system through the intended lifetime of 25 years, regular maintenance is very important. Generally, maintenance is required for most offshore structures and equipment. Normally, a risk assessment is established to define the maintenance routines. This thesis will not include a full risk assessment, but will describe a normal procedure to maintain the deployment system based on routines on other submerge equipment.

Normally, the maintenance routine should follow the theory of preventive maintenance, also referred to as "the bathtub curve", illustrated in Figure 2-36. In the startup and early lifetime phases, the equipment will most likely carry some "early infant mortality failures". In other words, events occur in the beginning due to unexpected happenings or miscalculations. After some time, routine failures will demand annual maintenance to the structure. At the end of the lifetime, "Wear out failures" will happen and more frequent maintenance and inspection is needed (Markeset, 2011).

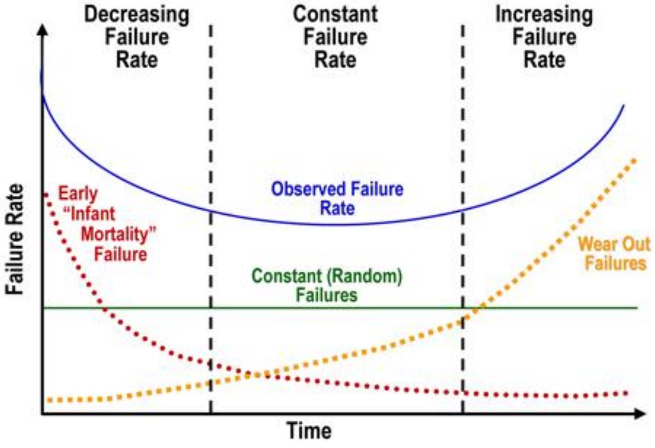

Figure 2-36: Typical example of a "bathtub curve" (Collins, 2009).

In the case of the deployment system, the startup phase should be frequently inspected and monitored for unwanted situations. Following, inspections by either turret deployed camera or ROV camera should be done monthly to ensure the capability of the deployment system. In case of rust and defect components, a larger maintenance and repair activity should be planned and executed at the same operation.

To be able to develop an IMP, other similar cases should be studied to make better estimation of how frequently inspections and maintenance should be carried out. Codes and regulations should be studied to satisfy involved parts, such as operators and owners. This thesis will not go further into an investigation like this.

2.2.9 Section summary

Throughout section 2.2 we have been presented with alternatives and solutions for the deployment system. Based on previous field trials and different evaluations, a preferred design principle is illustrated. Criteria and limitations are taken into account when material properties and design of components are shown. As a part of the theory, brief information regarding maintenance, installation and safety is given to provide a covering overview of the problem.

2.3 Design Loads

Under this chapter, we will look into the different load conditions the equipment will experience through its lifetime. The main purpose of this chapter is to examine how the deployment system should be calculated. This includes a study of how to transfer vessel and ocean motions into forces acting on the equipment.

The theory and formulas that are presented under this chapter will be used for analysis and calculations, which can be found in the Appendix B. Only special formulas closely related to the problem will be a part of this chapter. Theory regarding knowledge is also presented to create a better understanding for readers who do not have hydrodynamics as specialties.

2.3.1 Environmental impacts theory

Wherever an item is located, environmental impacts will affect its behavior. A submerged structure will experience global loads and local impacts. This section is presenting the most applicable environmental loads we can expect the deployment system to experience.

2.3.1.1 Metocean data

Ocean movement determines most of the critical criteria regarding offshore operations. Subsequently, sufficient data needs to be collected before the load and capacity analysis can be carried out. In this subchapter, we will look into different parameters describing extreme sea states followed by a brief explanation of different parameters.

Metocean data is carried out by measuring the different sea state parameters for a specific location. This has to be done by a buoy or a surveillance vessel. The main parameters it is worth paying attention to is the *Tp*, *Hs* and *Maxwave.* According to (Newgard)*, Tp* is described as the peak period and expresses the time interval between the waves. *Tp* is associated with the peak of the wave spectrum, i.e. the wave period at which the highest wave occurs. *Tp* is therefore a good indicator to use together with *Hs* when we are using 10 years and 100 years waves into calculations. *Hs* is expressed as the significant wave height. It is the average height of the highest 1/3 of the waves. This parameter is together with *Tp* the key parameters to use in the calculations.

For extreme condition criteria, *Maxwave* is used to describe the largest wave that will occur. According to most standards and recommendations, *Maxwave* is a result of *Hs* multiplied with 1.8 or 1.6 depending on how clean the swells are (DNV-RP-H103, 2011).

For this report, the main parameter that describes the loads from ocean movement is the current velocity, wave velocity and wave acceleration. The environmental data used in this thesis is based on 6 hour periods for about 55 years from the Norwegian Hindcast Database (Aker Kvaerner, 2008). The parameters used at Skarv development field is presented in chapter 3, where the analysis and results are shown.

2.3.1.2 Waves

To get a rough approximation of the wave characteristics, linear wave theory is used. The theory can be described with three parameters, which is wave period (T), wave height (H) and water depth (d). The formulas used in linear theory depend on the water depth. DNV differs between shallow water and deep water (DNV-RP-C205, 2010), while Professor Ove Tobias Gudmestad have categorized it as shallow water, intermediate water and deep water (Gudmestad, 2011), which is the theory we will follow in this thesis. To determine the water category, following criteria is used:

Water characteristics	
Shallow water:	$\frac{Water\ depth(d)}{Wave\ length(L)} > 0.5$
Intermediate water:	$0.05 < \frac{Water\ depth(d)}{Wave\ length(L)} < 0.5$
Deep water:	$\frac{Water\ depth(d)}{Wave\ length(L)} < 0.05$

Table 2-14 Water characteristic categories (Gudmestad, 2011)

As we can see, the wave length (L) needs to be determined before a category can be decided. The wave length for deep water is described as:

$$L = \frac{g}{2\pi} \cdot T^2$$

Equation 2-1

Where:

L = Wave length [m]

T = Wave period [s]

Further, the velocity potential $Ø(x, y, z)$ of a wave component is (Gudmestad, 2011):

$$Ø(x, y, z) = \frac{\xi_0 \cdot g}{\omega} \cdot e^{kz} \cdot \cos(\omega t - kx)$$

Equation 2-2

Where:

k = Wave number [m^{-1}]

x = Distance of propagation [m]

z = Vertical distance from mean free surface (positive upward) [m]

ω = Wave frequency described by $\omega = \frac{2\pi}{T}$ and $\omega^2 = k \cdot g$ [rad/s]

ξ_0 = Maximum wave amplitude: $\xi_0 = \frac{H_{max}}{2}$ and $H_{max} = H_s \cdot 1.8$ [m] (DNV-RP-H103, 2011)

From equation 4-2, we can derive the formulas of velocity and accelerations. The horizontal velocity is given by: $u = \frac{\partial Ø}{\partial x}$ (Gudmestad, 2011)

$$u = \frac{\xi_0 \cdot k \cdot g}{\omega} \cdot e^{kz} \cdot \sin(\omega t - kx)$$

Equation 2-3

The horizontal acceleration is given by: $\mathring{u} = \frac{\partial u}{\partial x}$ (Gudmestad, 2011)

$$\mathring{u} = \xi_0 \cdot k \cdot g \cdot e^{kz} \cdot \cos(\omega t - kx) \qquad \textit{Equation } 2\text{-}4$$

By taking the $\cos(\omega t - kx) = 1$ and $\sin(\omega t - kx) = 1$, we can derive the maximum velocity and acceleration. Maximum velocity for deep water (Gudmestad, 2011):

$$u = \frac{\xi_0 \cdot k \cdot g}{\omega} \cdot e^{kz} \qquad \textit{Equation } 2\text{-}5$$

Maximum acceleration for deep water (Gudmestad, 2011):

$$\mathring{u} = \xi_0 \cdot k \cdot g \cdot e^{kz} \qquad \textit{Equation } 2\text{-}6$$

As we can see from previous equations, the velocity and acceleration are exponential dependent on water depth (z).

2.3.1.3 *Ocean currents*

Ocean currents occur in many ways. The most common categories of ocean currents are, according to DNV, wind generated currents, tail currents, circulation currents, loop currents and eddy currents. It can be very difficult to predict the profiles of currents. For Skarv Field Development, a report providing field measurement and other oceanographic data is the most reliable. Currents are usually assumed to be dependent on only water depth and if measurements are unavailable, simple models for the design current profiles can be estimated (DNV-RP-C205, 2010)

2.3.1.4 *Hydrostatics*

Concerning hydrostatics we will in this subchapter discuss the hydrostatic pressure and buoyancy. Hydrostatic pressure is taken into account when submerged equipment is evaluated. Pressure increases approximately 1 atm per every 10 m of depth. In this thesis, we will design the equipment to be located at the hull of the vessel. The deployment system is also made of steel, which in combination of shallow water depth can neglect the Hydrostatic pressure as a force parameter.

Hydrostatic buoyancy can be seen as an upward force, acting on all submerged items. Buoyancy force is based on Archimedes law, where the displaced volume of the object is multiplied with fluid density and acceleration of gravity:

$$F_b = \rho \cdot \nabla \cdot g \qquad \textit{Equation } 2\text{-}7$$

Where:

F_b = Buoyancy [N]
ρ = Density of fluid [kg/m^3]
∇ = Volume of object [m^3]
g = Acceleration of gravity [m/s^2]

2.3.2 Hydrodynamics for intended design

The deployment system is located inside and below the hull of the vessel. To describe the relationship between the deployment system and the environmental loads, hydrodynamic theory is fundamental. While the vessel is in constant motion, surrounded by waves and current, water particles are moving around the deployment tool. The hydrodynamic forces are generating horizontal and vertical loads to the structure. These forces that can be calculated with Morison's equation are referred to as drag, lift and inertia forces.

2.3.2.1 Morison's equation

The Morison's equation is, as earlier explained, used to determine hydrodynamic forces on marine objects. The formula uses the fluid velocity and projected area, or fluid acceleration and the objects' volume to respectively calculate the drag force and inertia force.

The Morison's equation is according to DNV applicable when following condition is satisfied (DNV-RP-H103, 2011):

The diameter of the exposed pipe (D) is smaller the one fifth of the wave length (L):

$$D < \frac{\lambda}{5}$$

Equation 2-8

Drag force

The drag force, which appears whenever there is relative motion between the fluid and a solid object can be described as a combination of friction drag and pressure drag. The friction drag is a result of the fluids friction to the objects surface, while pressure drag is formed by vortex shedding behind the object.

Drag force can according to DNV be calculated by using Morison's classic drag force equation (DNV-RP-H103, 2011):

$$F_d = \frac{1}{2} \cdot \rho \cdot C_d \cdot A \cdot v_f \cdot |v_f|$$

Equation 2-9

Where:

F_d = Drag force [N]
ρ = Density of fluid [kg/m^3]
C_d = Drag coefficient [-]
A = Cross sectional area perpendicular to the flow [m^2]
v_f = Relative velocity between fluid and solid object [m/s]

The relative velocity between the fluid and solid object v_r, is the maximum horizontal/vertical water particle velocity, which interferes with the solid body. The drag force F_d, will act in the same direction as the water particle movement. When taking the v_r equal to maximum water particle velocity, the formula describes moving fluids onto a solid body without movement.

Inertia force

Inertia force is the product of mass (including added mass) and acceleration required to accelerate the mass (DNV-RP-H103, 2011). The Inertia force is a result of the objects matter to resist the change of either motion or velocity. According to DNV, the inertia force can be described as:

$$F_i = \rho(1 - C_m) \cdot \nabla \cdot a_f \qquad \textit{Equation 2-10}$$

Where:

F_i = Inertia force [N]
ρ = Density of fluid [kg/m^3]
C_m = Mass coefficient [-]
∇ = Volume of the body [m^3]
a_f = fluid particle acceleration [m/s^2]

Lift force

The lift force will act in a perpendicular direction to the water particle movement. Since it is in this task, only conservatively evaluating considering the maximum load, lift force will not be covered.

Total sectional force

The combinations of drag and inertia force will create the most common form of Morison's equation. In the previous sections we described the drag and inertia as a total force. As a sectional combination to retrieve the total sectional force per meter, DNV describes it as (DNV-RP-C205, 2010):

$$F_s = \rho(1 - C_m) \cdot A \cdot a_f + \frac{1}{2} \cdot \rho \cdot C_d \cdot D \cdot v_f \cdot |v_f| \qquad \textit{Equation } 2\text{-}11$$

Where:

F_s = Total sectional force [N/m]
A = cross sectional area [m^2]
D = Diameter or typical cross sectional dimension [m]

For our case, this would be the correct equation to use if the deployment system did not have any other movement. However, in our case, the vessel will have other motions affecting horizontal forces.

2.3.2.2 Roll motions

A vessel will have its own motion affected by the environmental impacts. Depending on geometric shape, volume and mass, the vessel will generate roll, pitch and heave motions. Pitch can be seen upon as a different type of roll motion and will therefore not be mentioned further in this sub chapter. When the vessel rolls, a combination of velocity and acceleration will be generated towards the water particle movement, creating a larger horizontal relative velocity and acceleration between the deployment system and the water particles. This needs to be taken into consideration when total horizontal load is estimated. DNV ship rules provide a method to calculate the tangential roll acceleration at a given distance from the Meta center (DNV Ship rules, 2010).

The Meta center is defined as the point about which a body starts oscillation when the body is titled by a certain angle (Bansal, 2008). In this thesis we refer to it as the center for rotation. In Figure 2-37, the Meta center is illustrated by M.

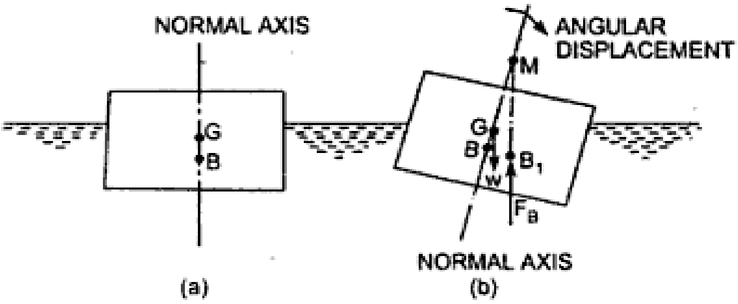

Fig. 4.5 *Meta-centre*

Figure 2-37: Meta center is defined as M and is the center of rotation (Bansal, 2008).

The method for calculating the tangential acceleration goes through 4 steps. First out is the distance from center of object to the center of rotation is defined as (DNV Ship rules, 2010):

$$R_R = \frac{L_o}{2} + R_{gyr}$$ *Equation* 2-12

Where:

R_R = Distance from center of mass to axis of rotation [m]
L_o = Length of protruding object [m]
R_{gyr} = Radius of gyration – Meta center to hull [m]

General roll period is given by (DNV Ship rules, 2010):

$$T_R = \frac{2 \cdot R_{gyr}}{\sqrt{GM}}$$ *Equation* 2-13

Where:

GM = Distance from geometric center to metacenter [m]
T_R = Roll period [s]

DNV further provides a simplified method of how to calculate the max roll angle (DNV Ship rules, 2010):

$$\theta = \frac{50 \cdot c}{B + 75}$$ *Equation* 2-14

Where:

θ = The roll angle (simple amplitude [rad]
c = $(1.25 - 0.025 \cdot T_R)k$ where $k = 0.8$ for ships with active roll damping [-]
B = Moulded breath of the ship[m]

Finally, the maximum roll acceleration can be calculated as (DNV Ship rules, 2010):

$$a_r = \theta \cdot \left(\frac{2 \cdot \pi}{T_R}\right)^2 \cdot R_R$$ *Equation* 2-15

Where:

a_r = maximum roll acceleration [m/s^2]

The maximum tangential velocity is experienced when the ship is at leveled position. According to Professor Ove Tobias Gudmestad, the maximum velocity can be estimated by (Ove Tobias Gudmestad, 2012):

$$v_r = \frac{a_r}{\omega_{roll}}$$ *Equation* 2-16

Where:

v_r = maximum tangential velocity [m/s]

ω_{roll} = Angular roll frequency: $\omega_{roll} = \frac{2 \cdot \pi}{T_R}$ [rad/s]

2.3.2.3 Heave motions

The vessels' heave motions will, in comparison to roll motions, create vertical velocity and acceleration. For most offshore operations, vessel accelerations are used to determine behavior and impact on objects on the vessel. For most vessels, this is calculated and presented in a vessel motion report. For this report, the acceleration is presented by different conditions and therefore describes the total accelerations of the vessel in sea motions. The vertical velocity can be estimated at the same principle as horizontal velocity in equation *Equation* 2-16

2.3.3 Horizontal forces

The deployment system is going to be submerged below the hull of Skarv FPSO. At this position, it will experience loads from waves, current and its own roll motions relative to the water particle velocity.

The loads on a submerged structure are a function of several flow processes (waves, currents and structural movement). These processes will act simultaneously and interact nonlinearly. (Merz, 2010) As shown in chapter 2.3.2.1 the Morison's equation can be used to calculate drag and Inertia forces on three dimensional objects in waves and current.

If the object is moving (roll motions in our case), it is necessary to apply two additional parts to the Morison's equation. The conservative way of evaluating the problem is to state that the roll movement is in directly opposite direction as the water particle flow. This way, the velocity and acceleration are added to movement of the water particle. A moving structure in currents and waves will, according to DNV, experience a distributed force of (DNV-RP-H103, 2011):

$$F_h = \rho \cdot A \cdot C_m \cdot a_r + \rho(1 - C_m) \cdot A \cdot a_f + \frac{1}{2} \cdot \rho \cdot C_d \cdot D \cdot v_f|v_f| + \frac{1}{2} \cdot \rho \cdot C_D \cdot D \cdot r_r|r_r|$$ *Equation* 2-17

Where:

F_h	=	Total horizontal sectional force [N/m]
a_r	=	acceleration of member towards fluid particle [m/s^2]
r_r	=	velocity of member towards fluid particle [m/s]
a_f	=	acceleration of fluid particle towards member [m/s^2]
v_f	=	velocity of fluid particle towards member [m/s]
C_D	=	Hydrodynamic damping coefficient [-]
C_m	=	Mass coefficient [-]
C_d	=	Drag coefficient [-]

By using equation *Equation* 2-11, we should be able to have a useful estimate on the sectional force acting horizontal onto the structure.

2.3.4 Vertical forces

As the vessel follows the sea motions, the vertical movement will generate vertical loads onto the structure. The loads can be described in the same way as horizontal load, where moving water up and down the I-tube will create forces and affect the stability of the structure. The loads will need to be met by the centralizers. In the following subchapters, simplifications and method of how to calculate the vertical loads are examined.

2.3.4.1 Definition of load situation

To be able to calculate the vertical loads onto the structure, it is important to have a clear understanding of the actual load situation. The inside turret can be seen as a special kind of a moonpool since the open I-tubes allow the water to flow in and out. The I-tubes therefore act as the bottom water entrance with the elevating surface level several meters above. See illustration in Figure 2-38 .

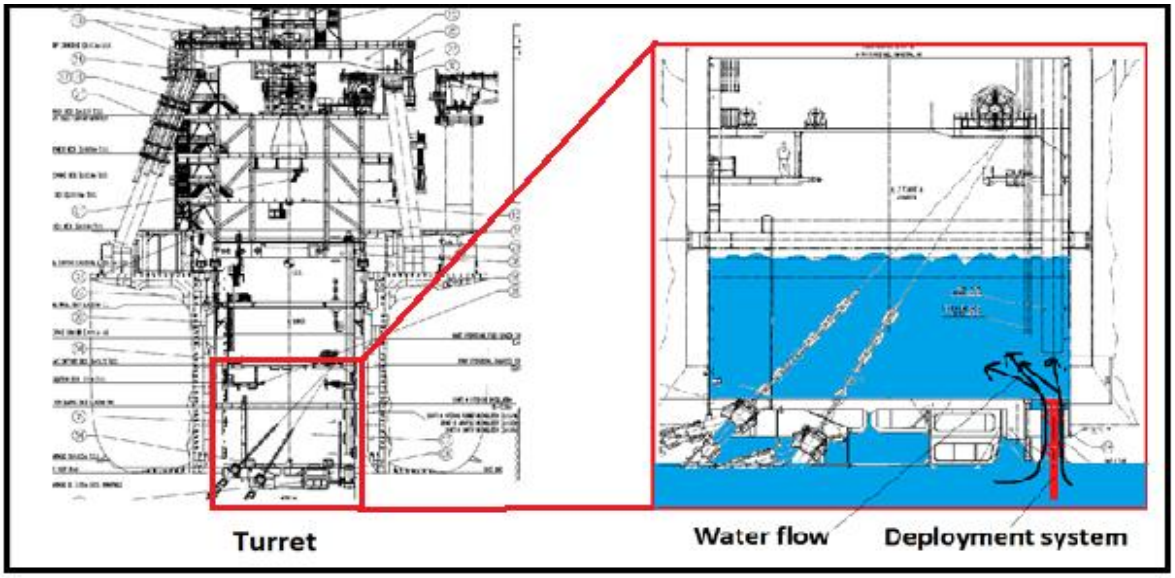

Figure 2-38: Illustration of water flow through I-tube and into the turret

Since the deployment system is located inside the entrance of the water flow, it will experience similar loads as a equipment submerged inside a circular moonpool. Vertical load theory is therefore taken from moonpool operations

2.3.4.2 Comprehensive moonpool calculation method

According to DNV, a comprehensive calculation method should be used if the solid projected area covers more than 80% of the moonpool section area. The interaction force between the water plug and object is found as (DNV-RP-H103, 2011):

$$F_v = \frac{1}{2} \cdot \rho \cdot C_D \cdot A_b \cdot v_r \cdot |v_r| + (\rho V + A_{33})\xi_w - A_{33} \cdot \xi_b \qquad \textit{Equation } 2\text{-}18$$

Where:

v_r = relative velocity between object and waterplug [m/s]
A_b = Solid projected area of object [m^2]
V = Volume of object body [m^3]
A_{33} = added mass of water plug [kg]
ξ_w = Vertical acceleration of water plug [m/s^2]
ξ_w = Vertical acceleration of object [m/s^2]

By using equation *Equation* 2-18, a good estimate of the vertical force onto the structure is evaluated. The equation requires a different way of calculatiing the drag coefficient (DNV-RP-H103, 2011):

$$C_d = 1 - 0.5 \cdot \frac{A_b}{A} \qquad \textit{Equation } 2\text{-}19$$

Where:

A = cross sectional area of I-tube [m^2]

The added mass for vertical oscillation of the water plug is expressed as (DNV-RP-H103, 2011):

$$A_{33} = \rho \cdot K \cdot A_i \cdot (-D) \cdot \sqrt{A_i \cdot (-D)}$$

Equation 2-20

Where:

K = equal to 0.48 for circular moonpool [-]
A_i = cross sectional area of water plug [m^2]
D = depth to entrance of I-tube [m]

2.3.4.3 Maximum total vertical force

The structure is supposed to rest on bend stiffener connection system. Vertical upward loads will work against the self-weight and the centralizer contact plates. The maximum total vertical force to consider is therefore the upward force minus the submerged weight of the equipment. The final force can be described as:

$$F_{vt} = F_v - F_{sub}$$

Equation 2-21

Where:

F_{vt} = Total vertical force upwards to be considering for centralizers [N]
F_v = Interaction force between water plug and object [N]
F_{sub} = Gravity force from submerged weight, see chapter 2.3.1.4 [N]

2.3.4.4 Hydraulic power needed to meet the vertical force

In the previous sections, the equations needed are revealed to analyze the forces action on the structure. The deployment system is designed to meet these forces by pressure and friction force. The principle can be seen in Figure 2-39. The eight hydraulic cylinders are required to create enough horizontal pressure so that the friction force is larger than the total upward vertical force:

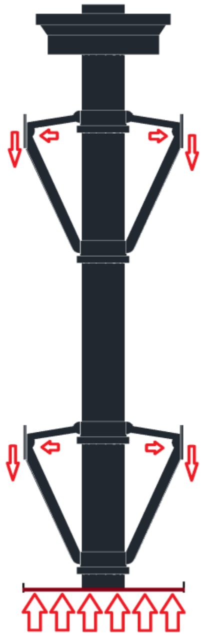

Figure 2-39: Friction force to meet the vertical force

$$\frac{F_f}{S_f} > F_{vt}$$

Equation 2-22

Where:

F_{vt} = Total force upwards [N]
F_f = Friction force created by hydraulic pressure [N]
S_f = Safety factor (chosen from proper regulations) [-]

With background of the European Standard (Standards, 2002) , the safety factor is 1.5 for marine constructions with unfavorable permanent loads.

The friction force F_f is generated by:

$$F_f = F_{hor} \cdot \mu_f \qquad \textit{Equation } 2\text{-}23$$

Where:

F_{hor} = Horizontal force needed to create the friction force [N]
μ_f = Friction factor [-]

The minimum horizontal force per contact plate that needed to meet the upward force is:

$$F_{hor} = \frac{F_{vt} \cdot S_f}{\mu_f \cdot n_p} \qquad \textit{Equation } 2\text{-}24$$

Where:

F_{hor} = Horizontal force needed to create the friction force [N]
n_p = Numbers of pads [-]

By combining *Equation* 2-22 to *Equation* 2-24, we derive the formula for the vertical force the hydraulic cylinders needs to pull at the connection ring:

$$F_{hyd} = \frac{\cos(\beta)}{\sin(\beta)} \cdot \frac{F_{vt} \cdot S_f}{\mu_f \cdot n_p} \qquad \textit{Equation } 2\text{-}25$$

Where:

F_{hyd} = Hydraulic power needed to meet the requirement in *Equation* 2-22 [N]

2.3.5 Section summary

Section 2.3 has presented the main equation theory in this thesis. By using this theory, a solid estimation of a realistic load case can be established. The chapter only contains equations and theory that applies to the actual case and use for calculations in appendix B.

3. Results and Capacity Analysis

In this chapter, theory from chapter two combined with calculations and capacity analysis from appendices are used to present the results.

3.1 Results

The deployment system has its criterion of minimum 25 years lifetime. To ensure the structure can withstand this criterion, an analysis of how the deployment system reacts on a 100 year storm needs to be evaluated.

By using the load theory from the previous chapter, it will in this chapter be shown:

1. The analysis of environmental data; this includes velocity and accelerations of waves, current as well as the vessel itself.
2. The analysis of the forces acting on the deployment system; environmental and vessel motions create forces to the structure.
3. Global capacity analysis of the structure components; each of the main components capacity need to be analyzed to verify the structure functionality.

3.1.1 Geometric data

The thesis has so far been evaluating the intended design of the deployment tool. This important data relates to analysis of environmental impact to the system. Different areas and volumes are important for different calculations. Therefore, the main parameters are presented in Table 3-1

Geometric data	
Exposed Length, L_E	4 m
Diameter, D	0.273 m
Weight	953 kg

Table 3-1: Geometric data of deployment system

3.1.2 Environmental data

Environmental data will in this chapter be analyzed to retrieve the maximum load combination the deployment system can experience. The metocean data is gathered and the maximum water particle velocity and acceleration are calculated. By using equations from chapter 2.3, the maximum load from environmental impact is calculated in Appendix B and presented under this chapter. From the Oceanographic Meteorological Design Data Summary (Grant/BP, 2009) and Motion analysis report (Aker Kvaerner, 2008), various data needed is collected for this task.

3.1.2.1 Load combinations – Waves and current

The waves and current loads are based on extreme conditions, which are gathered from metocean data. According to DNV (DNV-RP-F109, 2010), in permanent operational conditions and temporary phases with duration over 12 months, a 100-year return period applies. In cases where detailed information about joint probability for waves and currents are not available, DNV recommends to approximate the condition by the most severe combination of two points:

1) The 100-year return condition for waves combined with the 10-year return condition for current.
2) The 10-year return condition for waves combined with the 100-year return condition for current.

In addition to waves and current, we have roll-induced velocity and accelerations. The most severe condition is either Inertia forces made by accelerations of waves and roll or drag forced made velocity from waves, current and roll. After calculation of total maximum condition for 10 and 100 year return period, we can decide which of the alternatives are most severe.

To find the most severe condition it is necessary to look into the wave and current data for the Skarv development field. Since the deployment tool is located in the lower I-tube, 4 meters below the hull of the FPSO, data from this depth is found. The draft of the FPSO depends on the vessels load conditions. Further analysis will be calculated on a fully loaded FPSO, which according to table 2.1 in the motion analysis report, the draft is 19.9 meters. (Aker Kvaerner, 2008)

Before further analysis is carried out in this thesis, we have to set the deployment system to protrude 4 meters below the hull. This means that our calculation area is roughly 20-25 meters below surface. As the Oceanographic Meteorological Design Data Summary (Grant/BP, 2009) gives current data at 25 meter steps from surface, further calculations is done at 25 m.

Water depth	
Water depth (d)	369 m
Structure depth (z)	25 m

Table 3-2: Water depth for Skarv development field (Grant/BP, 2009).

3.1.2.2 Waves

The Metocean data to be used for this thesis is gathered form the oceanographic meteorological design data summary report (Grant/BP, 2009) performed for BP in the research of Skarv development. The main parameters that are relevant for this study are:

Return period, years	Hs (m)	Tp (s)
10 years	13.7	16.1
100 years	16.3	17.4

Table 3-3: Metocean data for waves (Grant/BP, 2009).

To evaluate the maximum values, an estimate of $H_{max} = H_s \cdot 1.8$ can according to DNV be used. This gives the following table (DNV-RP-H103, 2011):

Return period, years	Hs (m)	Tp (s)
10 years	24.66	16.1
100 years	29.34	17.4

Table 3-4: Maximum values of wave height according to (DNV-RP-H103, 2011).

3.1.2.3 Current

Extreme total current speed used for the Skarv development project is taken directly form statistical analysis of the Norne measurements. It is found in table 6.1 from the Oceanographic Meteorological Design Data Summary (Grant/BP, 2009). Relevant data is gathered in Table 3-5 below:

Return period, years	Current speed at 25 meter water depth
10 years	0.65 m/s
100 years	0.74 m/s

Table 3-5: Metocean data for current (Grant/BP, 2009).

3.1.2.4 Water characteristic

When looking at wave properties and using linear wave theory, the theory is using different equations for different categories. We use equations from section 2.3.1.2 to calculate the following parameters. Calculations are shown in section 1.1 in Appendix B

	10 years return period	100 years return period
Angular frequency (ω)	0.39 rad/s	0.36 rad/s
Wave length (L)	404 m	472 m
Water Characteristic	Deep Water	Deep Water
k-factor (k)	0.016	0.013

Table 3-6: Water characteristics for 10 and 100 year conditions.

3.1.2.5 Maximum water particle velocity and acceleration

For this chapter, we split up the loads into horizontal and vertical categories. The reason for this is to make it easier to differ between the load and what direction they are applied from.

Horizontal maximum water particle velocity and acceleration is calculated under section 2 in appendix B. Both the 10 and 100 year conditions are calculated by using equation *Equation 2-2* to *Equation* 2-6 in chapter 2.3.1. Including the currents, the total horizontal water particle velocity and acceleration is found to be:

	Total Horizontal velocity	Total Horizontal acceleration
10 years wave and 100 years current return period (10 years acceleration)	4.00 m/s	1.27 m/s^2
100 years wave and 10 years current return period (100 years acceleration)	4.45 m/s	1.37 m/s^2

Table 3-7: Total horizontal wave and current acceleration.

These results only represent the water particle motions relative to a still object. Since this case also depends on vessel motions, the final loads are not estimated only with respect to sea motions.

3.1.3 Vessel motion study

The vessel motion needs to be studied to get a more realistic picture of the loads, which is applied to the deployment system. Heave and roll motions is studied at extreme sea states. The roll motion will affect the deployment system in horizontal direction, while heave motions in the vertical direction.

3.1.3.1 Roll motion

Roll and pitch movements will have an impact to the structure. Since roll and pitch never act from the same direction, we choose to look at roll movement as they have larger acceleration and angular velocity than pitch motions. Firstly, roll motion is created from external force and puts a system (the vessel) into motions. Roll movement has pendulum motions. The maximum forces will act on the deployment system when the direction of roll induced acceleration and velocity opposite to the direction of current and waves. The deployment system will be forced through the water and experience the same forces as if the deployment system was still, and the water had velocity and acceleration onto the deployment system.

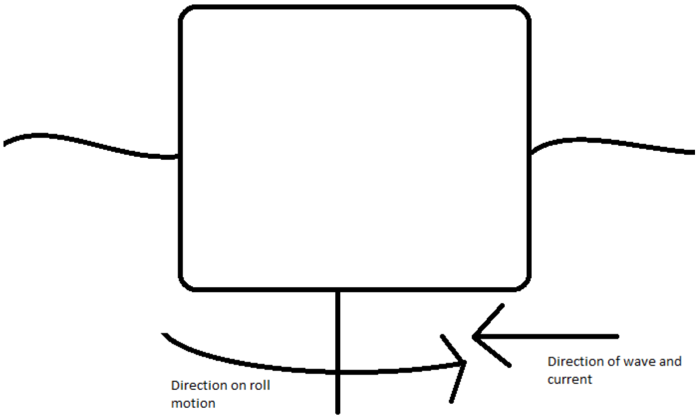

Figure 3-1: Illustration of the roll motions opposite to the wave and current direction.

After the roll motions isolated from waves and current are calculated, we use equation *Equation* 2-17 to combine them. This is done in chapter 3.1.4.1 when horizontal loads are calculated. The current and wave velocities will contribute to drag force, while the acceleration contributes to inertia forces. By using equations *Equation* 2-12 to *Equation* 2-16 in chapter 2.3.2.2, we get following results from section 2.4 in appendix B:

	Max velocity	Max acceleration
Roll induced tangential motions	1.672 m/s	0.519 m/s^2

Table 3-8: Roll induced tangential motions.

3.1.3.2 Heave motion

Heave motions, forces the water to the sides of the vessel and into the I-tubes. Water level inside the turret is the same as outside and will therefore try to oscillate in the same frequency as the draft outside the vessel. Because of this, we can look at the I-tube as a submerged part of a circular moonpool. The maximum velocity and acceleration for heave motion on the Skarv FPSO is measured and calculated by Samsung (Samsung Heavy Ind.,LTD, 2007). For the turret area the result is:

	Max velocity	Max acceleration
Heave induced motions	3.340 m/s	1.837 m/$s^{2.}$

Table 3-9: Heave induced motions (Samsung Heavy Ind.,LTD, 2007).

3.1.4 Resulting forces

The resulting forces, induced by ocean and vessel motion, will impact the deployment system. Under this sub chapter, the total horizontal and vertical forces acting on the system is presented. These parameters are the most important ones when the analysis is presented in the capacity review.

3.1.4.1 Horizontal loads

Horizontal loads are induced by the relative velocity and/or acceleration between the object and the water particle. By implementing Morison's equation *Equation* 2-17 from chapter 2.3.3, we can retrieve the sectional maximum horizontal load that has an impact on the exposed area (4 m)

Horizontal Load	
Sectional Horizontal load	2.69 kN/m

Table 3-10: Sectional horizontal load.

3.1.4.2 Vertical loads

By following the equations in chapter 2.3.4.2, the maximum vertical force acting upwards onto the system is calculated in appendix B, section 3.2. The vertical load is to be seen as a point load acting upwards. This is the resulting load of the water plug velocity minus the self-weight of the structure.

Vertical Load	
Interaction between water plug and body	53.23 kN
Downward force due to submerged weight	13.54 kN
Total upwards vertical point load	*39.69 kN*

Table 3-11: vertical loads.

3.2 Capacity Analysis

As stated in the introduction, the goal of this thesis is to come up with a design that suits the given requirements. Due to necessary limitations on level of detailed calculations, the capacity review will look at the structure and its members through Focus 3D software. This means that specific capacity analysis of welds and joints are not part of this thesis.

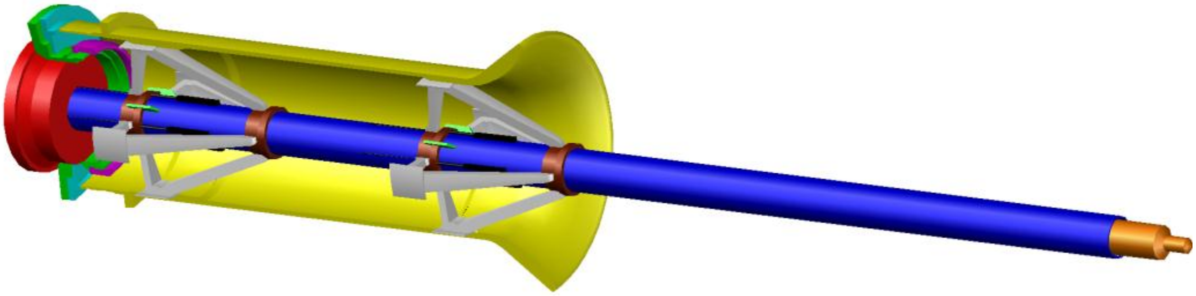

Figure 3-2: Intended design used for the capacity review.

This thesis has so far presented all the needed parameters. The main criteria related to the structural side of the deployment system are as listed in chapter 2.2.1:

1. Maximum horizontal deflection of the sonar head below 10 mm in a 100 year storm
2. The structure components should be robust enough to withstand the loads of a 100 year storm

The first capacity review is executed with all specification described earlier in this report. In Appendix C, the output file from Focus Construction software is providing a full capacity analysis of the deployment system components. From above mentioned criteria the following results are in focus:

Criteria	Result	Comment
10 mm deflection	11.7 mm	exceeds
Maximum utility factor < 1	5.35	exceeds

Table 3-12: result from first capacity analysis (Appendix C).

From the appendix C, we can see that segment 13 fails with a utilisation factor of 5.35. The upper and lower arms from chapter 2.2.4.2 require enlargement. For the next capacity analysis, the arms are made out of 60x20mm steel instead of 40x10. The results from the analysis executed in Appendix D, is shown in Table 3-13 below:

Criteria	Result	Comment
10 mm deflection	9.7 mm	clear
Maximum utility factor < 1	0.61	clear

Table 3-13: result from second capacity analysis

As we can see by enlarging the arms, both criteria are met. The deflection of 9.7 mm and maximum utility factor of 0.61 are satisfying results.

3.2.1 Hydraulic components

By using the equations from 2.3.4.4, Appendix - ch 4 shows that each of the hydraulic components needs to deliver a minimum force of 20 kN. According to the Enerpac website (Enerpac, 2011) the maximum capacity of the smallest hydraulic cylinder can provide a force up to 30 kN. With a safety factor of 1.5 (Appendix B) in mind, the chosen hydraulic components is sufficient for the task.

3.2.2 Capacity discussion of the Lynx connection system

The lynx connection system has been designed to handle a riser pull-in. In this case, the loads will be smaller than a riser pull-in. As a reference point, we can mention that the Bend stiffener connection system was tested with 100 tons overpull. This means that the capacity itself should not be a problem.

3.3 Chapter summary

Chapter 3 has presented the results from calculations and analysis done in the appendixes. The deployment tool needed some adjustments by enlarging the size of the centralizer arms. By doing so, the deployment tool met the given requirements for handling environmental impacts. The capacity analysis can be seen in appendix C and D.

4. Discussion

Through the discussion part, an evaluation of the concept described in previous chapters is presented. A discussion of pros and cons regarding the total solution will provide a better understanding of what could be done differently. The main discussion for the selected design is mainly done through the design process in chapter two. Other problems or solutions that are, or are not included in this thesis are highlighted and commented for changes, or further development.

4.1 Evaluation of Intended Design

In section 4.1, it is provided a general discussion of the intended design, which is evaluated in this thesis. It is categorized in the same way as the previous sections/chapters.

4.1.1 Monitoring device

There are many types of monitoring devices on the market. Testing and analysis of other transducer heads could be done to ensure that the Tritech device is the most suitable. On the other hand, when one have confirmed that something meets ones requirements, it may be better to involved further development instead of looking at other types of equipment. The discussion of why Tritech's sonar device is considered as the suitable can be found in more detail at chapter 2.1.3.

The Tritech sonar device itself has been seen as a great success. According to Lorraine Wallington from the RAMS investigation meeting (appendix A), Tritech is keen on following up the process and developing the technology further. This is a great opportunity for the contractor (in this case BP) to take part of the development of this type of equipment that is not very well known throughout the industry. This way, BP can form a position as an industry leader for this type of equipment.

4.1.2 Deployment tool

The deployment tool is designed in a similar way as the Foinaven field trial version. It does not have the same arrangement of the centralizer arms and the deployment tool itself is smaller than the tool used at Foinaven FPSO. This a lesson learned from the 2007 and the more recent failure investigation from appendix A is considered in the design of a new tool. A more detailed discussion of chosen design can be found in chapter 2.2.3 and 2.2.4.

The top-hat is new in this design and allows the deployment tool to rest on top of the BSCS. The top-hat design in this thesis will close the passage of the water. Even though the analysis shows that it will withstand its position, a redesign would be preferable. This redesign is discussed in further detail in section 4.2.2

As long as the deployment system has an empty I-tube slot to be mounted in, this solution is regarded as a preferable solution due to expenses and online feedback to control center.

4.1.3 Calculations and analysis

The calculations and analysis for this thesis are done in a simplified manner. If the deployment tool was to be fabricated, an engineering company would be hired to perform detailed calculations and analysis.

The capacity analysis performed in Focus Construction Software should also be double checked, or performed with more complex computer software than was used for this thesis.

4.2 Further Development

As we have been presented throughout this thesis, the concept has a fair opportunity of being fully developed and fabricated for use at Skarv FPSO. If this is going to occur, the concept still have many areas of investigation and engineering work before a final product could be fabricated. Areas of further investigation are discussed in this section.

4.2.1 Hydraulic & electrical system

One of the areas of concern regarding the functionality is the supply of hydraulic fluid. The proposed design includes 4 hydraulic components per centralizer unit, which each of them are connected to the lock ring. If one of these components fail, we can expect an unsymmetrical force distribution to the lock ring. In that case, all of the expanding devices could fail to deliver enough pressure onto the inner I-tube. This would result in a loose device and could in the worst case damage the BSCS. As an alternative, each of the hydraulic components could be connected to its own expander. If this was the case, a failure would only affect one arm and by doing so, the hydraulic fluid supply would need one fluid supply tube each. This would on the other hand create a more complex hydraulic system. Analysis regarding pressure loss and what type of connections, valves and hydraulic fluid that should be used is also considered as a part of further analysis.

The electrical signal system should be delivered in cooperation with Tritech international. For further development, an analysis should be performed on the signal strength and cable properties

4.2.2 Top-hat

The top-hat is intended to be able to support the entire deployment systems' weight, by resting on the top end of the lower I-tube. Considering the vessels vertical movement, water will move up and down the open I-tubes. This may indicate a problem if the top-hat is "sealing" the entrance of the moving waterplug. A suggested re-design would be to make holes in the top-hat for allowing the water to pass by.

Figure 4-1: The top-hat as intended to the left while redesigned with holes in the middle and to the right.

4.2.3 Bumper

Under installation, the deployment tool and delicate Tritech sonar equipment is expected to bounce into the inner wall of the I-tube. To prevent any unwanted incidents where parts of the system could be damaged, a bumper could be mounted to the lower part of the main pipe.

4.2.4 Vortex induced vibration preventer

Helical strakes could be added to the 10 ¾“ casing pipe. By doing so, The strakes would combat the system fatigue that would otherwise be caused by Vortex induced Vibrations (VIV). Since the casing pipe is subjected to perpendicular water flow, the current design could be sensitive to the VIV without the addition of the strakes. Analysis should be performed to conclude if this is necessary to include.

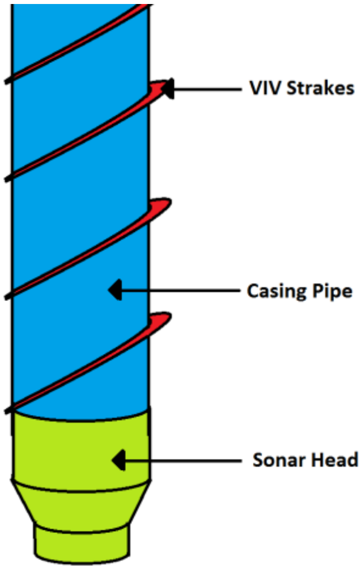

Figure 4-2: Strakes added on the main casing pipe to combat VIV.

4.2.5 Camlock

For additional fail-safe, a Camlock could be used to keep the centralizers/hydraulics in place. Camlock is a hydraulically actuated rotary motion unit. Hydraulics extends the camlock via small hydraulic rotary actuator. Once extended, the ring joint cannot retract centralizers, even if hydraulic pressure is lost, until camlock is removed. For further development, It could be suggested to use two Camlocks, 180^{o} apart. High reliability is require to ensure they will retract when required to ensure the tool does not get stuck. An illustration of the camlocks is provided in Figure 4-3.

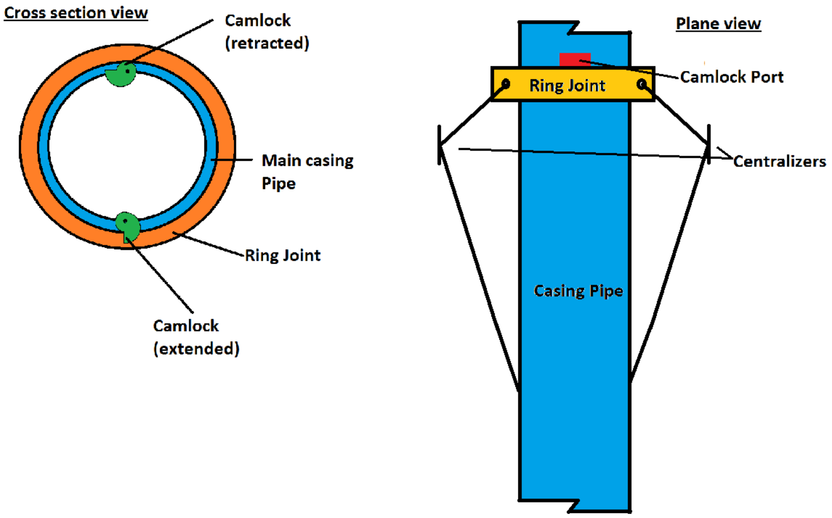

Figure 4-3: Illustration of a possible Camlock solution.

4.2.6 Hydraulic line routing

Hydraulic line routing could be implemented to ensure that the hydraulic & electrical umbilicals to the surface do not have the chance to pass over any 90° bends or sharp surfaces, therefore preventing premature aging/failure. Polymer radius controllers could be used for this purpose.

5. Conclusion

This master thesis has evaluated and investigated a possible design for an online riser monitoring system, designed to fit the specification at the Skarv FPSO. The objective was to develop a design suited for the vessel as well as the environments. With its limitations, the main purpose of this thesis was to create an idea, suggest a design and with basic analysis, evaluate if it is worth further development or fabrication.

This report is an evaluation of the transducer head and its deployment system. The transducer head is the actual monitoring device, and the deployment system needs to deploy the transducer head in preferred position. Tritechs transducer head was chosen due to earlier successful field trials and the willingness of further development cooperation from Tritech International.

To be able deploy the transducer head into preferred position, the deployment system needs to be fastened inside one of the unoccupied I-tubes. Different fastening methods have been discussed before a hydraulic expandable mounting system was evaluated as the best solution. The solution includes two sets of four expandable devices that keep the deployment system in position with help of friction force generated by the hydraulic components. To be able to conclude if the design meets its requirements or not, calculations regarding the environmental impacts were analyzed.

Due to an underestimation of the expanding arms, an improvement was needed to makes sure the system met the given requirements. After the improvement discussed in chapter 4.2, the arms were increased and therefore met the requirement regarding maximum deflection. The top-hat is, as evaluated in the discussion part, to be seen as one of the most important components to investigate for further development.

This report has shown that a monitoring system, deployed inside one of the unoccupied I-tubes, can withstand a 100 year storm and maintain full functionality. With further development and investigation it should be possible to fabricate the online monitoring system for use at Skarv FPSO.

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I. List of Figures

II. List of Tables

III. Appendix A – Meeting reviews

Design review meeting

3rd of May at 13.00 to 14.20

Meeting called by: Sveinung Rasmussen

Attendees: Jan Strøm, Subsea Engineer, BP – Eivind Grude, Control Engineer, BP

Sonar Equipment	- By experience is Kongsberg Simrad a better alternative than Tritech when it comes to sonar equipment. Kongsberg sonar MS1000 or MS 1100 should be evaluated but are rotational acoustic sensors compared to a multibeam sensor from Tritech - Sensor used on the Valhall field to recognize gas bubbles are extreme sensitive. Good idea to check that out - Background noise affecting the result can be discussed if necessary. - Check out sensors from Reson
Control unit	- Space inside the control unit for the surveillance system - HPU and valve control - Maintenance
Hydraulic verses mechanical	- Discussed and concluded that hydraulic is safer and easier when it comes to installation and retrieval. - Hydraulics are already used on several subsea structures with an intended lifetime of over 25 years
Top-hat and structural design. Water movement in lower turret	- Water movement will cause large forces to the structure. The more water that is allowed to pass by. The more water that is allowed to pass by. The lesser forces - Talk to hull design engineers and discuss the movement of water in the turret. This will help realize what's happening to the structure. - Find out Movement of the vessel compared to the turret position and

Alarm sensitivity	- Sensitivity needed by the alarm compared to the sensitivity on movement of risers
Supply of hydraulic fluid and electrical signals to the deployment system	- There should be a double hydraulic line to each of the centralizer sections. i.e. 4 hydraulic cables and 2 electronic cables plus a signal cable. Fluid could be Tellus 32 and HPU should be located for easy connection to the already existing hydraulic pressure system. If not, it should be possible with manual pumping to extend the centralizers.
Deflection sensitivity	- Deflection criteria of 10 mm should be considered out form the movement of the risers. Under a 100 year storm, there should be an acceptance criterion that allows the system to give an acceptable alarm. If the system comes back in to low area after special conditions, then it should be ok
Centralizers	- Centralizers could be steel against steel since pressure is applied. If not a hard plastic material could be used like in other centralizers. - The intended design should work good for the lower part but not for the top-hat if the water has huge movements at this area

Additional Information:

- Another meeting should be held with other Tom Brown to discuss these matters.

Design review meeting

9th of May at 14.00 to 15.30

Meeting called by: Sveinung Rasmussen

Attendees: Thomas Brown, Subsea Engineer, BP

Sensitivity and alarms	- If an alarm is released, all of the risers should have larger movement due to conditions. If one of the risers are moving more relative to the other, we might have a loose bend stiffener - It is also possible to have a reference point that is fixed to the turret. This way we can eliminate larger movement due to deflection in the sonar pipe - 4 meters depth considered as the best depth for catching all the risers due to spreading after leaving the bend stiffener. This should also be the case on Skarv
Maintenance	- For the hydraulic to stay intact, hydraulic pressure should be checked with by predefined routines
Installation and Safety	- It should be mentioned how the equipment is delivered, it will not fit a 20 foot container. For further development the pipe structure should be able to be divided in two pieces - BP's code for manual lifting is applicable. - Hydraulic fluid should be Tellus t32 shell for safe handling
Top-hat and structural design. Water movement in lower turret	- Good with holes in the Top-hat to allow flow through the equipment - Flow movement increasing by 1.5 inside the tube
Alarm sensitivity	- The risers are measured on a normal day to move up to 1.5 cm at 4 meters. 10 mm is therefore still an important requirement

Foinaven RAMS investigation meeting

23th of May at 09:30 to 12.00

Location: Subsea 7, Forus, Stavanger

Meeting called by: Lorraine Wallington, BP Aberdeen

Attendees:	Thomas Brown, Subsea Engineer, BP Norway
	Graham T. Smith, BP
	Stephan Garnham, BP
	Sveinung Rasmussen

Background	- Thomas brown was involved in the field trial at Foinaven FPSO. The trial was done in one week at good conditions. SP1 was the manufactory of the deployment system and took part of the installation - The permanent structure failed due to corrosion, hydraulic pressure and loss of electronic signals. - The report from Thomas Brown was passed on to a new team working on the permanent deployment tool. Few improvements were done and no maintenance routines were planned
Failure	- The top part of the main pipe was exposed to heavy rusting. This resulted in sharp edges which cut the hydraulic fluid supply and electronic signals from the sonar head
Maintenance	- Maintenance should be scheduled every 6 month to ensure the equipment is fully intact. - Removal of sea growth and inspection of corrosion
CP	- It is not clarified if there was any corrosion protection at all on the deployment system. Anodes should be a part of the system
Possible solutions	- A new design that could rest the full weight on something instead of only friction through the pads.

IV. Appendix B – Calculations

1. Input data:

General Input		***Reference***
Water density	$\rho := 1026 \frac{kg}{m^3}$	
Water viscosity	$\nu := 1.56 \cdot 10^{-6} \frac{m^2}{s}$	
Wave Input		
Water depth	$d_w := 370m$	(Aker Kvearner, 2008)
Extreme wave length	$W_l := 470m$	
10 year significant wave height	$H_{s10} := 13.7m$	(Aker Kvearner, 2008)
100 year significant wave height	$H_{s100} := 16.3m$	
10 year maximum wave height	$H_{max10} := 1.8 \cdot H_{s10}$	DNV- RP-R103
100 year maximum wave height	$H_{max100} := 1.8 \cdot H_{s100}$	DNV- RP-R103
10 year peak wave period	$T_{p10} := 16.1s$	(Aker Kvearner, 2008)
100 year peak wave period	$T_{p100} := 17.4s$	
10 year wave amplitude	$\zeta_{10} := \frac{H_{max10}}{2} = 12.33\,m$	
100 Wave amplitude	$\zeta_{100} := \frac{H_{max100}}{2} = 14.67\,m$	

General structure parameters

Exposed length of Structure	$L_o := 4\text{m}$	
Water depth of structure	$z := 25\text{m}$	
Diameter of Structure	$D_d := 0.273\text{m}$	
Cross section area	$\text{Area}_{cross} := \pi \cdot \left(\frac{D_d}{2}\right)^2 = 0.059\,\text{m}^2$	
Friction coefficient	$\mu := 0.6$	

Geometric Values of Vessel (Intermediate loaded)

Length between vessel perpendicular	$L_{vessel} := 277.6\text{m}$	
Depth moulded, D	$D_{moulded} := 29.0\text{m}$	
Breath moulded, B	$B := 50.6\text{m}$	
Center of turret from AP	$L_{cap} := 182\text{m}$	
Radius of gyration in roll	$R_{gyr} := 20.75\text{m}$	
Vertical center of gravity above keel; KG	$KG := 17.92\text{m}$	ref report om motions
GM	$GM := 4.2\text{m}$	
Maximum roll angle	$\theta_{roll} := 13\text{deg}$	

1.1 Water characteristics

1.1.1 - 10 year return period

ref gudmestad

Angular frequency: $\omega_{10} := \dfrac{2\pi}{T_{p10}}$ $\omega_{10} = 0.39\,\dfrac{1}{s}$

Wave length: $\lambda_{10} := \dfrac{g}{2\pi}\cdot T_{p10}^{\;2}$ $\lambda_{10} = 404.569\,m$

Deep water check (D/L>0.5) $\dfrac{d_w}{\lambda_{10}} = 0.915$ ok!

k-factor, given deep water $k_{10} := \dfrac{\omega_{10}^{\;2}}{g} = 0.016\,\dfrac{1}{m}$

1.1.2 - 100 year return period

Angular frequency: $\omega_{100} := \dfrac{2\pi}{T_{p100}}$ $\omega_{100} = 0.361\,\dfrac{1}{s}$

Wave length: $\lambda_{100} := \dfrac{g}{2\pi}\cdot T_{p100}^{\;2}$ $\lambda_{100} = 472.541\,m$

Deep water check (D/L>0.5) $\dfrac{d_w}{\lambda_{100}} = 0.783$ ok!

k-factor, given deep water $k_{100} := \dfrac{\omega_{100}^{\;2}}{g} = 0.013\,\dfrac{1}{m}$

2. Calculation of Loads

2.1 Wave Acceleration and Velocity

2.1.1 - 10 year return period

Water Particle velocity: $V_{w10}(\theta) := \left(\frac{\zeta_{10} \cdot k_{10} \cdot g}{\omega_{10}}\right) \cdot e^{\left(-z \cdot k_{10}\right)} \cdot \sin(\theta)$

Phase angle(max for velocity): $V_{w10}\left(\frac{\pi}{2}\right) = 3.264 \frac{m}{s}$

Water particle acceleration: $A_{w10}(\theta) := \left(\zeta_{10} \cdot k_{10} \cdot g\right) \cdot e^{\left(-z \cdot k_{10}\right)} \cdot \cos(\theta)$

Phase angle (max for acceleration) $A_{w10}(0) = 1.274 \frac{m}{s^2}$

2.1.2 - 100 year return period

Water Particle velocity: $V_{w100}(\theta) := \left(\frac{\zeta_{100} \cdot k_{100} \cdot g}{\omega_{100}}\right) \cdot e^{\left(-z \cdot k_{100}\right)} \cdot \sin(\theta)$

Phase angle(max for velocity): $V_{w100}\left(\frac{\pi}{2}\right) = 3.799 \frac{m}{s}$

Water particle acceleration: $A_{w100}(\theta) := \left(\zeta_{100} \cdot k_{100} \cdot g\right) \cdot e^{\left(-z \cdot k_{100}\right)} \cdot \cos(\theta)$

Phase angle (max for acceleration) $A_{w100}(0) = 1.372 \frac{m}{s^2}$

2.2 Current Velocity

Current velocity 10 year return period: $V_{c10} := 0.65\frac{m}{s}$

Current velocity 100 year return period: $V_{c100} := 0.74\frac{m}{s}$

2.3 Total Horizontal Waves and currents total velocity and accelerations

2.3.1 - 10 year wave and 100 year current return period

$V_{tot10}(\theta) := V_{w10}(\theta) + V_{c100}$ $V_{tot10}\left(\frac{\pi}{2}\right) = 4.004\frac{m}{s}$

$A_{tot10}(\theta) := A_{w10}(\theta)$ $A_{tot10}(0) = 1.274\frac{m}{s^2}$

2.3.1 - 100 year wave and 10 year current return period

$V_{tot100}(\theta) := V_{w100}(\theta) + V_{c10}$ $V_{tot100}\left(\frac{\pi}{2}\right) = 4.449\frac{m}{s}$

$A_{tot100}(\theta) := A_{w100}(\theta)$ $A_{tot100}(0) = 1.372\frac{m}{s^2}$

2.4 Total Horizontal roll induced movement velocity and acceleration

2.4.1 Acceleration induced by roll

Following is taken from DNV ship rules pt.3 ch.1 sec. 4

DNV active damping coefficient $K_a := 0.8$

Converting units to use in DNV $k_r := 1\frac{s}{m}\cdot R_{gyr}$ $GM_{non} := 1\cdot\frac{1}{m}\cdot GM$

Generally roll period is given as: $T_R := \frac{2\cdot k_r}{\sqrt{GM_{non}}} = 20.25\cdot s$

Max roll angle is generally written as $\Theta := \frac{50rad\cdot\frac{m}{s}\cdot\left(1.25s - 0.025\cdot T_R\right)K_a}{B + 75m} = 13.571\cdot deg$

Distance form center of object mass to the axis of rotation: $R_R := \frac{L_o}{2} + R_{gyr} = 22.75\,m$

The tangential roll acceleration (gravity component not included) is generally given by: $A_r := \Theta\cdot\left(\frac{2\cdot\pi}{T_R}\right)^2\cdot R_R = 0.519\frac{m}{s^2}$

2.4.2 To find the maximum velocity. We need to find the angular frequency

Angular roll frequecy motions can be described as: $\omega_{roll} := \frac{2\cdot\pi}{T_R} = 0.31\cdot\frac{rad}{s}$

The maximum Velocity will then be $Vmax_{roll} := \frac{A_r}{\omega_{roll}} = 1.672\frac{m}{s}$

From roll indeced movment, maximum acceleration and velocity will not act at the same time. Below, forces from inertia (acceleration) and from Drag (velocity) will be calculated to see which one to use in a conservative prespective.

Provosoric Coefficients $C_A := 0.8 \quad C_D := 0.8$

Inertia force due to roll induced acceleration: $Froll_{inertia} := \rho \cdot C_A \cdot Area_{cross} \cdot A_r$

Inertia force due to roll induced velocity: $Froll_{velocity} := \frac{1}{2} \rho \cdot C_D \cdot D_d \cdot Vmax_{roll}^{2}$

2.4.3 Dominating load case from roll induces movement

$$F_{roll} := \text{if}\left(Froll_{velocity} > Froll_{inertia}, Froll_{velocity}, Froll_{inertia}\right) = 0.313 \cdot \frac{kN}{m}$$

2.5 Coefficients

$$R_e := \frac{\left(Vmax_{roll} + V_{tot100}\left(\frac{\pi}{2}\right)\right) \cdot D_d}{\nu} = 1.071 \times 10^{6}$$

$\frac{L_o}{D_d} = 14.652$ ref DNV-rp-h103 appendix B

1.2.1 - Drag Coefficient

$C_{DS} := 1.0$

$K_D := \frac{(0.9 - 0.82)}{10} \cdot \left(\frac{L_o}{D_d} - 10\right) + 0.82 = 0.857$ interpolerer

$C_D := C_{DS} \cdot K_D$

$C_D = 0.857$

1.2.1 - Added Mass

$C_A := 0.98$

$C_L := 0.7$

2.6 Maximum load for Moving structure in waves and current

2.6.1 - 10 year wave and 100 year current return period

$$Fn_{10w100c}(\theta) := \rho \cdot (1 + C_A) \cdot Area_{cross} \cdot A_{tot10}(\theta) + \frac{1}{2} \cdot \rho \cdot C_D \cdot D_d \cdot V_{tot10}(\theta)^2$$

$\theta_{m10} := 0$ $\theta_{m10} := \text{Maximize}(Fn_{10w100c}, \theta_{m10})$ $\theta_{m10} = 1.523$

$$F_{max10} := Fn_{10w100c}(\theta_{m10}) = 1.928 \cdot \frac{kN}{m}$$

$$Drag := 0.5 \cdot \rho \cdot C_D \cdot D_d \cdot V_{tot10}(\theta_{m10})^2 = 1.921 \times 10^3 \cdot \frac{N}{m}$$

$$Inertia := \rho \cdot (1 + C_A) \cdot Area_{cross} \cdot A_{tot10}(\theta_{m10}) = 7.31 \cdot \frac{N}{m}$$

$$Drag + Inertia = 1.928 \times 10^3 \cdot \frac{N}{m}$$

2.6.2 - 100 year wave and 10 year current return period

$$Fn_{100w10c}(\theta) := \rho \cdot (1 + C_A) \cdot Area_{cross} \cdot A_{tot100}(\theta) + \frac{1}{2} \cdot \rho \cdot C_D \cdot D_d \cdot V_{tot100}(\theta)^2$$

$\theta_{m100} := 0.1$ $\quad$ $\theta_{m100} := \mathrm{Maximize}(Fn_{100w10c}, \theta_{m100})$ $\quad$ $\theta_{m100} = 1.531$

$$F_{max100} := Fn_{100w10c}(\theta_{m100}) = 2.38 \cdot \frac{kN}{m}$$

$$Drag100 := 0.5 \cdot \rho \cdot C_D \cdot D_d \cdot V_{tot100}(\theta_{m100})^2 = 2.373 \cdot \frac{kN}{m}$$

$$Inertia100 := \rho \cdot (1 + C_A) \cdot Area_{cross} \cdot A_{tot100}(\theta_{m100}) = 6.557 \times 10^{-3} \cdot \frac{kN}{m}$$

$$Drag100 + Inertia100 = 2.38 \cdot \frac{kN}{m}$$

F_{dom} := if($F_{max100} < F_{max10}$, "10 year waves and 100 year current" , "100 year waves and 10 year current"

Dominating combination is $\quad F_{dom}$ = "100 year waves and 10 year current"

$$F_{wNc} := \mathrm{if}(F_{max100} < F_{max10}, F_{max10}, F_{max100}) = 2.38 \cdot \frac{kN}{m}$$

2.7 Maximum Horizontal load

In the most extreme condition, where the FPSO has an roll direction prpendicular to the waves and current, maximum horizontal load will be a combination of max roll induced load and loads from waves and currents

$$Fmax_{Horizontal} := F_{wNc} + F_{roll} = 2.693 \cdot \frac{kN}{m}$$

3. Vertical Loads on strucure

3.1 Input details

Friction coefficient $\mu = 0.6$

Steel density $\rho_{steel} := 7850 \frac{kg}{m^3}$

Safety factor $SF_f := 1$

Total structure weight $W_t := 953kg$ ref: Chapter 3

Total structure volume $V_{ts} := \frac{W_t}{\rho_{steel}} = 0.121 \cdot m^3$

Bouyancy $W_b := \rho \cdot V_{ts} = 124.558\, kg$

Submerged structure weight $W_{tsub} := W_t - W_b = 828.442\, kg$

Contact area pr clamp $A_c := 150mm \cdot 100mm = 0.015\, m^2$

Number of clamps $N_c := 8$

contact area between clamps and I-tube $A_{ci} := A_c \cdot N_c = 0.12\, m^2$

Force to hold the structure $F_{gravity} := \frac{W_{tsub} \cdot g}{\mu} \cdot SF_f = 13.54 \cdot kN$

3.2 Heave motions

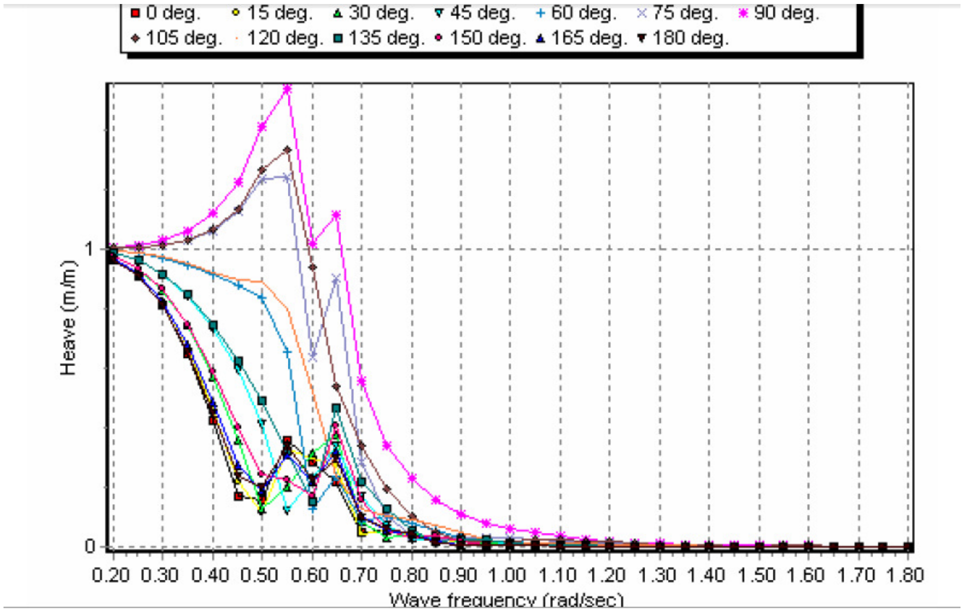
Heave period $T_h := 16\mathrm{s}$

Diameter I-tube $D_{itube} := 1.042\mathrm{m}$

Cross section I-tube $A_{itube} := \pi \cdot \left(\frac{D_{itube}}{2}\right)^2 = 0.853\,\mathrm{m}^2$

3.2.1 RAO heave motions

(Samsung Heavy Ind.,LTD, 2007)



Resonans frequency $\omega_h := 0.55\,\frac{\mathrm{rad}}{\mathrm{s}}$

Resonans Period $T_{mh} := \frac{2 \cdot \pi}{\omega_h} = 11.424\,\mathrm{s}$

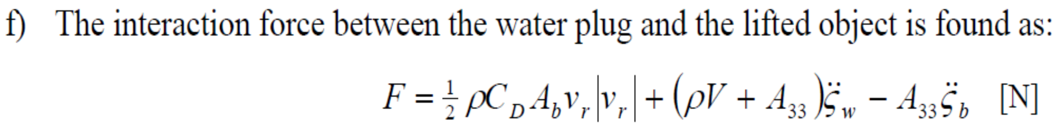
3.2.2 Max vertival heave acceleration at turret position

100 year Hs and Tp at turret position

100 year vertical acceleration taken from Samsung report at turret position $a_{vmax} := 1.837 \frac{m}{s^2}$ (Samsung Heavy Ind.,LTD, 2007)

Max velocity is then: $V_{maxheave} := \frac{a_{vmax}}{\omega_h} = 3.34 \frac{m}{s}$ (Gudmestad, 2012)

3.5 DNV calculations Moonpool
As an simplification, we can look at the I tube as a moonpool and then follow the DNV-RP-H103 section 3.5. Since the we look at the problem as a piston problem that covers more then 80 % of the "moonpool area" we need to look at the Comprehensive calculation method at 3.5.7



f) The interaction force between the water plug and the lifted object is found as:

$$F = \tfrac{1}{2}\rho C_D A_b v_r |v_r| + (\rho V + A_{33})\ddot{\varsigma}_w - A_{33}\ddot{\varsigma}_b \quad [\mathrm{N}]$$

Solid projected area of object Top-hat region $A_b := A_{itube} \cdot 0.8 = 0.682\,m^2$

Drag coefficient in comprehensive method $C_{Dm} := 1 - 0.5 \cdot \frac{A_b}{A_{itube}} = 0.6$

Relative velocity between body and waterplug -in this case. the max velocity induced by heave motions $V_r := V_{maxheave} = 3.34 \frac{m}{s}$

DNV 3.5.4.5 k for circular moonpool $\kappa := 0.48 \frac{1}{m^{1.5}}$

Added mass of body $A_{33} := \rho \cdot \kappa \cdot A_{itube} \cdot z \cdot \sqrt{z \cdot A_{itube}} = 4.848 \times 10^{4} \cdot kg$

Volume of body submerged body $V_{sb} := 2.25 m^{3}$

Vertical acceleration of waterplug is in this case taken to the same as the max heave acceleration times moonpool factor of 1.5 for conservative calculations $a_{wp} := a_{vmax} \cdot 1.5 = 2.756 \frac{m}{s^{2}}$

Vertical acceleration of body $a_{b} := a_{vmax} = 1.837 \frac{m}{s^{2}}$

Interaction force between waterplug and body according to DNV

$$F_{wp} := \frac{1}{2} \cdot \rho \cdot C_{Dm} \cdot A_{b} \cdot V_{r}^{2} + \left(\rho \cdot V_{sb} + A_{33}\right) \cdot a_{wp} - A_{33} \cdot a_{b} = 53.23 \cdot kN$$

Total upward force including submerged weight

$$F_{up} := F_{wp} - F_{gravity} = 39.689 \cdot kN$$

4. Calculation of needed hydraulic pressure

The Deployment system will need to stand against the vertical forecs from water pushing into the structure. The expandlble stabilizers are driven by hydraulic pressure.

Forces in Vertical direction $F_{up} = 39.689 \cdot kN$

Friction Coefficient, Steel / Tyflon $\mu_{st} := 0.2$

Number of pads $No_{pads} := 8$

Safety factor $S_f := 1.5$ (NS-EN 1990:2002/NA2008)

Minimum horizontal force pr pad $F_{pad} := \dfrac{F_{up} \cdot S_f}{\mu_{st} \cdot No_{pads}} = 37.209 \cdot kN$

4.1 Calculation of hydraulic component features

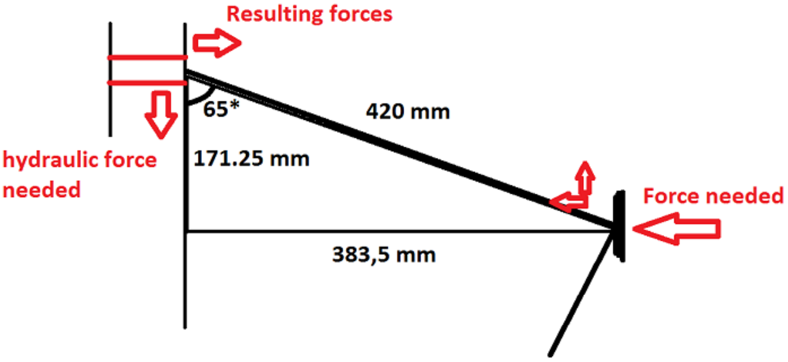
When the hydraulic components are pulling with enough force. the pas will experience a F.pad onto it.to find out how much force that actually is pulled we need to find out the vertical component at the lock ring

Angle when expanded $\beta := 65deg$

Force in upper arm component $F_{arm} := \dfrac{F_{pad}}{\sin(\beta)} = 41.055 \cdot kN$

Force component in vertical direction to be produced by the Hydraulic components $F_{vert} := \cos(\beta) \cdot F_{arm} = 17.351 \cdot kN$

A typical hydraulic sylinder with capacity of 20kN + should be chosen



4.2 Stroke needed to fully expand the pads

Length of lower arm	$l_{larm} := 870\text{mm}$
Length of upper arm	$l_{uarm} := 420\text{mm}$
Angle when fully expanded	$\beta_{exp} := 65\text{deg}$
Angle when in deployment mode	$\beta_{dep} := 30\text{deg}$
Distance due to angluare movent of lower arm	$b_1 := l_{larm} \cdot \left(1 - \cos\left(\beta_{exp} - \beta_{dep}\right)\right) = 0.157 \cdot \text{m}$
Distance du to angular movement of upper arm	$a := l_{uarm} \cdot \sin\left(\beta_{dep}\right) = 0.21\,\text{m}$

V. Appendix C – Capacity Analysis

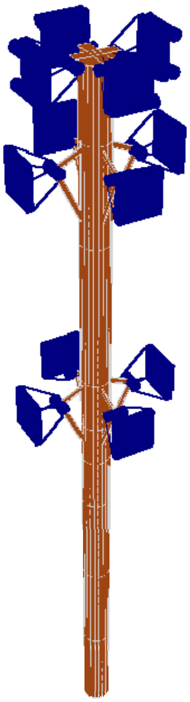
Capacity Analysis

Note - First trial

Beregning utført: 29.05.2012 16:08:40

Focus Konstruksjon 2012

1. KONSTRUKSJONSMODELL OG LASTER



1.1. KNUTEPUNKTSDATA

Nr.	X [mm]	Y [mm]	Z [mm]
1	10000	1000	0
2	10000	1000	1000
3	10000	1000	2000
4	10000	1000	3000
5	10000	1000	3569
6	10000	1000	4000
7	9617	1000	4350
8	10384	1000	4350
9	10000	1000	4521
10	10000	1000	5984
11	9617	1000	6765
12	10384	1000	6765
13	10000	1000	6936
14	10200	1000	7615
15	10000	1000	7615
16	9800	1000	7615
17	10000	617	4350

Nr.	X [mm]	Y [mm]	Z [mm]
18	10000	617	6765
19	10000	1384	4350
20	10000	1384	6765
21	10000	1200	7615
22	10000	800	7615

1.2. TVERRSNITTSDATA

Nr.	Navn	Parametre	
1	Firkantstål 100	A [mm^2]	10000
		Ix [mm^4]	1,2333e+007
		Iy [mm^4]	8,3333e+006
		Iz [mm^4]	8,3333e+006
		Total vekt [kN]	0,62
2	KF Rør 273.0x10.0	A [mm^2]	8262
		Ix [mm^4]	1,4308e+008
		Iy [mm^4]	7,1541e+007
		Iz [mm^4]	7,1541e+007
		Total vekt [kN]	4,85
3	Flatstål 40x10	A [mm^2]	400
		Ix [mm^4]	1,1233e+004
		Iy [mm^4]	5,3333e+004
		Iz [mm^4]	3,3333e+003
		Total vekt [kN]	0,32

1.3. MATERIALDATA

1 **Stål** Material: Stål

Fasthetsklasse: S355

Varmeutv.koeff.: 1,20e-005 °C^-1 Tyngdetetthet: 77,01 kN/m^3

E-modul: 2,1000e+005 N/mm^2 G-modul: 8,1000e+004 N/mm^2

Karakteristiske fasthetsparametre:

f_y = 355,00 N/mm^2 for godstykkelse <= 40,0 mm
f_y = 335,00 N/mm^2 for godstykkelse <= 80,0 mm
f_y = 335,00 N/mm^2 for godstykkelse > 80,0 mm

1.4. SEGMENTDATA

Seg Nr.	Kn.pkt 1	Kn.pkt 2	Tvsn 1	Tvsn 2	Material
1	1	2	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
2	2	3	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
3	3	4	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
4	4	5	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
5	5	6	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål

Seg Nr.	Kn.pkt 1	Kn.pkt 2	Tvsn 1	Tvsn 2	Material
6	6	9	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
7	9	10	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
8	10	13	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
9	13	15	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
10	9	7	Flatstål 40x10	Flatstål 40x10	Stål
11	9	8	Flatstål 40x10	Flatstål 40x10	Stål
12	7	5	Flatstål 40x10	Flatstål 40x10	Stål
13	5	8	Flatstål 40x10	Flatstål 40x10	Stål
14	13	11	Flatstål 40x10	Flatstål 40x10	Stål
15	13	12	Flatstål 40x10	Flatstål 40x10	Stål
16	11	10	Flatstål 40x10	Flatstål 40x10	Stål
17	10	12	Flatstål 40x10	Flatstål 40x10	Stål
18	20	13	Flatstål 40x10	Flatstål 40x10	Stål
19	13	18	Flatstål 40x10	Flatstål 40x10	Stål
20	18	10	Flatstål 40x10	Flatstål 40x10	Stål
21	20	10	Flatstål 40x10	Flatstål 40x10	Stål
22	19	9	Flatstål 40x10	Flatstål 40x10	Stål
23	17	9	Flatstål 40x10	Flatstål 40x10	Stål
24	17	5	Flatstål 40x10	Flatstål 40x10	Stål
25	19	5	Flatstål 40x10	Flatstål 40x10	Stål
26	14	15	Firkantstål 100	Firkantstål 100	Stål
27	22	15	Firkantstål 100	Firkantstål 100	Stål
28	16	15	Firkantstål 100	Firkantstål 100	Stål
29	21	15	Firkantstål 100	Firkantstål 100	Stål

1.4.1. SEGMENTDATA EN 1993

Seg. nr.	Gamma_M0	Gamma_M1	L_ky [mm]	L_kz [mm]	L_eff [mm]
1	1,05	1,05	1000	1000	1000
2	1,05	1,05	1000	1000	1000
3	1,05	1,05	1000	1000	1000
4	1,05	1,05	569	569	569
5	1,05	1,05	431	431	431
6	1,05	1,05	521	521	521
7	1,05	1,05	1463	1463	1463
8	1,05	1,05	952	952	952
9	1,05	1,05	679	679	679
10	1,05	1,05	420	420	420
11	1,05	1,05	420	420	420
12	1,05	1,05	870	870	870
13	1,05	1,05	870	870	870
14	1,05	1,05	420	420	420
15	1,05	1,05	420	420	420
16	1,05	1,05	870	870	870
17	1,05	1,05	870	870	870

Seg. nr.	Gamma_M0	Gamma_M1	L_ky [mm]	L_kz [mm]	L_eff [mm]
18	1,05	1,05	420	420	420
19	1,05	1,05	420	420	420
20	1,05	1,05	870	870	870
21	1,05	1,05	870	870	870
22	1,05	1,05	420	420	420
23	1,05	1,05	420	420	420
24	1,05	1,05	870	870	870
25	1,05	1,05	870	870	870
26	1,05	1,05	200	200	200
27	1,05	1,05	200	200	200
28	1,05	1,05	200	200	200
29	1,05	1,05	200	200	200

1.5. RANDBETINGELSER

Seg Nr.	X [mm]	Y [mm]	Z [mm]	Frih.gr. X	Y	Z	RotX	RotY	RotZ
11	10384	1000	4350	F	F	F	F		F
10	9617	1000	4350	F	F	F	F		F
22	10000	1384	4350	F	F	F	F	F	
23	10000	617	4350	F	F	F	F	F	
26	10200	1000	7615	F	F		F		F
28	9800	1000	7615	F	F		F		F
29	10000	1200	7615	F	F			F	F
27	10000	800	7615	F	F			F	F
14	9617	1000	6765	F	F	F	F		F
19	10000	617	6765	F	F	F	F	F	
15	10384	1000	6765	F	F	F	F		F
18	10000	1384	6765	F	F	F	F	F	

Forklaring til frihetsgrader: F = fastholdt, (blank) = fri
Tall betyr foreskreven forskyvning [mm]

1.7. LASTTILFELLER

1 Nyttelast

Lastvarighet: Korttidslast
Lasttype: Annen variabel
1 Fordelt last P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 3569 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 4000 mm
Retning = [1; 0; 0]
Virker på segment: 5

2 Fordelt last P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 3000 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 3569 mm
Retning = [1; 0; 0]
Virker på segment: 4

3 Fordelt last P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 2000 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 3000 mm
Retning = [1; 0; 0]
Virker på segment: 3

4 Fordelt last P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 1000 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 2000 mm
Retning = [1; 0; 0]
Virker på segment: 2

5 Fordelt last P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 0 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 1000 mm
Retning = [1; 0; 0]
Virker på segment: 1

6 Punktlast P = 39,70 kN
X = 10000 mm Y = 1000 mm Z = 0 mm
Retning = [0; 0; 1]
Virker på segment: 1

1.8. LASTKOMBINASJON

Beregning utført for lastkombinasjon
(1) LASTKOMBO 1

Grensetilstand: Brudd

1,20 * <Konstruksjonens tyngde>
1,50 * Nyttelast

2. STATISKE BEREGNINGER

2.1. KNUTEPUNKTSRESULTATER

2.1.1. Forskyvninger

Nr.	u [mm]	v [mm]	w [mm]	rotX [°]	rotY [°]	rotZ [°]
1	11,7	0,0	0,2	0,0	-0,2	0,0
2	8,2	0,0	0,2	0,0	-0,2	0,0
3	4,8	0,0	0,1	0,0	-0,2	0,0
4	1,9	0,0	0,1	0,0	-0,1	0,0
5	0,8	0,0	0,1	0,0	-0,1	0,0
6	0,3	0,0	0,1	0,0	-0,1	0,0
7	0,0	0,0	0,0	0,0	0,0	0,0
8	0,0	0,0	0,0	0,0	0,0	0,0

Nr.	u [mm]	v [mm]	w [mm]	rotX [°]	rotY [°]	rotZ [°]
9	0,0	0,0	0,1	0,0	0,0	0,0
10	-0,1	0,0	0,0	0,0	0,0	0,0
11	0,0	0,0	0,0	0,0	0,0	0,0
12	0,0	0,0	0,0	0,0	0,0	0,0
13	0,0	0,0	0,0	0,0	0,0	0,0
14	0,0	0,0	0,0	0,0	0,0	0,0
15	0,0	0,0	0,0	0,0	0,0	0,0
16	0,0	0,0	0,0	0,0	0,0	0,0
17	0,0	0,0	0,0	0,0	0,0	0,0
18	0,0	0,0	0,0	0,0	0,0	0,0
19	0,0	0,0	0,0	0,0	0,0	0,0
20	0,0	0,0	0,0	0,0	0,0	0,0
21	0,0	0,0	0,0	0,0	0,0	0,0
22	0,0	0,0	0,0	0,0	0,0	0,0

2.1.2. Residualkrefter

Nr.	Rx [kN]	Ry [kN]	Rz [kN]	Mx [kN·m]	My [kN·m]	Mz [kN·m]
1	0,00	0,00	0,00	0,00	0,00	0,00
2	0,00	0,00	0,00	0,00	0,00	0,00
3	0,00	0,00	0,00	0,00	0,00	0,00
4	0,00	0,00	0,00	0,00	0,00	0,00
5	0,00	0,00	0,00	0,00	0,00	0,00
6	0,00	0,00	0,00	0,00	0,00	0,00
7	-12,98	0,00	24,66	0,00	0,00	0,00
8	-9,79	0,00	-41,36	0,00	0,00	0,00
9	0,00	0,00	0,00	0,00	0,00	0,00
10	0,00	0,00	0,00	0,00	0,00	0,00
11	3,75	0,00	-7,95	0,00	0,00	0,00
12	6,36	0,00	-1,67	0,00	0,00	0,00
13	0,00	0,00	0,00	0,00	0,00	0,00
14	-1,37	0,00	0,00	0,00	0,00	0,00
15	0,00	0,00	0,00	0,00	0,00	0,00
16	-1,37	0,00	0,00	0,00	0,00	0,00
17	0,00	-1,60	-8,35	0,00	0,00	0,00
18	0,01	-1,30	-4,81	0,00	0,00	0,00
19	0,00	1,60	-8,35	0,00	0,00	0,00
20	0,01	1,30	-4,81	0,00	0,00	0,00
21	-0,19	0,00	0,00	0,00	-0,04	-0,02
22	-0,19	0,00	0,00	0,00	-0,04	0,02

2.2. OPPLEGGSKREFTER

Seg Nr.	X [mm]	Y [mm]	Z [mm]	Rx [kN]	Ry [kN]	Rz [kN]	Mx [kN·m]	My [kN·m]	Mz [kN·m]

Seg Nr.	X [mm]	Y [mm]	Z [mm]	Rx [kN]	Ry [kN]	Rz [kN]	Mx [kN·m]	My [kN·m]	Mz [kN·m]
11	10384	1000	4350	-9,79	0,00	-41,36	0,00	0,00	0,00
10	9617	1000	4350	-12,98	0,00	24,66	0,00	0,00	0,00
22	10000	1384	4350	0,00	1,60	-8,35	0,00	0,00	0,00
23	10000	617	4350	0,00	-1,60	-8,35	0,00	0,00	0,00
26	10200	1000	7615	-1,37	0,00	0,00	0,00	0,00	0,00
28	9800	1000	7615	-1,37	0,00	0,00	0,00	0,00	0,00
29	10000	1200	7615	-0,19	0,00	0,00	0,00	-0,04	-0,02
27	10000	800	7615	-0,19	0,00	0,00	0,00	-0,04	0,02
14	9617	1000	6765	3,76	0,00	-7,95	0,00	0,00	0,00
19	10000	617	6765	0,00	-1,30	-4,81	0,00	0,00	0,00
15	10384	1000	6765	6,36	0,00	-1,67	0,00	0,00	0,00
18	10000	1384	6765	0,00	1,30	-4,81	0,00	0,00	0,00
	Sum			-15,78	0,00	-52,61			

2.3. SEGMENTRESULTATER

Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
1	0	0,00	0,00	0,00	-59,51	0,00	0,20	11,7	0,0	0,2
	1000	1,97	0,00	0,00	-58,82	0,00	3,75	8,2	0,0	0,2
	1000	1,97	0,00	0,00	-58,82	0,00	3,75	8,2	0,0	0,2
2	0	1,97	0,00	0,00	-58,75	0,00	4,14	8,2	0,0	0,2
	1000	7,89	0,00	0,00	-58,06	0,00	7,69	4,8	0,0	0,1
	1000	7,89	0,00	0,00	-58,06	0,00	7,69	4,8	0,0	0,1
3	0	7,89	0,00	0,00	-57,98	0,00	8,09	4,8	0,0	0,1
	1000	17,75	0,00	0,00	-57,30	0,00	11,64	1,9	0,0	0,1
	1000	17,75	0,00	0,00	-57,30	0,00	11,64	1,9	0,0	0,1
4	0	17,75	0,00	0,00	-57,24	0,00	11,95	1,9	0,0	0,1
	569	25,13	0,00	0,00	-56,85	0,00	13,97	0,8	0,0	0,1
	569	25,13	0,00	0,00	-56,85	0,00	13,97	0,8	0,0	0,1
5	0	25,09	0,00	0,00	-31,78	0,00	-16,32	0,8	0,0	0,1
	0	25,09	0,00	0,00	-31,78	0,00	-16,32	0,8	0,0	0,1
	431	18,35	0,00	0,00	-31,51	0,00	-14,96	0,3	0,0	0,1
6	0	18,35	0,00	0,00	-31,46	0,00	-14,79	0,3	0,0	0,1
	0	18,35	0,00	0,00	-31,46	0,00	-14,79	0,3	0,0	0,1
	521	10,65	0,00	0,00	-31,10	0,00	-14,79	0,0	0,0	0,1
7	0	10,56	0,00	0,00	-22,46	0,00	-7,00	0,0	0,0	0,1
	0	10,56	0,00	0,00	-22,46	0,00	-7,00	0,0	0,0	0,1

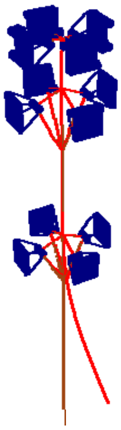
Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
	1463	0,32	0,00	0,00	-21,46	0,00	-7,00	-0,1	0,0	0,0
8	0	0,32	0,00	0,00	-7,48	0,00	-2,66	-0,1	0,0	0,0
	952	-2,22	0,00	0,00	-6,82	0,00	-2,66	0,0	0,0	0,0
	952	-2,22	0,00	0,00	-6,82	0,00	-2,66	0,0	0,0	0,0
9	0	-2,20	0,00	0,00	-1,23	0,00	3,12	0,0	0,0	0,0
	0	-2,20	0,00	0,00	-1,23	0,00	3,12	0,0	0,0	0,0
	679	-0,08	0,00	0,00	-0,77	0,00	3,12	0,0	0,0	0,0
10	0	-0,03	0,00	0,00	0,80	0,00	0,10	0,0	0,0	0,1
	0	-0,03	0,00	0,00	0,80	0,00	0,10	0,0	0,0	0,1
	420	0,02	0,00	0,00	0,79	0,00	0,11	0,0	0,0	0,0
11	0	0,06	0,00	0,00	9,47	0,00	-0,21	0,0	0,0	0,1
	0	0,06	0,00	0,00	9,47	0,00	-0,21	0,0	0,0	0,1
	420	-0,03	0,00	0,00	9,46	0,00	-0,20	0,0	0,0	0,0
12	0	-0,02	0,00	0,00	27,71	0,00	0,00	0,0	0,0	0,0
	261	-0,02	0,00	0,00	27,71	0,00	0,00	0,1	0,0	0,0
	870	-0,01	0,00	0,00	27,69	0,00	0,01	0,8	0,0	0,1
13	0	0,02	0,00	0,00	-41,62	0,00	0,01	0,8	0,0	0,1
	870	0,03	0,00	0,00	-41,59	0,00	0,02	0,0	0,0	0,0
	870	0,03	0,00	0,00	-41,59	0,00	0,02	0,0	0,0	0,0
14	0	0,02	0,00	0,00	0,16	0,00	-0,07	0,0	0,0	0,0
	0	0,02	0,00	0,00	0,16	0,00	-0,07	0,0	0,0	0,0
	420	-0,01	0,00	0,00	0,15	0,00	-0,06	0,0	0,0	0,0
15	0	0,00	0,00	0,00	6,45	0,00	0,00	0,0	0,0	0,0
	84	0,00	0,00	0,00	6,45	0,00	0,00	0,0	0,0	0,0
	420	0,00	0,00	0,00	6,45	0,00	0,01	0,0	0,0	0,0
16	0	0,01	0,00	0,00	-8,74	0,00	-0,02	0,0	0,0	0,0
	0	0,01	0,00	0,00	-8,74	0,00	-0,02	0,0	0,0	0,0
	870	0,00	0,00	0,00	-8,76	0,00	-0,01	-0,1	0,0	0,0
17	0	0,00	0,00	0,00	1,03	0,00	-0,01	-0,1	0,0	0,0
	783	0,00	0,00	0,00	1,06	0,00	0,00	0,0	0,0	0,0
	870	0,00	0,00	0,00	1,06	0,00	0,00	0,0	0,0	0,0

Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
18	0	0,00	0,00	0,00	3,30	0,00	0,03	0,0	0,0	0,0
	420	0,01	0,00	0,00	3,30	0,00	0,04	0,0	0,0	0,0
	420	0,01	0,00	0,00	3,30	0,00	0,04	0,0	0,0	0,0
19	0	0,01	0,00	0,00	3,30	0,00	-0,04	0,0	0,0	0,0
	0	0,01	0,00	0,00	3,30	0,00	-0,04	0,0	0,0	0,0
	420	0,00	0,00	0,00	3,30	0,00	-0,03	0,0	0,0	0,0
20	0	0,00	0,00	0,00	-3,84	0,00	-0,01	0,0	0,0	0,0
	696	0,00	0,00	0,00	-3,86	0,00	0,00	-0,1	0,0	0,0
	870	0,00	0,00	0,00	-3,87	0,00	0,00	-0,1	0,0	0,0
21	0	0,00	0,00	0,00	-3,84	0,00	-0,01	0,0	0,0	0,0
	696	0,00	0,00	0,00	-3,86	0,00	0,00	-0,1	0,0	0,0
	870	0,00	0,00	0,00	-3,87	0,00	0,00	-0,1	0,0	0,0
22	0	-0,01	0,00	0,00	5,13	0,00	0,04	0,0	0,0	0,0
	420	0,01	0,00	0,00	5,13	0,00	0,05	0,0	0,0	0,1
	420	0,01	0,00	0,00	5,13	0,00	0,05	0,0	0,0	0,1
23	0	-0,01	0,00	0,00	5,13	0,00	0,04	0,0	0,0	0,0
	420	0,01	0,00	0,00	5,13	0,00	0,05	0,0	0,0	0,1
	420	0,01	0,00	0,00	5,13	0,00	0,05	0,0	0,0	0,1
24	0	0,01	0,00	0,00	-6,94	0,00	-0,01	0,0	0,0	0,0
	783	0,00	0,00	0,00	-6,97	0,00	0,00	0,7	0,0	0,1
	870	0,00	0,00	0,00	-6,97	0,00	0,00	0,8	0,0	0,1
25	0	0,01	0,00	0,00	-6,94	0,00	-0,01	0,0	0,0	0,0
	783	0,00	0,00	0,00	-6,97	0,00	0,00	0,7	0,0	0,1
	870	0,00	0,00	0,00	-6,97	0,00	0,00	0,8	0,0	0,1
26	0	0,00	0,00	0,00	-1,37	0,00	0,03	0,0	0,0	0,0
	200	0,02	0,00	0,00	-1,37	0,00	0,15	0,0	0,0	0,0
	200	0,02	0,00	0,00	-1,37	0,00	0,15	0,0	0,0	0,0
27	0	0,00	0,04	-0,02	0,00	0,19	0,03	0,0	0,0	0,0
	200	0,02	0,04	0,02	0,00	0,19	0,15	0,0	0,0	0,0
	200	0,02	0,04	0,02	0,00	0,19	0,15	0,0	0,0	0,0
28	0	0,00	0,00	0,00	1,37	0,00	0,03	0,0	0,0	0,0
	200	0,02	0,00	0,00	1,37	0,00	0,15	0,0	0,0	0,0
	200	0,02	0,00	0,00	1,37	0,00	0,15	0,0	0,0	0,0

Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
29	0	0,00	-0,04	0,02	0,00	-0,19	0,03	0,0	0,0	0,0
	200	0,02	-0,04	-0,02	0,00	-0,19	0,15	0,0	0,0	0,0
	200	0,02	-0,04	-0,02	0,00	-0,19	0,15	0,0	0,0	0,0

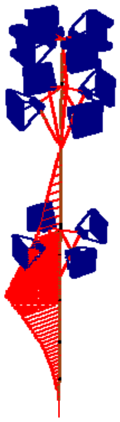
2.4. STATISKE RESULTATER GRAFISK

2.4.1. Forskyvning



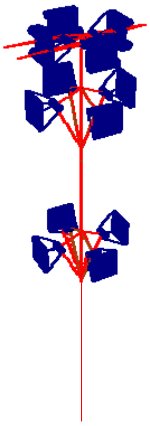
Største forskyvning: 11,7 mm

2.4.2. Moment om y-akse



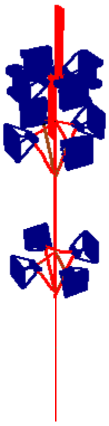
Største moment om y-akse: 25,13 kN·m

2.4.3 Moment om z-akse



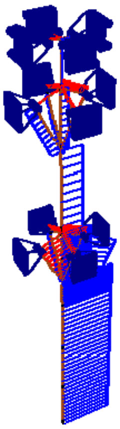
Største moment om z-akse: 0,02 kN·m

2.4.4 Torsjonsmoment



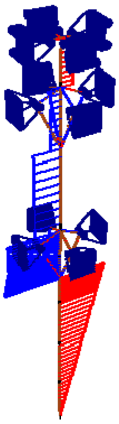
Største torsjonsmoment: 0,04 kN·m

2.4.5 Aksialkraft



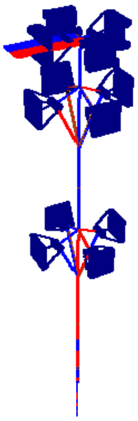
Største aksialkraft: -59,51 kN

2.4.6. Skjærkraft i z-retning



Største skjærkraft i z-retn.: 16,32 kN

2.4.7 Skjærkraft i y-retning



Største skjærkraft i y-retn.: 0,19 kN

3. KAPASITETSKONTROLL

3.1. UTNYTTELSESGRAD EN 1993

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
1	0	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	200	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	300	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	400	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	500	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	600	0,02	0,02	0,03	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	700	0,03	0,02	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	800	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	900	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1000	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
2	0	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	200	0,03	0,03	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	300	0,04	0,03	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	400	0,04	0,03	0,04	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	500	0,04	0,03	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	600	0,04	0,04	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	700	0,05	0,04	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	800	0,05	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	900	0,05	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1000	0,05	0,05	0,07	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
3	0	0,05	0,05	0,07	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,06	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	200	0,06	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	300	0,07	0,05	0,08	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	400	0,07	0,06	0,08	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	500	0,07	0,06	0,09	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	600	0,08	0,06	0,10	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	700	0,08	0,07	0,10	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	800	0,09	0,07	0,11	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	900	0,09	0,07	0,11	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1000	0,10	0,08	0,12	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
4	0	0,10	0,08	0,12	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	57	0,10	0,08	0,12	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	114	0,10	0,08	0,13	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	171	0,11	0,08	0,13	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	228	0,11	0,09	0,14	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	285	0,11	0,09	0,14	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	341	0,11	0,09	0,14	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	398	0,12	0,10	0,15	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	455	0,12	0,10	0,15	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	512	0,12	0,10	0,16	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	569	0,13	0,11	0,16	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
5	0	0,12	0,11	0,15	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	43	0,12	0,10	0,15	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	86	0,11	0,10	0,15	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	129	0,11	0,10	0,14	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	172	0,11	0,10	0,14	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	216	0,10	0,09	0,13	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	259	0,10	0,09	0,13	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	302	0,10	0,09	0,13	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	345	0,10	0,08	0,12	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	388	0,09	0,08	0,12	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	431	0,09	0,08	0,11	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
6	0	0,09	0,08	0,11	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	52	0,09	0,08	0,11	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	104	0,08	0,07	0,11	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	156	0,08	0,07	0,10	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	208	0,08	0,07	0,10	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	261	0,07	0,06	0,09	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	313	0,07	0,06	0,09	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	365	0,07	0,06	0,08	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	417	0,06	0,05	0,08	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	469	0,06	0,05	0,08	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	521	0,06	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
7	0	0,05	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	146	0,05	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	293	0,04	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	439	0,04	0,03	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	585	0,04	0,03	0,04	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	732	0,03	0,03	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	878	0,03	0,02	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1024	0,02	0,02	0,03	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1170	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1317	0,01	0,01	0,02	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1463	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
8	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	95	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	190	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	286	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	381	0,01	0,00	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	476	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	571	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	666	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	762	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	857	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	952	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
9	0	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	68	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	136	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	204	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	272	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	340	0,01	0,00	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	407	0,00	0,00	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	475	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	543	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	611	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	679	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
10	0	0,02	0,02	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,02	0,02	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,01	0,01	0,03	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,01	0,01	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,01	0,01	0,02	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	252	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,01	0,01	0,02	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,01	0,01	0,03	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
11	0	0,05	0,04	0,13	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,04	0,04	0,12	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,03	0,03	0,11	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,03	0,02	0,10	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,02	0,02	0,09	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,01	0,01	0,08	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,01	0,00	0,08	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,01	0,00	0,07	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,01	0,01	0,08	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,02	0,01	0,09	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,03	0,02	0,10	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
12	0	0,05	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	87	0,06	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	174	0,06	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	261	0,06	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	348	0,06	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	435	0,06	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	522	0,05	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	609	0,05	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	696	0,05	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	783	0,05	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	870	0,05	0,01	0,22	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
13	0	0,11	5,32	0,33	5,29	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,11	5,32	0,33	5,28	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,11	5,31	0,33	5,28	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,11	5,31	0,33	5,27	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,11	5,30	0,33	5,26	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,11	5,30	0,33	5,26	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,11	5,29	0,33	5,25	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,11	5,29	0,33	5,24	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,11	5,28	0,34	5,23	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,11	5,27	0,34	5,22	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,12	5,26	0,34	5,20	EN 1993-1-1 6.3.3 Ligning (6.62)
14	0	0,01	0,01	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,01	0,01	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	84	0,01	0,01	0,02	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,00	0,00	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
15	0	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,00	0,00	0,05	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
16	0	0,01	1,13	0,08	1,14	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,01	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,01	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,01	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,01	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,01	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,00	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	1,13	0,06	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,01	1,13	0,07	1,13	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,01	1,13	0,07	1,14	EN 1993-1-1 6.3.3 Ligning (6.62)
17	0	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	87	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	174	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	261	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	348	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	435	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	522	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	609	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	696	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	783	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

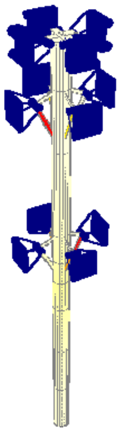
Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	870	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
18	0	0,00	0,00	0,03	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,00	0,00	0,03	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,00	0,00	0,03	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,03	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	336	0,01	0,01	0,03	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,01	0,01	0,04	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,01	0,01	0,04	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
19	0	0,01	0,01	0,04	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,01	0,01	0,04	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,01	0,01	0,03	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,00	0,03	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	168	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,03	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,00	0,00	0,03	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,00	0,00	0,03	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,00	0,00	0,03	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
20	0	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
21	0	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	522	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,50	0,03	0,50	EN 1993-1-1 6.3.3 Ligning (6.62)
22	0	0,01	0,00	0,04	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,01	0,00	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,01	0,01	0,06	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
23	0	0,01	0,00	0,04	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,01	0,00	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,01	0,01	0,06	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
24	0	0,01	0,91	0,07	0,92	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,01	0,91	0,07	0,92	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,01	0,91	0,06	0,92	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,90	0,05	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,90	0,05	0,90	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,90	0,05	0,90	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,90	0,05	0,90	EN 1993-1-1 6.3.3 Ligning (6.62)
25	0	0,01	0,91	0,07	0,92	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,01	0,91	0,07	0,92	EN 1993-1-1 6.3.3 Ligning (6.62)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	174	0,01	0,91	0,06	0,92	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,01	0,91	0,06	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,90	0,05	0,91	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,90	0,05	0,90	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,90	0,05	0,90	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,90	0,05	0,90	EN 1993-1-1 6.3.3 Ligning (6.62)
26	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
27	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
28	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
29	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)

3.2. KAPASITETSKART



Største kapasitetsutnyttelse: 535,21 % (EN 1993-1-1 6.3.3 Ligning (6.62))

VI. Appendix D – Modified Capacity Analysis

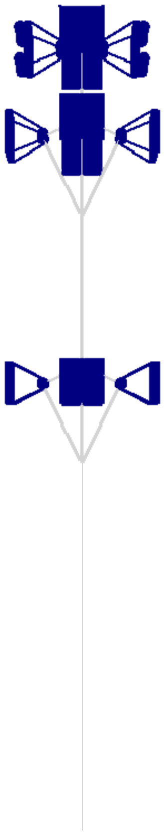
Capacity Analysis - Modified

Note - Centralizer arms made stronger: 60-20 dim instead of 40-10

Beregning utført: 29.05.2012 16:00:13

Focus Konstruksjon 2012

1. KONSTRUKSJONSMODELL OG LASTER



1.1. KNUTEPUNKTSDATA

Nr.	X [mm]	Y [mm]	Z [mm]
1	10000	1000	0
2	10000	1000	1000
3	10000	1000	2000
4	10000	1000	3000
5	10000	1000	3569
6	10000	1000	4000
7	9617	1000	4350
8	10384	1000	4350
9	10000	1000	4521
10	10000	1000	5984
11	9617	1000	6765
12	10384	1000	6765
13	10000	1000	6936
14	10200	1000	7615
15	10000	1000	7615
16	9800	1000	7615
17	10000	617	4350

Nr.	X [mm]	Y [mm]	Z [mm]
18	10000	617	6765
19	10000	1384	4350
20	10000	1384	6765
21	10000	1200	7615
22	10000	800	7615

1.2. TVERRSNITTSDATA

Nr.	Navn	Parametre	
1	Firkantstål 100	A [mm^2]	10000
		Ix [mm^4]	1,2333e+007
		Iy [mm^4]	8,3333e+006
		Iz [mm^4]	8,3333e+006
		Total vekt [kN]	0,62
2	KF Rør 273.0x10.0	A [mm^2]	8262
		Ix [mm^4]	1,4308e+008
		Iy [mm^4]	7,1541e+007
		Iz [mm^4]	7,1541e+007
		Total vekt [kN]	4,85
3	Flatstål 60x20	A [mm^2]	1200
		Ix [mm^4]	1,2640e+005
		Iy [mm^4]	3,6000e+005
		Iz [mm^4]	4,0000e+004
		Total vekt [kN]	0,95

1.3. MATERIALDATA

1 **Stål** Material: Stål

Fasthetsklasse: S355

Varmeutv.koeff.: 1,20e-005 °C^-1 Tyngdetetthet: 77,01 kN/m^3

E-modul: 2,1000e+005 N/mm^2 G-modul: 8,1000e+004 N/mm^2

Karakteristiske fasthetsparametre:

f_y = 355,00 N/mm^2 for godstykkelse <= 40,0 mm
f_y = 335,00 N/mm^2 for godstykkelse <= 80,0 mm
f_y = 335,00 N/mm^2 for godstykkelse > 80,0 mm

1.4. SEGMENTDATA

Seg Nr.	Kn.pkt 1	Kn.pkt 2	Tvsn 1	Tvsn 2	Material
1	1	2	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
2	2	3	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
3	3	4	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
4	4	5	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
5	5	6	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål

Seg Nr.	Kn.pkt 1	Kn.pkt 2	Tvsn 1	Tvsn 2	Material
6	6	9	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
7	9	10	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
8	10	13	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
9	13	15	KF Rør 273.0x10.0	KF Rør 273.0x10.0	Stål
10	9	7	Flatstål 60x20	Flatstål 60x20	Stål
11	9	8	Flatstål 60x20	Flatstål 60x20	Stål
12	7	5	Flatstål 60x20	Flatstål 60x20	Stål
13	5	8	Flatstål 60x20	Flatstål 60x20	Stål
14	13	11	Flatstål 60x20	Flatstål 60x20	Stål
15	13	12	Flatstål 60x20	Flatstål 60x20	Stål
16	11	10	Flatstål 60x20	Flatstål 60x20	Stål
17	10	12	Flatstål 60x20	Flatstål 60x20	Stål
18	20	13	Flatstål 60x20	Flatstål 60x20	Stål
19	13	18	Flatstål 60x20	Flatstål 60x20	Stål
20	18	10	Flatstål 60x20	Flatstål 60x20	Stål
21	20	10	Flatstål 60x20	Flatstål 60x20	Stål
22	19	9	Flatstål 60x20	Flatstål 60x20	Stål
23	17	9	Flatstål 60x20	Flatstål 60x20	Stål
24	17	5	Flatstål 60x20	Flatstål 60x20	Stål
25	19	5	Flatstål 60x20	Flatstål 60x20	Stål
26	14	15	Firkantstål 100	Firkantstål 100	Stål
27	22	15	Firkantstål 100	Firkantstål 100	Stål
28	16	15	Firkantstål 100	Firkantstål 100	Stål
29	21	15	Firkantstål 100	Firkantstål 100	Stål

1.4.1. SEGMENTDATA EN 1993

Seg. nr.	Gamma_M0	Gamma_M1	L_ky [mm]	L_kz [mm]	L_eff [mm]
1	1,05	1,05	1000	1000	1000
2	1,05	1,05	1000	1000	1000
3	1,05	1,05	1000	1000	1000
4	1,05	1,05	569	569	569
5	1,05	1,05	431	431	431
6	1,05	1,05	521	521	521
7	1,05	1,05	1463	1463	1463
8	1,05	1,05	952	952	952
9	1,05	1,05	679	679	679
10	1,05	1,05	420	420	420
11	1,05	1,05	420	420	420
12	1,05	1,05	870	870	870
13	1,05	1,05	870	870	870
14	1,05	1,05	420	420	420
15	1,05	1,05	420	420	420
16	1,05	1,05	870	870	870
17	1,05	1,05	870	870	870

Seg. nr.	Gamma_M0	Gamma_M1	L_ky [mm]	L_kz [mm]	L_eff [mm]
18	1,05	1,05	420	420	420
19	1,05	1,05	420	420	420
20	1,05	1,05	870	870	870
21	1,05	1,05	870	870	870
22	1,05	1,05	420	420	420
23	1,05	1,05	420	420	420
24	1,05	1,05	870	870	870
25	1,05	1,05	870	870	870
26	1,05	1,05	200	200	200
27	1,05	1,05	200	200	200
28	1,05	1,05	200	200	200
29	1,05	1,05	200	200	200

1.5. RANDBETINGELSER

Seg Nr.	X [mm]	Y [mm]	Z [mm]	Frih.gr. X	Y	Z	RotX	RotY	RotZ
11	10384	1000	4350	F	F	F	F		F
10	9617	1000	4350	F	F	F	F		F
22	10000	1384	4350	F	F	F	F	F	
23	10000	617	4350	F	F	F	F	F	
26	10200	1000	7615	F	F		F		F
28	9800	1000	7615	F	F		F		F
29	10000	1200	7615	F	F			F	F
27	10000	800	7615	F	F			F	F
14	9617	1000	6765	F	F	F	F		F
19	10000	617	6765	F	F	F	F	F	
15	10384	1000	6765	F	F	F	F		F
18	10000	1384	6765	F	F	F	F	F	

Forklaring til frihetsgrader: F = fastholdt, (blank) = fri
Tall betyr foreskreven forskyvning [mm]

1.7. LASTTILFELLER

1 Nyttelast

Lastvarighet: Korttidslast
Lasttype: Annen variabel
1 Fordelt last
P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 3569 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 4000 mm
Retning = [1; 0; 0]
Virker på segment: 5

2 Fordelt last
P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 3000 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 3569 mm
Retning = [1; 0; 0]
Virker på segment: 4

3 Fordelt last
P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 2000 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 3000 mm
Retning = [1; 0; 0]
Virker på segment: 3

4 Fordelt last
P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 1000 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 2000 mm
Retning = [1; 0; 0]
Virker på segment: 2

5 Fordelt last
P1 = 2,63 kN/m
X1 = 10000 mm Y1 = 1000 mm Z1 = 0 mm
P2 = 2,63 kN/m
X2 = 10000 mm Y2 = 1000 mm Z2 = 1000 mm
Retning = [1; 0; 0]
Virker på segment: 1

6 Punktlast
P = 39,70 kN
X = 10000 mm Y = 1000 mm Z = 0 mm
Retning = [0; 0; 1]
Virker på segment: 1

1.8. LASTKOMBINASJON

Beregning utført for lastkombinasjon
(1) LASTKOMBO 1

Grensetilstand: Brudd

1,20 * <Konstruksjonens tyngde>
1,50 * Nyttelast

2. STATISKE BEREGNINGER

2.1. KNUTEPUNKTSRESULTATER

2.1.1. Forskyvninger

Nr.	u [mm]	v [mm]	w [mm]	rotX [°]	rotY [°]	rotZ [°]
1	9,1	0,0	0,2	0,0	-0,2	0,0
2	6,2	0,0	0,1	0,0	-0,2	0,0
3	3,4	0,0	0,1	0,0	-0,1	0,0
4	1,1	0,0	0,1	0,0	-0,1	0,0
5	0,3	0,0	0,0	0,0	-0,1	0,0
6	0,1	0,0	0,0	0,0	0,0	0,0
7	0,0	0,0	0,0	0,0	0,0	0,0
8	0,0	0,0	0,0	0,0	0,0	0,0

Nr.	u [mm]	v [mm]	w [mm]	rotX [°]	rotY [°]	rotZ [°]
9	0,0	0,0	0,0	0,0	0,0	0,0
10	0,0	0,0	0,0	0,0	0,0	0,0
11	0,0	0,0	0,0	0,0	0,0	0,0
12	0,0	0,0	0,0	0,0	0,0	0,0
13	0,0	0,0	0,0	0,0	0,0	0,0
14	0,0	0,0	0,0	0,0	0,0	0,0
15	0,0	0,0	0,0	0,0	0,0	0,0
16	0,0	0,0	0,0	0,0	0,0	0,0
17	0,0	0,0	0,0	0,0	0,0	0,0
18	0,0	0,0	0,0	0,0	0,0	0,0
19	0,0	0,0	0,0	0,0	0,0	0,0
20	0,0	0,0	0,0	0,0	0,0	0,0
21	0,0	0,0	0,0	0,0	0,0	0,0
22	0,0	0,0	0,0	0,0	0,0	0,0

2.1.2. Residualkrefter

Nr.	Rx [kN]	Ry [kN]	Rz [kN]	Mx [kN·m]	My [kN·m]	Mz [kN·m]
1	0,00	0,00	0,00	0,00	0,00	0,00
2	0,00	0,00	0,00	0,00	0,00	0,00
3	0,00	0,00	0,00	0,00	0,00	0,00
4	0,00	0,00	0,00	0,00	0,00	0,00
5	0,00	0,00	0,00	0,00	0,00	0,00
6	0,00	0,00	0,00	0,00	0,00	0,00
7	-9,78	0,00	33,89	0,00	0,00	0,00
8	-8,29	0,00	-54,27	0,00	0,00	0,00
9	0,00	0,00	0,00	0,00	0,00	0,00
10	0,00	0,00	0,00	0,00	0,00	0,00
11	1,00	0,00	-4,95	0,00	0,00	0,00
12	1,87	0,00	-0,64	0,00	0,00	0,00
13	0,00	0,00	0,00	0,00	0,00	0,00
14	-0,27	0,00	0,00	0,00	0,00	0,00
15	0,00	0,00	0,00	0,00	0,00	0,00
16	-0,27	0,00	0,00	0,00	0,00	0,00
17	0,03	-0,74	-10,19	0,00	0,01	0,00
18	0,01	-0,43	-2,79	0,00	0,00	0,00
19	0,03	0,74	-10,19	0,00	0,01	0,00
20	0,01	0,43	-2,79	0,00	0,00	0,00
21	-0,04	0,00	0,00	0,00	0,01	0,00
22	-0,04	0,00	0,00	0,00	0,01	0,00

2.2. OPPLEGGSKREFTER

Seg Nr.	X [mm]	Y [mm]	Z [mm]	Rx [kN]	Ry [kN]	Rz [kN]	Mx [kN·m]	My [kN·m]	Mz [kN·m]

Seg Nr.	X [mm]	Y [mm]	Z [mm]	Rx [kN]	Ry [kN]	Rz [kN]	Mx [kN·m]	My [kN·m]	Mz [kN·m]
11	10384	1000	4350	-8,30	0,00	-54,26	0,00	0,00	0,00
10	9617	1000	4350	-9,77	0,00	33,90	0,00	0,00	0,00
22	10000	1384	4350	0,02	0,74	-10,18	0,00	0,01	0,00
23	10000	617	4350	0,02	-0,74	-10,18	0,00	0,01	0,00
26	10200	1000	7615	-0,27	0,00	0,00	0,00	0,00	0,00
28	9800	1000	7615	-0,27	0,00	0,00	0,00	0,00	0,00
29	10000	1200	7615	-0,04	0,00	0,00	0,00	0,01	0,00
27	10000	800	7615	-0,04	0,00	0,00	0,00	0,01	0,00
14	9617	1000	6765	1,01	0,00	-4,94	0,00	0,00	0,00
19	10000	617	6765	0,00	-0,43	-2,79	0,00	0,00	0,00
15	10384	1000	6765	1,86	0,00	-0,63	0,00	0,00	0,00
18	10000	1384	6765	0,00	0,43	-2,79	0,00	0,00	0,00
	Sum			-15,78	0,00	-51,85			

2.3. SEGMENTRESULTATER

Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
1	0	0,00	0,00	0,00	-59,51	0,00	0,20	9,1	0,0	0,2
	1000	1,97	0,00	0,00	-58,82	0,00	3,75	6,2	0,0	0,1
	1000	1,97	0,00	0,00	-58,82	0,00	3,75	6,2	0,0	0,1
2	0	1,97	0,00	0,00	-58,75	0,00	4,14	6,2	0,0	0,1
	1000	7,89	0,00	0,00	-58,06	0,00	7,69	3,4	0,0	0,1
	1000	7,89	0,00	0,00	-58,06	0,00	7,69	3,4	0,0	0,1
3	0	7,89	0,00	0,00	-57,98	0,00	8,09	3,4	0,0	0,1
	1000	17,75	0,00	0,00	-57,30	0,00	11,64	1,1	0,0	0,1
	1000	17,75	0,00	0,00	-57,30	0,00	11,64	1,1	0,0	0,1
4	0	17,75	0,00	0,00	-57,24	0,00	11,95	1,1	0,0	0,1
	569	25,13	0,00	0,00	-56,85	0,00	13,97	0,3	0,0	0,0
	569	25,13	0,00	0,00	-56,85	0,00	13,97	0,3	0,0	0,0
5	0	24,80	0,00	0,00	-24,44	0,00	-24,09	0,3	0,0	0,0
	0	24,80	0,00	0,00	-24,44	0,00	-24,09	0,3	0,0	0,0
	431	14,71	0,00	0,00	-24,18	0,00	-22,73	0,1	0,0	0,0
6	0	14,71	0,00	0,00	-24,13	0,00	-22,56	0,1	0,0	0,0
	0	14,71	0,00	0,00	-24,13	0,00	-22,56	0,1	0,0	0,0
	521	2,96	0,00	0,00	-23,77	0,00	-22,56	0,0	0,0	0,0
7	0	2,87	0,00	0,00	-14,76	0,00	-2,24	0,0	0,0	0,0
	0	2,87	0,00	0,00	-14,76	0,00	-2,24	0,0	0,0	0,0

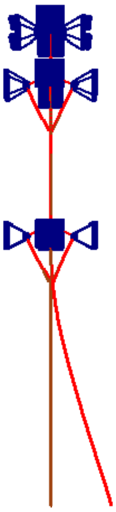
Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
	1463	-0,42	0,00	0,00	-13,75	0,00	-2,24	0,0	0,0	0,0
8	0	-0,41	0,00	0,00	-4,87	0,00	-0,01	0,0	0,0	0,0
	952	-0,42	0,00	0,00	-4,22	0,00	-0,01	0,0	0,0	0,0
	952	-0,42	0,00	0,00	-4,22	0,00	-0,01	0,0	0,0	0,0
9	0	-0,41	0,00	0,00	-1,23	0,00	0,63	0,0	0,0	0,0
	0	-0,41	0,00	0,00	-1,23	0,00	0,63	0,0	0,0	0,0
	679	0,02	0,00	0,00	-0,77	0,00	0,63	0,0	0,0	0,0
10	0	-0,01	0,00	0,00	-5,97	0,00	0,02	0,0	0,0	0,0
	0	-0,01	0,00	0,00	-5,97	0,00	0,02	0,0	0,0	0,0
	420	0,00	0,00	0,00	-5,98	0,00	0,05	0,0	0,0	0,0
11	0	0,08	0,00	0,00	16,36	0,00	-0,26	0,0	0,0	0,0
	0	0,08	0,00	0,00	16,36	0,00	-0,26	0,0	0,0	0,0
	420	-0,03	0,00	0,00	16,35	0,00	-0,23	0,0	0,0	0,0
12	0	0,00	0,00	0,00	34,90	0,00	-0,18	0,0	0,0	0,0
	870	-0,15	0,00	0,00	34,82	0,00	-0,14	0,3	0,0	0,0
	870	-0,15	0,00	0,00	34,82	0,00	-0,14	0,3	0,0	0,0
13	0	0,15	0,00	0,00	-52,82	0,00	-0,16	0,3	0,0	0,0
	0	0,15	0,00	0,00	-52,82	0,00	-0,16	0,3	0,0	0,0
	870	0,03	0,00	0,00	-52,74	0,00	-0,13	0,0	0,0	0,0
14	0	0,02	0,00	0,00	1,32	0,00	-0,08	0,0	0,0	0,0
	0	0,02	0,00	0,00	1,32	0,00	-0,08	0,0	0,0	0,0
	420	-0,01	0,00	0,00	1,30	0,00	-0,05	0,0	0,0	0,0
15	0	0,01	0,00	0,00	1,98	0,00	-0,03	0,0	0,0	0,0
	0	0,01	0,00	0,00	1,98	0,00	-0,03	0,0	0,0	0,0
	420	0,00	0,00	0,00	1,97	0,00	0,00	0,0	0,0	0,0
16	0	0,01	0,00	0,00	-4,89	0,00	-0,02	0,0	0,0	0,0
	522	0,00	0,00	0,00	-4,93	0,00	0,00	0,0	0,0	0,0
	870	0,00	0,00	0,00	-4,96	0,00	0,01	0,0	0,0	0,0
17	0	0,00	0,00	0,00	0,10	0,00	-0,02	0,0	0,0	0,0
	435	0,00	0,00	0,00	0,14	0,00	0,00	0,0	0,0	0,0
	870	0,00	0,00	0,00	0,17	0,00	0,02	0,0	0,0	0,0

Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
18	0	-0,01	0,00	0,00	1,64	0,00	0,03	0,0	0,0	0,0
	420	0,01	0,00	0,00	1,65	0,00	0,06	0,0	0,0	0,0
	420	0,01	0,00	0,00	1,65	0,00	0,06	0,0	0,0	0,0
19	0	0,01	0,00	0,00	1,65	0,00	-0,06	0,0	0,0	0,0
	0	0,01	0,00	0,00	1,65	0,00	-0,06	0,0	0,0	0,0
	420	-0,01	0,00	0,00	1,64	0,00	-0,03	0,0	0,0	0,0
20	0	0,01	0,00	0,00	-2,36	0,00	-0,02	0,0	0,0	0,0
	435	0,00	0,00	0,00	-2,40	0,00	0,00	0,0	0,0	0,0
	870	0,00	0,00	0,00	-2,43	0,00	0,02	0,0	0,0	0,0
21	0	0,01	0,00	0,00	-2,36	0,00	-0,02	0,0	0,0	0,0
	435	0,00	0,00	0,00	-2,40	0,00	0,00	0,0	0,0	0,0
	870	0,00	0,00	0,00	-2,43	0,00	0,02	0,0	0,0	0,0
22	0	-0,01	0,00	0,00	5,18	0,02	0,09	0,0	0,0	0,0
	420	0,03	0,00	0,00	5,20	0,02	0,12	0,0	0,0	0,0
	420	0,03	0,00	0,00	5,20	0,02	0,12	0,0	0,0	0,0
23	0	-0,01	0,00	0,00	5,18	-0,02	0,09	0,0	0,0	0,0
	420	0,03	0,00	0,00	5,20	-0,02	0,12	0,0	0,0	0,0
	420	0,03	0,00	0,00	5,20	-0,02	0,12	0,0	0,0	0,0
24	0	0,01	0,00	-0,01	-8,92	0,00	-0,03	0,0	0,0	0,0
	609	0,00	0,00	-0,01	-8,98	0,00	0,00	0,2	0,0	0,0
	870	0,00	0,00	-0,01	-9,00	0,00	0,01	0,3	0,0	0,0
25	0	0,01	0,00	0,01	-8,92	0,00	-0,03	0,0	0,0	0,0
	609	0,00	0,00	0,01	-8,98	0,00	0,00	0,2	0,0	0,0
	870	0,00	0,00	0,01	-9,00	0,00	0,01	0,3	0,0	0,0
26	0	0,00	0,00	0,00	-0,27	0,00	0,03	0,0	0,0	0,0
	200	0,02	0,00	0,00	-0,27	0,00	0,15	0,0	0,0	0,0
	200	0,02	0,00	0,00	-0,27	0,00	0,15	0,0	0,0	0,0
27	0	0,00	-0,01	0,00	0,00	0,04	0,03	0,0	0,0	0,0
	200	0,02	-0,01	0,00	0,00	0,04	0,15	0,0	0,0	0,0
	200	0,02	-0,01	0,00	0,00	0,04	0,15	0,0	0,0	0,0
28	0	0,00	0,00	0,00	0,27	0,00	0,03	0,0	0,0	0,0
	200	0,02	0,00	0,00	0,27	0,00	0,15	0,0	0,0	0,0
	200	0,02	0,00	0,00	0,27	0,00	0,15	0,0	0,0	0,0

Seg Nr.	Snitt mm	My [kN·m]	Mx [kN·m]	Mz [kN·m]	N [kN]	Vy [kN]	Vz [kN]	u [mm]	v [mm]	w [mm]
29	0	0,00	0,01	0,00	0,00	-0,04	0,03	0,0	0,0	0,0
	200	0,02	0,01	0,00	0,00	-0,04	0,15	0,0	0,0	0,0
	200	0,02	0,01	0,00	0,00	-0,04	0,15	0,0	0,0	0,0

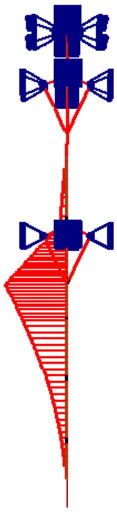
2.4. STATISKE RESULTATER GRAFISK

2.4.1. Forskyvning



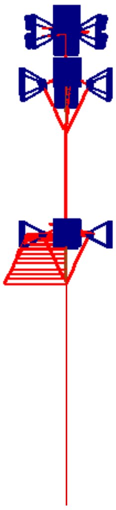
Største forskyvning: 9,1 mm

2.4.2. Moment om y-akse



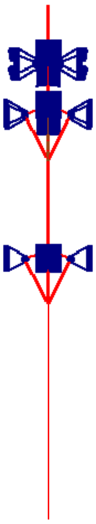
Største moment om y-akse: 25,13 kN·m

2.4.3 Moment om z-akse



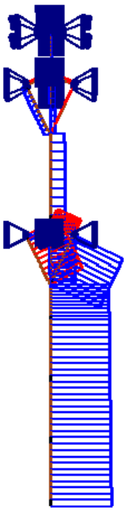
Største moment om z-akse: 0,01 kN·m

2.4.4 Torsjonsmoment



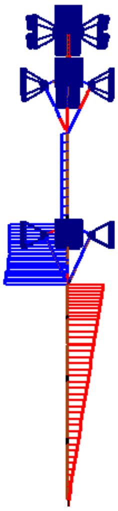
Største torsjonsmoment: 0,01 kN·m

2.4.5 Aksialkraft



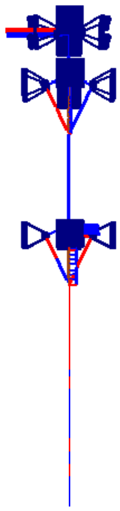
Største aksialkraft: -59,51 kN

2.4.6. Skjærkraft i z-retning



Største skjærkraft i z-retn.: 24,09 kN

2.4.7 Skjærkraft i y-retning



Største skjærkraft i y-retn.: 0,04 kN

3. KAPASITETSKONTROLL

3.1. UTNYTTELSESGRAD EN 1993

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
1	0	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	200	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	300	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	400	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	500	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	600	0,02	0,02	0,03	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	700	0,03	0,02	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	800	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	900	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1000	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
2	0	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,03	0,03	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	200	0,03	0,03	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	300	0,04	0,03	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	400	0,04	0,03	0,04	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	500	0,04	0,03	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	600	0,04	0,04	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	700	0,05	0,04	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	800	0,05	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	900	0,05	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1000	0,05	0,05	0,07	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
3	0	0,05	0,05	0,07	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,06	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	200	0,06	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	300	0,07	0,05	0,08	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	400	0,07	0,06	0,08	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	500	0,07	0,06	0,09	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	600	0,08	0,06	0,10	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	700	0,08	0,07	0,10	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	800	0,09	0,07	0,11	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	900	0,09	0,07	0,11	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1000	0,10	0,08	0,12	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
4	0	0,10	0,08	0,12	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	57	0,10	0,08	0,12	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	114	0,10	0,08	0,13	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	171	0,11	0,08	0,13	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	228	0,11	0,09	0,14	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	285	0,11	0,09	0,14	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	341	0,11	0,09	0,14	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	398	0,12	0,10	0,15	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	455	0,12	0,10	0,15	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	512	0,12	0,10	0,16	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	569	0,13	0,11	0,16	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
5	0	0,11	0,11	0,15	0,14	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	43	0,11	0,10	0,14	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	86	0,11	0,10	0,14	0,13	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	129	0,10	0,09	0,13	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	172	0,10	0,09	0,13	0,12	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	216	0,09	0,08	0,12	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	259	0,09	0,08	0,11	0,11	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	302	0,08	0,08	0,11	0,10	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	345	0,08	0,07	0,10	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	388	0,08	0,07	0,10	0,09	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	431	0,07	0,06	0,09	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
6	0	0,07	0,06	0,09	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	52	0,07	0,06	0,09	0,08	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	104	0,06	0,05	0,08	0,07	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	156	0,06	0,05	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	208	0,05	0,04	0,07	0,06	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	261	0,05	0,04	0,06	0,05	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	313	0,04	0,03	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	365	0,04	0,03	0,05	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	417	0,03	0,03	0,04	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	469	0,03	0,02	0,03	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	521	0,02	0,02	0,03	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
7	0	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	146	0,02	0,02	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	293	0,01	0,01	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	439	0,01	0,01	0,02	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	585	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	732	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	878	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1024	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1170	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1317	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	1463	0,01	0,01	0,01	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
8	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	95	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	190	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	286	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	381	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	476	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	571	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	666	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	762	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	857	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	952	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
9	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	68	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	136	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	204	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	272	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	340	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	407	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	475	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	543	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	611	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	679	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
10	0	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	168	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	210	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	252	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	294	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	336	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,00	0,02	0,02	0,02	EN 1993-1-1 6.3.3 Ligning (6.62)
11	0	0,01	0,01	0,06	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,01	0,01	0,06	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,01	0,01	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,01	0,00	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,00	0,00	0,04	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,01	0,00	0,05	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
12	0	0,01	0,00	0,09	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	87	0,01	0,00	0,09	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	174	0,01	0,01	0,09	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	261	0,02	0,01	0,10	0,01	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	348	0,02	0,01	0,10	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	435	0,02	0,01	0,11	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	522	0,02	0,02	0,11	0,02	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	609	0,03	0,02	0,11	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	696	0,03	0,02	0,12	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	783	0,03	0,02	0,12	0,03	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	870	0,03	0,02	0,12	0,04	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
13	0	0,04	0,67	0,17	0,68	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,04	0,67	0,16	0,67	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,04	0,66	0,16	0,67	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,04	0,66	0,16	0,67	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,03	0,66	0,15	0,67	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,03	0,66	0,15	0,66	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,03	0,66	0,15	0,66	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,03	0,65	0,15	0,66	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,03	0,65	0,14	0,66	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,02	0,65	0,14	0,65	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,02	0,65	0,14	0,65	EN 1993-1-1 6.3.3 Ligning (6.62)
14	0	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	84	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
15	0	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	42	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	84	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	126	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	252	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	336	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	378	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	420	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
16	0	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,06	0,01	0,06	EN 1993-1-1 6.3.3 Ligning (6.62)
17	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	87	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	174	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	261	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	348	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	435	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	522	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	609	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	696	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	783	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

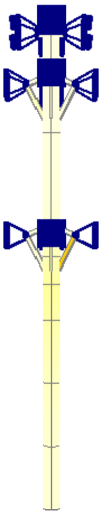
Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	870	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
18	0	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	168	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	210	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	252	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	294	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	336	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
19	0	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	168	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	210	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	252	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	336	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
20	0	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
21	0	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	522	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,00	0,01	0,01	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
22	0	0,00	0,00	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	210	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	252	0,00	0,00	0,02	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	336	0,00	0,00	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,00	0,01	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,01	0,01	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
23	0	0,00	0,00	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	42	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	84	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	126	0,00	0,00	0,01	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	168	0,00	0,00	0,01	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	210	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	252	0,00	0,00	0,02	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	294	0,00	0,00	0,02	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	336	0,00	0,00	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	378	0,00	0,01	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
	420	0,01	0,01	0,02	0,01	EN 1993-1-1 6.3.3 Ligning (6.62)
24	0	0,00	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	174	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,01	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,01	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,01	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
25	0	0,00	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	87	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)

Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	174	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	261	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	348	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	435	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	522	0,00	0,11	0,03	0,11	EN 1993-1-1 6.3.3 Ligning (6.62)
	609	0,00	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	696	0,01	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	783	0,01	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
	870	0,01	0,11	0,03	0,12	EN 1993-1-1 6.3.3 Ligning (6.62)
26	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
27	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
28	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)

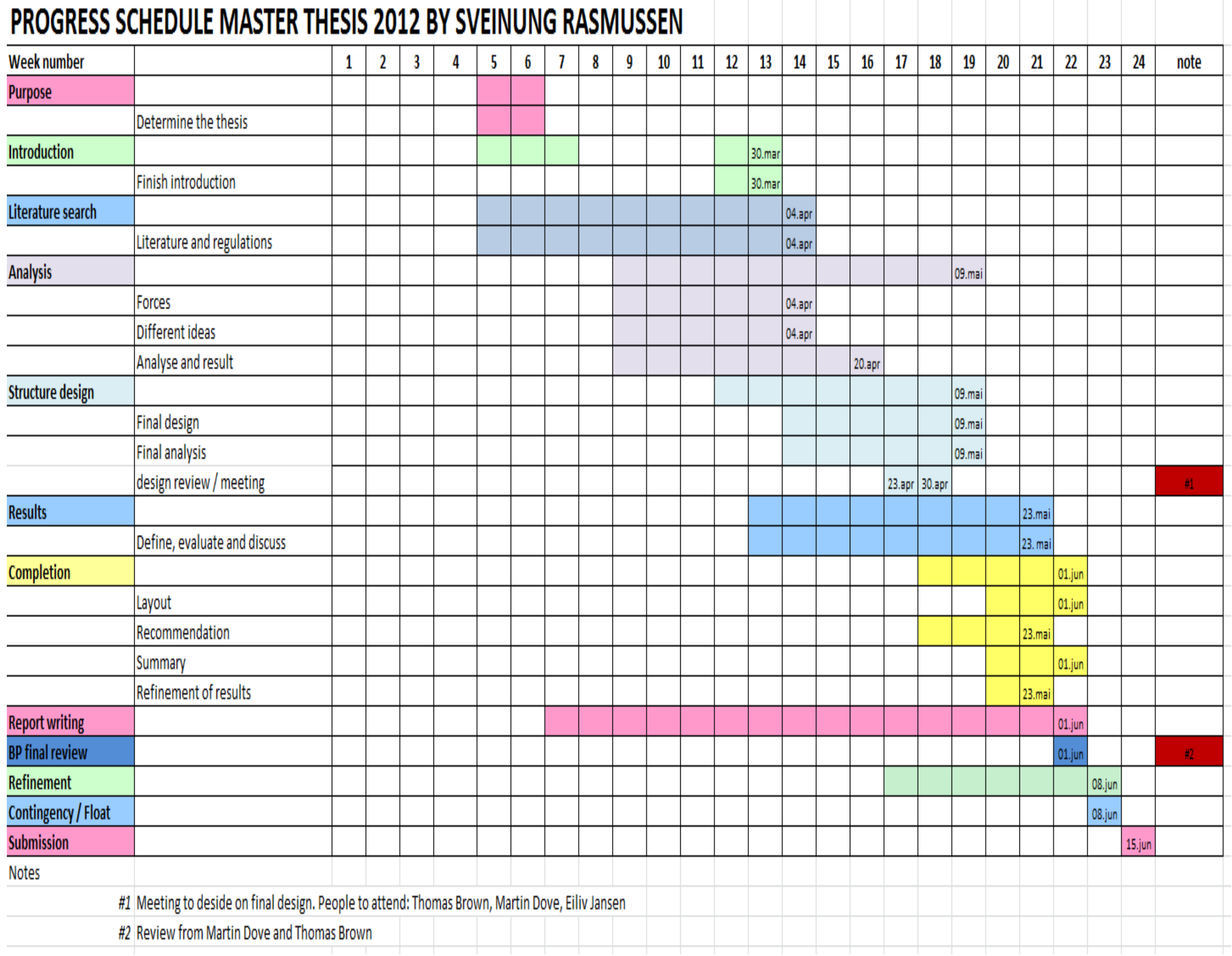
Seg. nr.	Snitt [mm]	Pl.tv	Pl.stab	El.tv	El.stab	Info
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.10 (bøyning, skjær og aksialkraft)
29	0	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	20	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	40	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	60	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	80	0,00	0,00	0,00	0,00	EN 1993-1-1 6.3.3 Ligning (6.62)
	100	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	120	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	140	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	160	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	180	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)
	200	0,00	0,00	0,00	0,00	EN 1993-1-1 6.2.8 (bøyning og skjær)

3.2. KAPASITETSKART



Største kapasitetsutnyttelse: 66,50 % (EN 1993-1-1 6.3.3 Ligning (6.62))

VII. Appendix E – Progress plan



PROGRESS SCHEDULE MASTER THESIS 2012 BY SVEINUNG RASMUSSEN

Week number		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	note
Purpose						■	■																			
	Determine the thesis					■	■																			
Introduction						■	■	■					■	30.mar												
	Finish introduction												■	30.mar												
Literature search						■	■	■	■	■	■	■	■	■	04.apr											
	Literature and regulations					■	■	■	■	■	■	■	■	■	04.apr											
Analysis										■	■	■	■	■	■	■	■	■	■	09.mai						
	Forces									■	■	■	■	■	04.apr											
	Different ideas									■	■	■	■	■	04.apr											
	Analyse and result									■	■	■	■	■	■	■	20.apr									
Structure design													■	■	■	■	■	■	■	09.mai						
	Final design														■	■	■	■	■	09.mai						
	Final analysis														■	■	■	■	■	09.mai						
	design review / meeting																	23.apr	30.apr							#1
Results														■	■	■	■	■	■	■	■	23.mai				
	Define, evaluate and discuss													■	■	■	■	■	■	■	■	23. mai				
Completion																			■	■	■	■	01.jun			
	Layout																				■	■	01.jun			
	Recommendation																		■	■	■	23.mai				
	Summary																				■	■	01.jun			
	Refinement of results																				■	23.mai				
Report writing								■	■	■	■	■	■	■	■	■	■	■	■	■	■	■	01.jun			
BP final review																							01.jun			#2
Refinement																		■	■	■	■	■	■	08.jun		
Contingency / Float																								08.jun		
Submission																									15.jun	

Notes

#1 Meeting to deside on final design. People to attend: Thomas Brown, Martin Dove, Eiliv Jansen

#2 Review from Martin Dove and Thomas Brown