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MASTER'S THESIS

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ABSTRACT

During the life of a well (typically 20 years), a number of operations need to be performed that requires the wellhead/wellhead connector to connect directly to a rig through a riser system. While connected to the vessel the wellhead and conductor system is subjected to many forces from the riser, BOP, waves and vessel.

In deep water a small movement from the vessel can mean a large movement in the riser and BOP stack, which leads to higher loads on the wellhead/wellhead connector.

This project will look into the angle of rotation and displacement of the wellhead datum considering bending stiffness and lateral support of the wellhead. Bending moment and shear forces obtained from a riser analysis of a drilling riser (done in OrcaFlex) will be applied at the wellhead datum.

ACKNOWLEDGEMENTS

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A special thanks goes to the best mum for the extra love, support and good advise she has always given me through life (whether it relates to school, work or personal life she always have a good answer).

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LIST OF SYMBOLS AND ABBREVIATIONS

Symbols:

F – shear force

M – bending moment

L – total length of wellhead system

L2 – length from support A to support B

a – length from wellhead datum to spring location

θ - rotation at wellhead datum

δ - deflection at wellhead datum

γ - non-dimensional peak shape parameter

I – area moment of inertia

The symbols used in Mathcad are explained in the calculations in “Appendix A: Mathcad calculations”, page xii.

Abbreviations:

ALARP – As Low As Reasonable Practicable

CAN™ - Conductor Anchor Node

BOP – Blow Out Preventer

LMRP – Lower Marine Riser Package

WH - Wellhead

WHC – Wellhead Connector

ROV – Remote Operated Vehicle

X-mas tree – Christmas tree

NCS – Norwegian Continental Shelf

DP – Dynamic Positioning

MSL – Mean Sea Level

NPD – Norwegian Petroleum Department

ISO – International Organization for Standardization

API – American Petroleum Institute

RP – Recommended Practice

DNV – Det Norske Veritas

NORSOK – Norwegian Standards

NOK – Norwegian Krone

RAO – Response Amplitude Operator

SCSSV – Surface Controlled Subsurface Safety Valve

BM – Bending Moment

MODU – Mobile Offshore Drilling Unit

JONSWAP – Joint North Sea Wave Project

C_A – Added mass coefficient

C_D – Drag coefficient

H_s – Significant wave height (highest 1/3 of the waves)

H_{max} – Maximum wave height ($1.9*H_s$)

T_p – Spectral peak period

T_z – Zero-crossing period (mean of the periods for a given wave record)

T_s – Time period corresponding to H_s

T_{max} – Time period corresponding to H_{max}

ID – Inner Diameter

OD – Outer Diameter

Sm^3 – Standard cubic metres (volume of gases changes with pressure and temperature)

O.e - oil equivalents

Arc length – total length of marine riser (OrcaFlex)

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BACKGROUND FOR THE PROJECT

This project is conducted on the basis of a problem encountered when evaluating drilling with or without the NeoDrill development "CAN". The CAN was rejected at a project on the basis of that the foundation was to stiff. This project will look into the effects bending moment and shear force at the wellhead datum have on the angle of rotation and deflection at wellhead datum during conventional drilling, followed by a discussion with the use of the CAN.

KEY OBJECTIVES FOR THE PROJECT

- → Give an introduction to drilling, drilling rigs and subsea drilling systems.
- → Explain and compare conventional subsea drilling and drilling with the NeoDrill CAN foundation development.
- → Study forces in the wellhead connector as a function of boundary conditions for a subsea wellhead.
- → Establish mechanical models for wellhead lateral support (conventional and with CAN).
- → Model BOP and riser for actual load cases in Orcaflex.
- → Analysis results for Wellhead forces.
- → Conclude and recommend further work.
- → Presentation of detailed information from OrcaFlex and calculations in appendices.

1 Introduction

The stress/forces/moments that a subsea wellhead needs to handle mainly arises from:

- The vertical and lateral reactions of the riser lower flex joint
- The gravity of the BOP and casing string
- The wave and current forces on the BOP and wellhead
- The vertical and lateral resistance of seabed soil

If the bending moment of the wellhead/wellhead connector overruns the design limit it will cause the wellhead to collapse. As the water depth limits get deeper and deeper, one gets more and more movements in the riser and the lower flex joint from smaller offsets during drilling.

NeoDrill is a company started by Harald Strand. The company have developed a foundation called CAN™ (Conductor Anchor Node) to deal with the swaying BOP. It makes the foundation very stiff. In one project in the Barents Sea the CAN™ was even concluded to be to stiff to be used for the drilling operation (Nergaard).

The stiffness of the CAN™ can be calculated as infinitely stiff. If it is possible to find the forces in the wellhead/wellhead connector with the possibility of varying the lateral support and stiffness the CAN™ could probably be used in a wider matter.

In this project it is presented a global analysis that investigates the angle of rotation and deflection in the wellhead from forces in the lower flex joint during a drilling operation.

Before the analysis is presented it is given an introduction to drilling history, conventional drilling, drilling with the NeoDrill CAN and background information/theory for the analysis.

1.1 Thesis organization

This project is developed into the following 10 chapters with the purpose of giving an organized and well-presented project for the reader.

Chapter 1: Introduction

Chapter 2: Drilling In Brief

Chapter 3: Conventional Subsea Drilling

Chapter 4: Drilling With The NeoDrill CAN™ Foundation Concept Development

Chapter 5: Background Information For Analysis

Chapter 6: Analysis Values And Parameters

Chapter 7: Model BOP And Riser For Load Cases

In the end it is found appendices with calculations and results from OrcaFlex.

1.2 Assumptions and limitations

For the purpose of obtaining the forces and simplify the calculations the following assumptions and limitations has been made:

- Analysis values are obtained from standards, recommended practices and previously written master theses.
- BOP is calculated as infinitely stiff.
- The wellhead system is calculated with mechanical models to obtain forces at the wellhead datum.
- The contribution from the soil is calculated as a spring with stiffness, K, modelled as a roller in the mechanical models.
- The stick-up height of the wellhead is not considered in the calculations.
- The spring stiffness with the use of CAN is not obtained but is assumed to be a lot stiffer than the spring stiffness using conventional drilling.
- The shear force and bending moment are transferred from the flex joint to the wellhead datum to find angle of rotation and deflection at the wellhead datum using the values obtained from OrcaFlex.

2 Drilling In Brief

2.1 History

The earliest known people to build a drilling rig were the Chinese. This happened 3000 years ago and the target was to get a hole down to the freshwater mud. They used a drilling bit attached to a bamboo "drive" [Statoil, 2007].

Modern drilling for hydrocarbons started in Pennsylvania in 1859. This method consisted of a drilling tower and a steam engine "drive" that pushed the drilling tool up and down [Statoil, 2007].

Rotary drilling for hydrocarbons started at the beginning of this century and we still use it today. Weight is applied on the drill bit to make it rotate and it is applied a continuous circulation of mud which removes the drill cuttings and cools down the bit. At the beginning the search for hydrocarbons was extended to the bottom of rivers and swamp areas. The road from here to offshore drilling was not very far [Statoil, 2007].

As the water depths got deeper and deeper, new platforms needed to be developed starting with the fixed platform moving further to jack-ups and after this to floating platforms (semi-submersibles) and ships.

Drilling in the past (1970-1980) had a typical well inclination limited from 50 to 60 degrees from vertical and a drilling length from 3000 to 4000 meters. Today we have drilling lengths from 6000 to 10 000 metres with a long section of horizontal wells (90 degrees or more). The long reach wells allow fewer platforms to be used to drain the field [Stangesland, 2012].

2.2 Charts from the NPD (Norwegian Petroleum Department)

Many wells are drilled on the Norwegian Continental Shelf. Appraisal wells are drilled after it has been made a discovery of oil and gas. These wells are drilled to establish the limits of the reservoir. The wildcat wells are drilled in places were there are no confirmed discoveries of oil and gas fields. From the chart represented below (figure 2-1) it is shown that there is a good forecast for the year 2014 on the NCS with more than 40 wells in total. It also represents the wildcat- and the appraisal wells drilled from 1982 until 2013. There are more predicted wells in 2014 than many of the previous years.

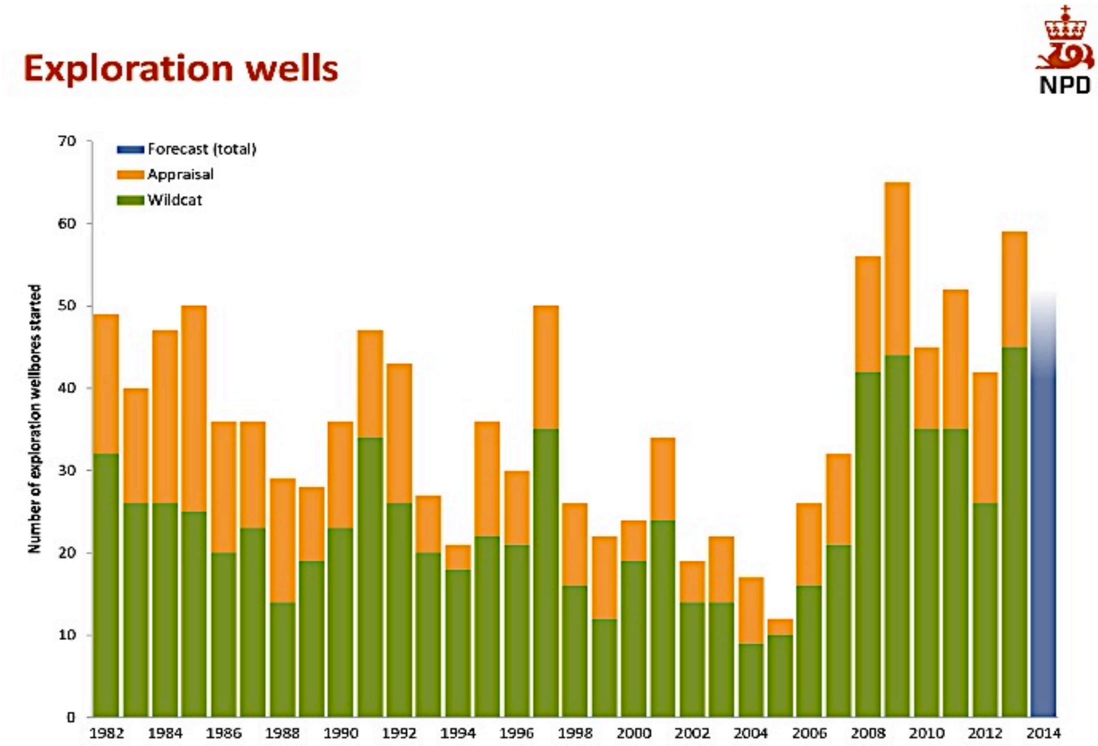


Figure 2-1: Exploration wells drilled on the NCS [NPD, 2014]

High investments are made on the Norwegian Continental Shelf throughout the years. It is hard to imagine these numbers if you compare with “other” investments. In the chart represented in figure 2-2 below the investments in various facilities are shown in billion NOK. From 2014 to 2017 the investments are predicted to exceed more than 200 billion NOK!

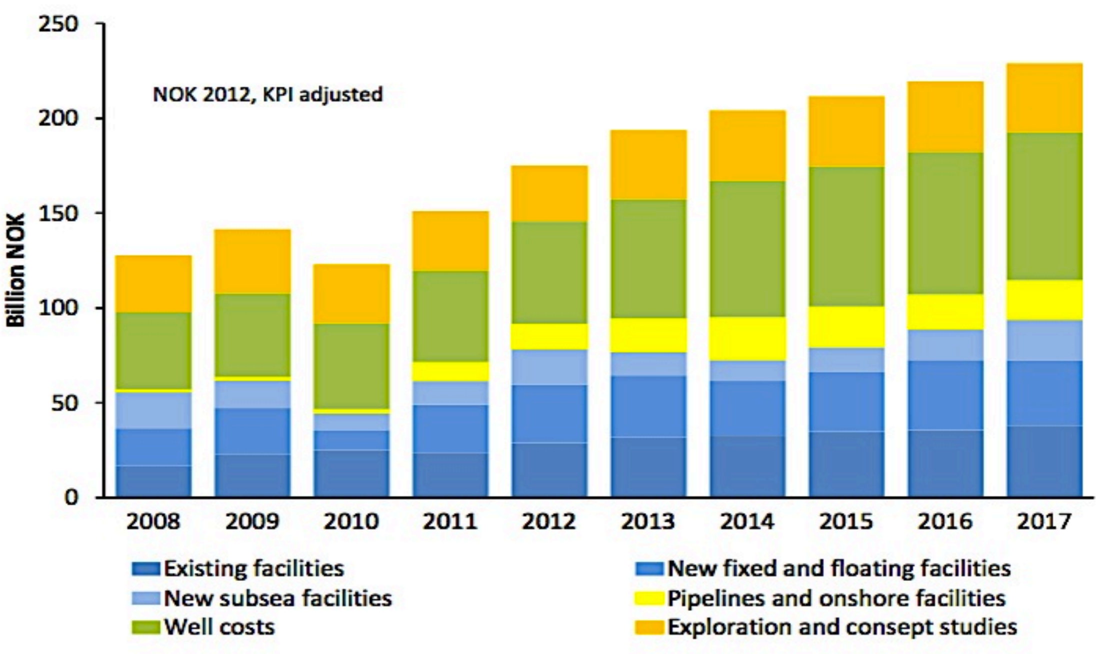


Figure 2-2: Investments made on the NCS [NPD, 2014]

It is still oil left on the Norwegian Continental Shelf, and it will probably last through some more generations. From the chart represented in figure 2-3 below the total (blue), gas (red) and liquid (green) are represented in billion Sm^3 oil equivalents. From the circular chart it is shown that there is predicted more oil and gas resources left than already produced, sold and delivered.

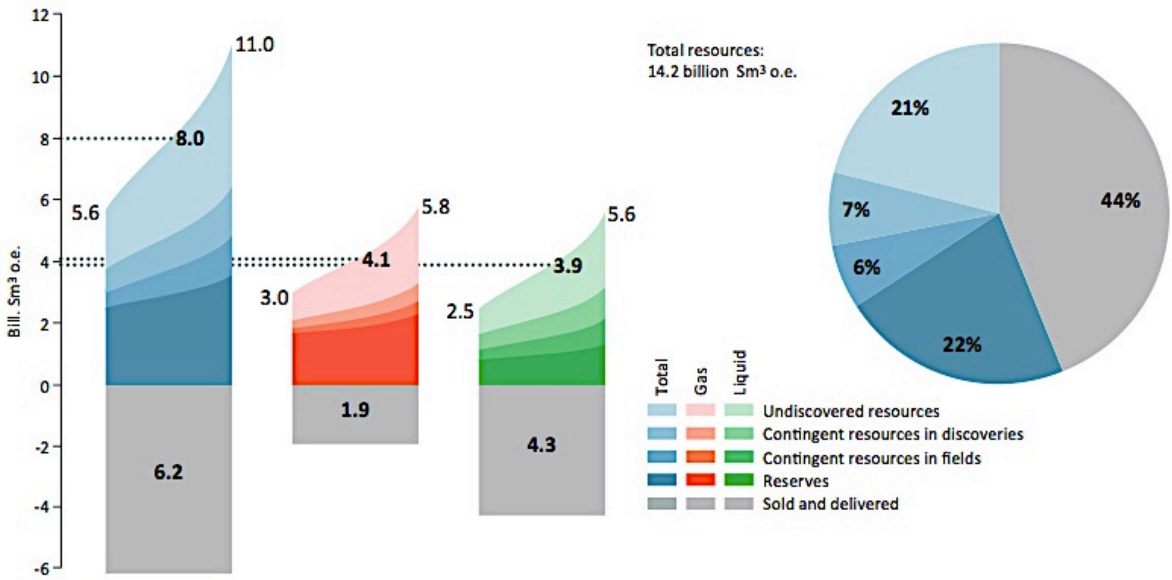


Figure 2-3: Recoverable resources per 31.12.13 [NPD, 2014]

It is always room for improvement in all industries. With new drilling techniques, equipment and vessels, more oil and gas can be produced from existing fields as well as new fields. An important aspect is to get the oil and gas industry “cleaner” with respect to the environment.

2.3 Drilling rigs

Offshore drilling rigs or platforms can be grouped under three main categories:

- Self-contained fixed platforms
- Fixed platforms with floating drilling tenders
- Mobile drilling units

The different types of drilling rigs are represented in the following figure 2-4:

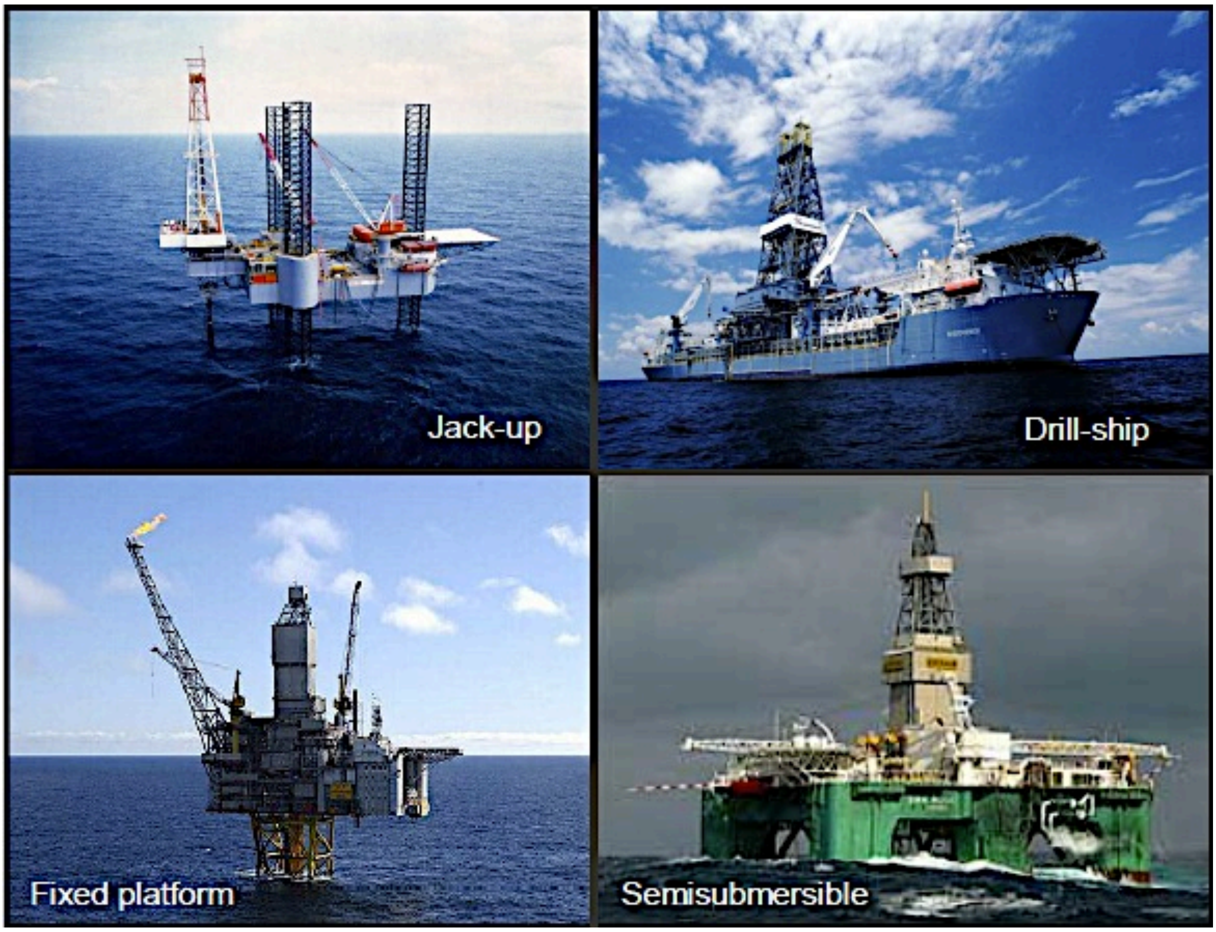


Figure 2-4: Drilling rigs [Odland, 2012]

Maximum water depths are approximately:

- Drillship: 3000 meters
- Semi-submersible: 2500 meters
- Jack-up: 130 meters
- Fixed platform: 250 meters

The mobile units are the rigs that do virtually all the exploratory drilling for the oil and gas industry. Fixed platform are basically production units, but some of them may also have drilling facilities.

The operator must choose the rig that is most capable of doing the job efficiently and safely, and be a type that is suitable for the nature of the operation. In general the exploration wells are drilled by floaters (semi-submersibles, drill ships and barges) or by self-elevating jack-up rigs, while development wells drilled to exploit a field already discovered/existing, are often drilled from fixed platforms [Odland, 2012].

Advantages and disadvantages for choosing a drilling rig:

Rig type	Advantages	Disadvantages
Drillship	▪ Can be used in deep water ▪ Good speed and mobility ▪ Suitable for dynamic positioning	▪ Very dependant on weather because of bad heave, roll and pitch motion characteristics
Semi-submersible	▪ Can be used in deep water ▪ High mobility (self-propelled) ▪ Can take large deck loads (especially the big 6th generation rigs) ▪ Good heave motion characteristics	▪ BOP at seabed ▪ Dependant on assistance from a vessel when placing the anchors ▪ Not very suitable for dynamic positioning (6th generation have it)
Jack-up	▪ Low operation expenses ▪ Not dependant on weather ▪ BOP at platform deck ▪ No mooring system ▪ Simplified equipment and drilling procedures because of fixed platform	▪ Cannot be used in deep water ▪ Unstable under relocation ▪ Dependant on weather and towing vessel under relocation

Table 2-1: Advantages and disadvantages for selecting drilling rigs

The drilling rig can be looked at as the machine to drill a wellbore. In this thesis a semi-submersible drilling rig is chosen for analysis, because it can be used in deep water and have good heave motion characteristics. Details about the vessel will be given in chapter 7.

Major components of a drilling rig include the:

- Mud tanks/pits: for mixing mud (mud engineer) and getting the right density for the mud
- Mud pumps: to pump the mud from mud pits up to the drill string and down to the wellbore (on the jack-up West Epsilon the capacity of these pumps is about 1000 litres per minute!)
- Derrick: located above the drill floor, accommodation for several pipe handling machines including the top drive
- Draw-works (hoisting machinery/winch): main function of raising and lowering the traveling block that allows the drill string to move up and down

- Rotary table/top drive: the mechanical device that provides the clockwise torque on the drill string to make it possible to drill a well. Top drive is the newest version of the rotary table and eliminates the swivel. It extends the drilling depth for a stand of drill pipe from 9 metres (rotary table) to 18-27 metres (top drive).
- Drill string, drill pipe and drilling riser/Marine riser: to be explained in detail in chapter 2.4.
- Blowout preventer (BOP): to be explained in detail in chapter 2.4.
- Power generator equipment: the huge power needed on a drilling is usually supplied by diesel engines.
- Auxiliary equipment: electronic systems on the rig. Some rigs have DC (direct current) power while most of the big new rigs have AC (alternating current) power.

2.4 Drilling system

Drilling with mobile drilling units has increased rapidly over the years. This is related to the water depth that keeps getting deeper and deeper. In figure 2-5 below the chart represents all the development fields drilled from mobile facilities and the development fields drilled from permanently placed drilling facilities from 1980 until predicted for 2014. It is an increasing trend for the mobile units with a peak for the development wells in 2001.

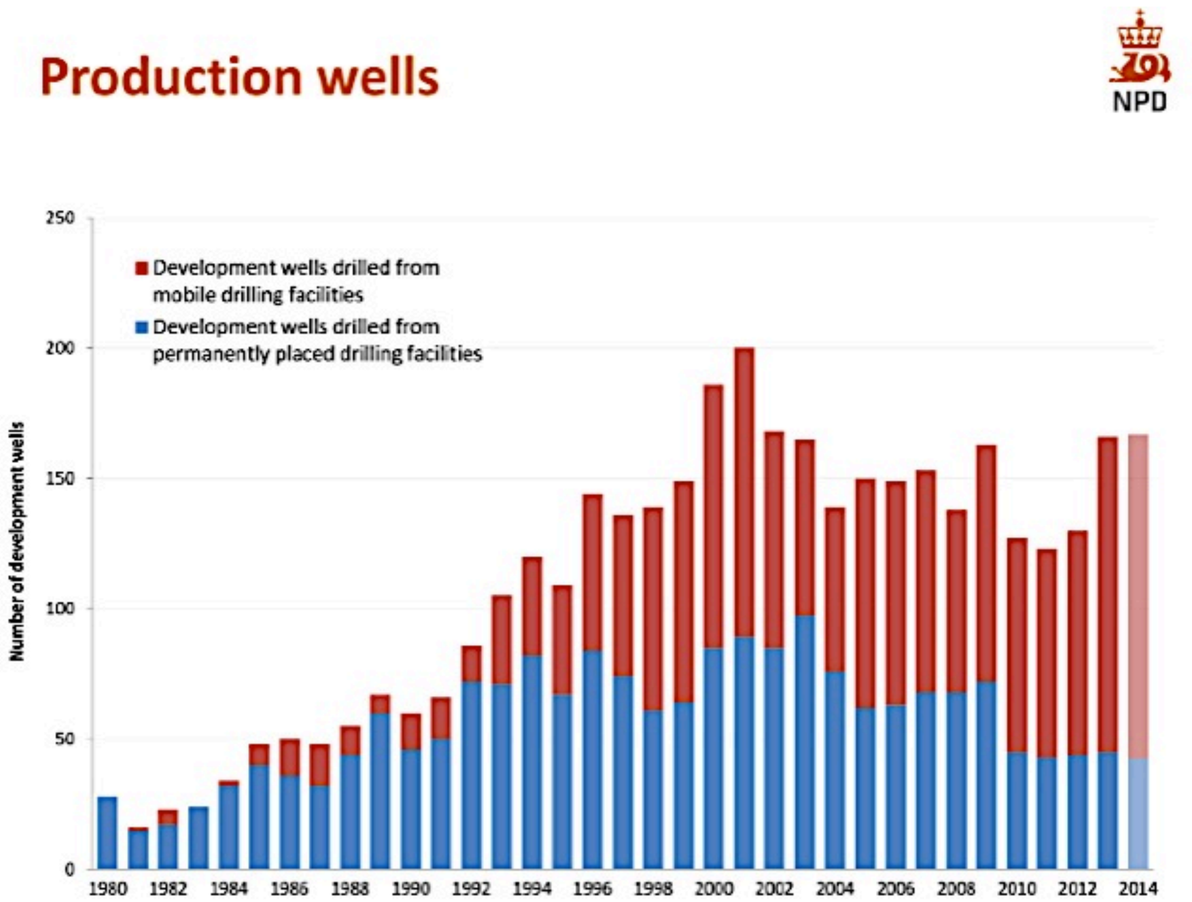


Figure 2-5: Number of development wells on the NCS [NPD, 2014]

Several components are needed for drilling a well, some located at the top of the rig and some located subsea. In this section a drawing will represent the key components of a drilling system using a semi-submersible drilling rig and each component will be explained in detail.

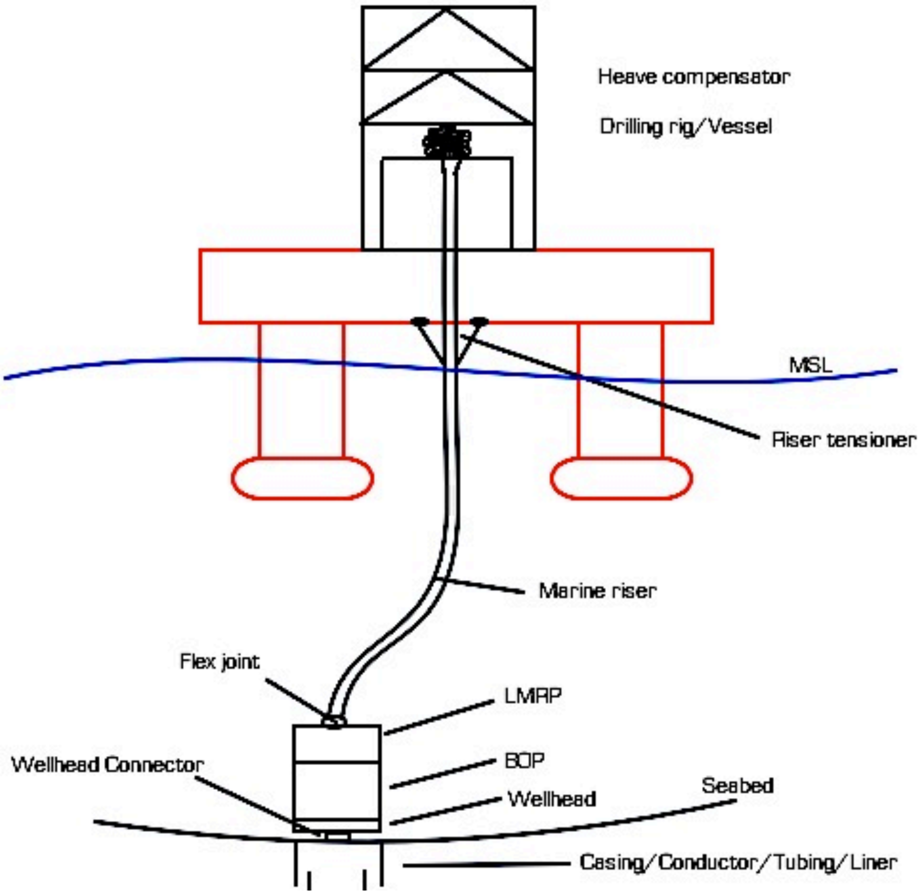


Figure 2-6: Drilling system semi-submersible

Heave Compensator

It exists different types of heave/motion compensators including crown mounted-, direct line- and drill string compensators. They are all designed to compensate for the vertical movement of offshore drilling rigs due to the heave motion. When it is bad weather the heave compensator are vital to prevent damage on the riser [NOV, 2014].

Riser Tensioner

The riser tensioner is also a motion compensator, which is used to apply constant tension in the marine riser to prevent riser buckling (due to compression) and to tension guidelines and pod lines [NOV, 2014].

Marine Riser

Drilling risers are categorized into marine drilling risers and tie-back drilling risers. The tie-back drilling risers are used with a surface BOP, and the marine drilling risers are used when the BOP is deployed at the seafloor.

The marine riser has its own system that connects the rig topside to the BOP and makes it possible to receive return mud and cuttings from the annulus. When the equipment is run and pulled it is the risers task to control/guide the tool into and out of the borehole. The marine riser consists of joints with approximately 10 to 15 meters between each joint [See Figure 2-7] [Statoil, 2007].

When the rig moves it create vertical and horizontal movements. The marine riser absorbs the vertical and the horizontal forces. A slip joint located on the rig bottom side absorbs the vertical forces and a flex joint located above the BOP absorbs the horizontal forces. A tensioner system is attached to the marine riser as previously explained. The amount of tension that must be applied to the riser depends on weight, buoyancy, wave and current forces, weight of internal fluids and allowances used for the design [Statoil, 2007].

The international standard ISO 13624-1:2009 covers the design, selection, operation and maintenance of marine riser systems for floating drilling operations. A more detailed drawing of the entire marine riser system is shown in figure 2-8 on the next page.

Figure 2-7 shows the spider, gimbal, telescopic joint, and the riser joints with the flex joint at the bottom. The spider has retractable jaws to hold and support the riser during running of the riser. The spider is found in the rotary table on the drill floor. The gimbal is found between the spider and the rotary table. Its purpose is to reduce shock and to evenly distribute load caused by the vessels pitch/roll motion, on the spider and riser sections. The telescopic joint/slick joint is designed to prevent damage to the umbilicals where they pass through the rotary table. It also protects the riser from damage due to the vessel heave motion [Bai & Bai, 2012].

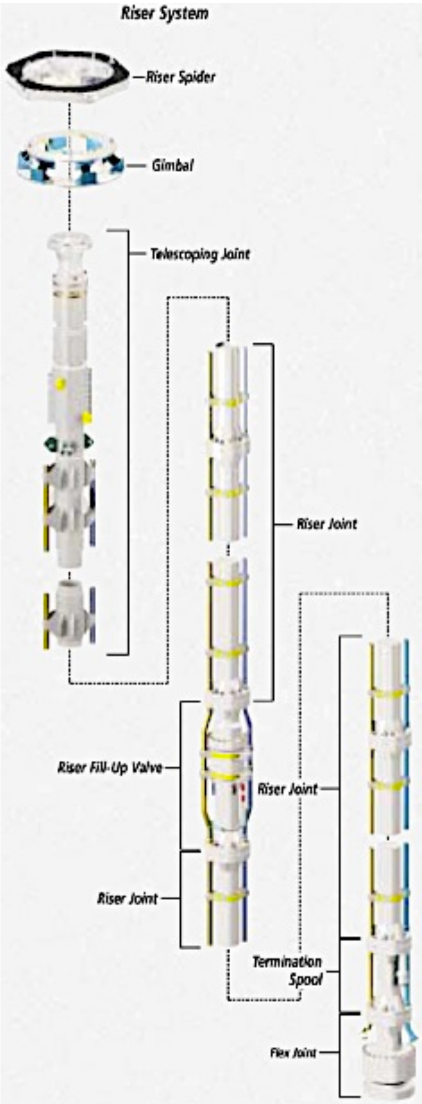
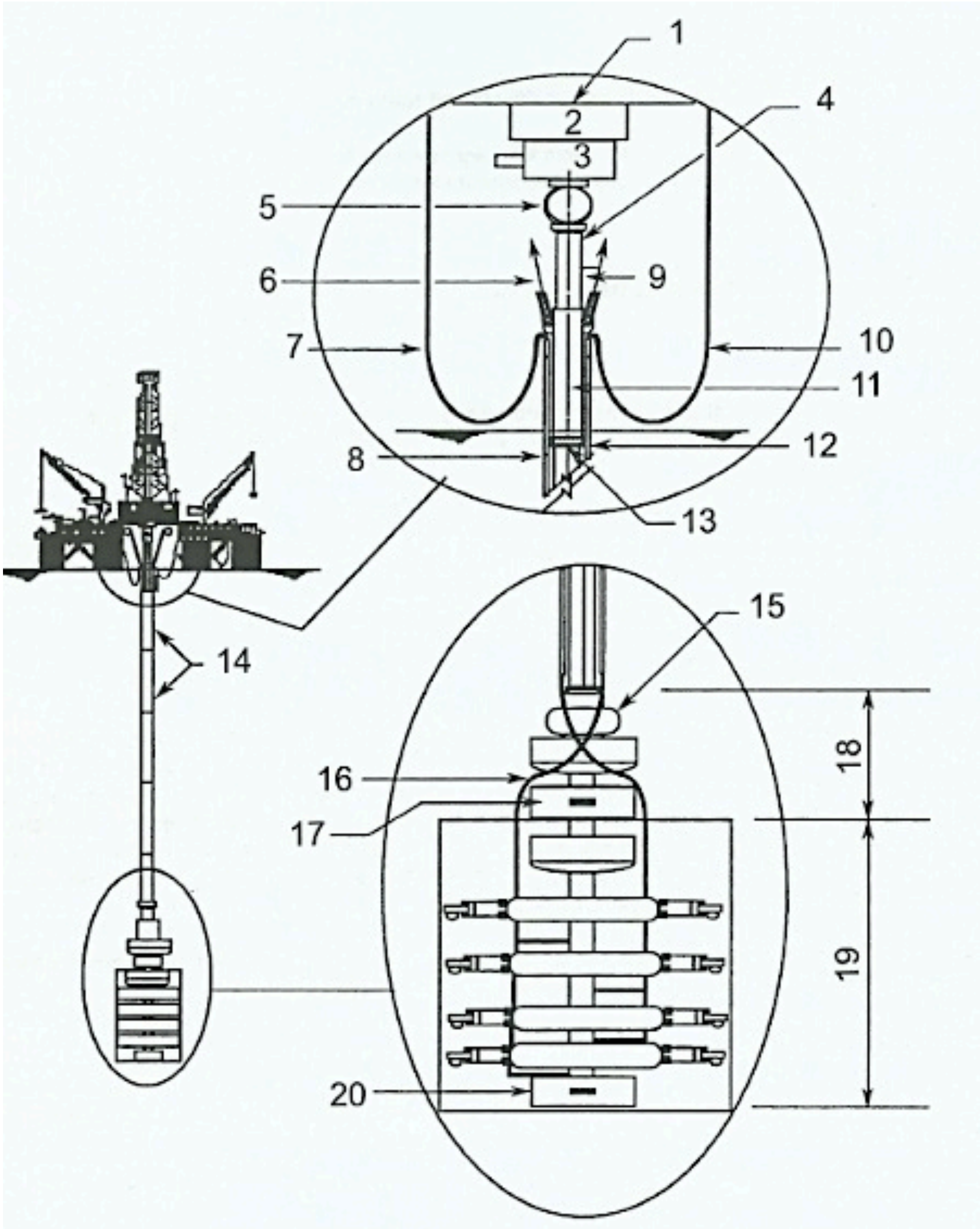


Figure 2-7: Riser system [Bai & Bai, 2012]



1 Rotary Kelly bushing (RKB)	8 Choke line	15 Flex/ball joint
2 Rotary	9 Fleet angle	16 Riser/BOP jumper hose
3 Diverter	10 Kill drape hose	17 LMRP connector
4 Telescopic joint inner barrel	11 Telescopic joint outer barrel	18 LMRP
5 Flex/ball joint	12 Kill line	19 BOP stack
6 Tensioner line	13 Riser coupling	20 Wellhead connector
7 Choke drape hose	14 Marine riser joints	

Figure 2-8: Marine riser system [ISO 13624-1]

Diverter

The diverter is a mini BOP whose task is to close around the drill pipe when gas or other fluids enter the hole under pressure. The flow is then diverted (by flare towers) away from the rig/wellbore.

Choke and kill lines and drill string

The choke and kill lines are attached outside of the riser pipe with braces. These lines are used to control high-pressure events by circulating the high pressure out of the wellbore while pumping heavier mud into the hole. If there is no possibility of getting the pressure under control by heavy mud the well is killed by pumping cement down the kill line. The drill string allows the circulation of mud. The most important function of the mud is to cool the drill bit, lubricate the drill string, and keep the hole free of cuttings by forcing it to circulate to the top. The mud also prevents wall cave-ins.

Flex joint/ball joint

There are normally two flex joints (also called ball joint) in a riser system called upper and lower flex joint. Their main task is to reduce local bending stresses at the top and bottom of the marine riser.

LMRP

The lower marine riser package is basically the upper part the BOP and it hence a mini BOP consisting of valves and connections for connecting the BOP to the marine drilling riser.

BOP

The blowout preventer should seal the well with specialized valves if it exists uncontrolled pressure and flow from the well. It is a part of the well control system to prevent blowouts and monitor well pressure and flow.

Wellhead And Wellhead Connector

The wellhead and wellhead connector is located below the BOP. It is a pressure-containing and structural anchoring point on the seabed for the drilling and completions systems. The wellhead consists of internal profiles for support of the casing strings and isolation of annulus. In addition it provides guidance, mechanical support and connection of the systems used to drill and complete the well (BOP and x-mas tree) (Bai & Bai, 2012).

According to Bai & Bai, 2012 the subsea wellhead has the following functional requirements:

- Provide support and interface with x-mas tree and BOP
- Be able to withstand all loads applied to the wellhead and wellhead connector from drilling, completion and production operations
- Ensure that the conductor housing and wellhead housing have alignment, concentricity and verticality

Conductor/Casing/Liner/Tubing

The different names are to separate the depth (location in the well) but all of them are different "pipes" with varying diameter and size going from large diameter to smaller further down in the well. For example for a 6000 metres deep well the diameter can get as small as 2 ½" = 6 cm. The setting depth is dependant on pore pressure and fracturing pressure. The different types are installed/run in the following order:

1. Conductor
2. Casings (different dimensions varying with drilling depth)
3. Liner
4. Production tubing

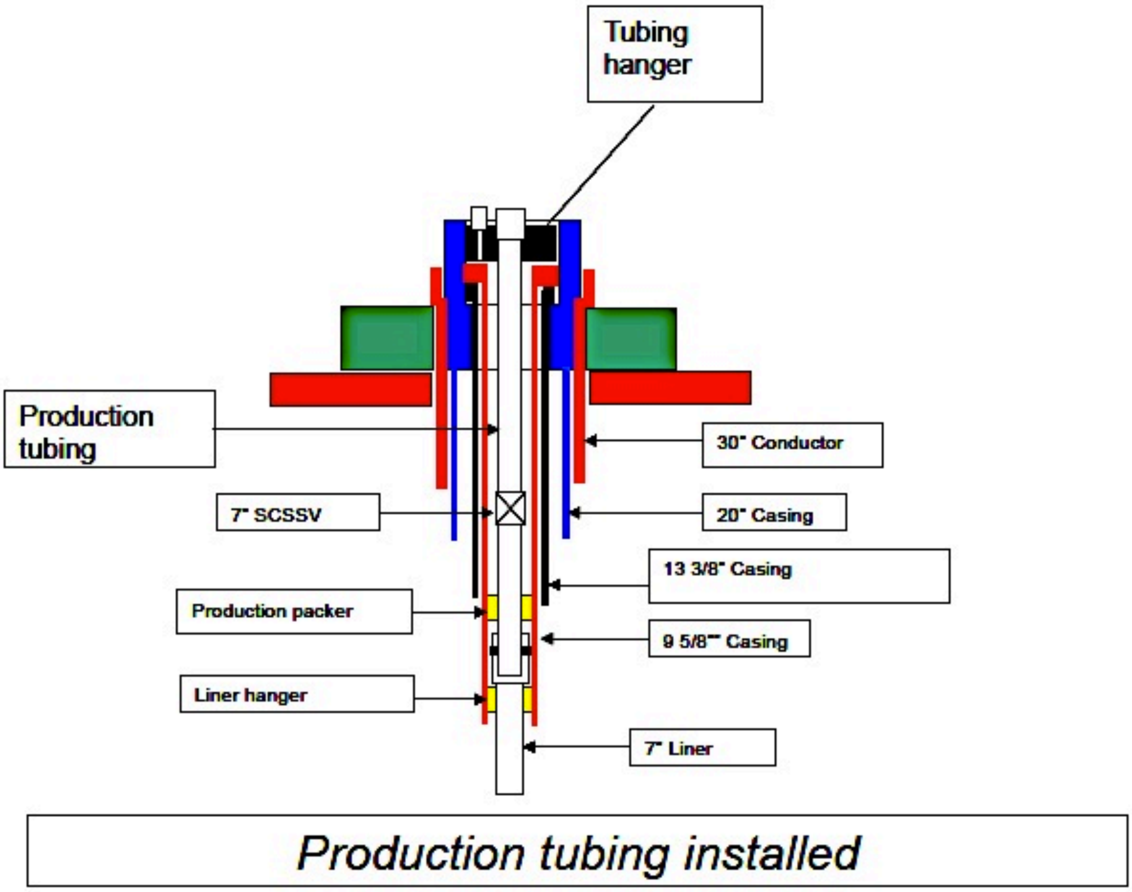


Figure 2-9: Conductor/Casing/Liner/Tubing overview (Stangesland, 2012)

3 Conventional Subsea Drilling

3.1 General

The drilling sequence/program for a well will almost never be exactly the same, but a general drilling sequence using a mobile drilling unit will be presented in this chapter together with some deep water drilling challenges.

3.2 Deep water drilling challenges

The main problem with deep water drilling is the soft sediments with low formation strengths. The problem leads to a low margin between pore pressure and fracture pressure witch means that a large number of casings are needed.

Well killing can be difficult in deep water due to high pressure in kill- and choke lines.

In deep water drilling it is a time-consuming riser/BOP handling, the large volume of drilling fluid in the riser and heavy loads leads to large and expensive drilling vessels.

3.3 Drilling sequence roughly explained

From figure 3-1 below a typical drilling sequence is explained:

STEP 1: A temporary guide base is run down to the seafloor supported by four guidelines. The guide base should provide support and guidance when the 36" hole opener and the 30" casing is run.

STEP 2: When the temporary guide base is placed horizontally (checked by underwater camera and an inclination indicator), run the 36" hole opener 60 to 80 meters below the seabed.

STEP 3: Viscous fluid is now used to prevent the wall from sliding out when the drillstring gets pulled. The 30" casing is run and cemented in place after the permanent guide base is hanged off. The 30" casing is cemented all the way to the surface.

STEP 4: The 26" hole is now drilled in two sections without a riser. First a 12 ¾" pilot hole is drilled down to the full depth. The hole is then expanded with an underreamer. The cuttings return will go to the sea bottom. If it is used a subsea template with several slots the cuttings will be transported 50-100 meters in a cutting hose. In some cases the 26" pilot hole will also be drilled with a riser to get the cuttings back to the rig.

STEP 5: Run and cement the 20" surface casing with 18 ¾" wellhead. The wellhead is landed in the permanent guide base. Normally the 20" surface casing is cemented all the way to the seabed.

STEP 6: After the wellhead is established the BOP (Blow Out Preventer) is used for all the remaining drilling (together with the LMRP). The BOP is attached to the top of the wellhead.

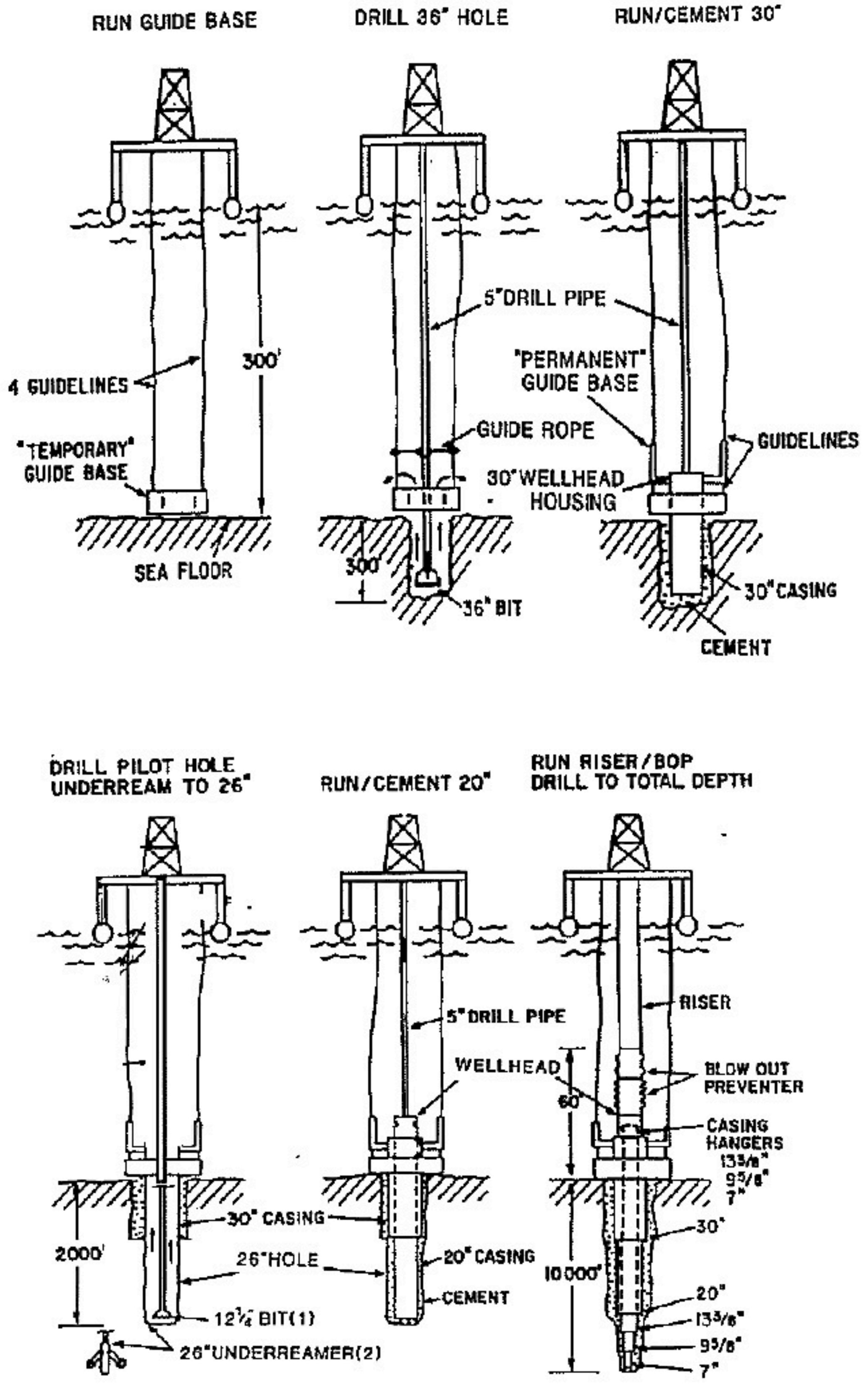


Figure 3-1: Subsea drilling sequence using a mobile drilling unit [Stangesland, 2012]

3.4 Typical force distribution on a subsea wellhead system

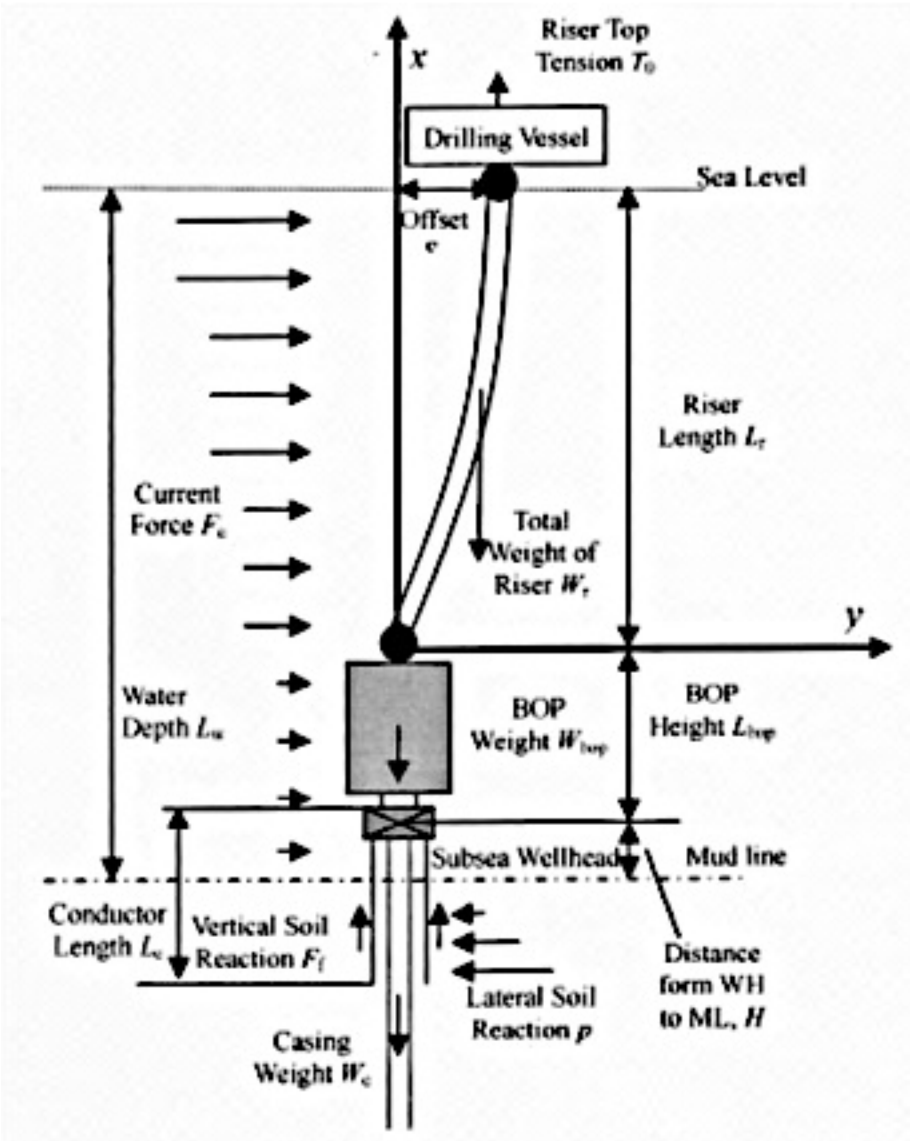


Figure 3-2: Typical force distribution on a wellhead system [Guan, Su & Su, 2010]

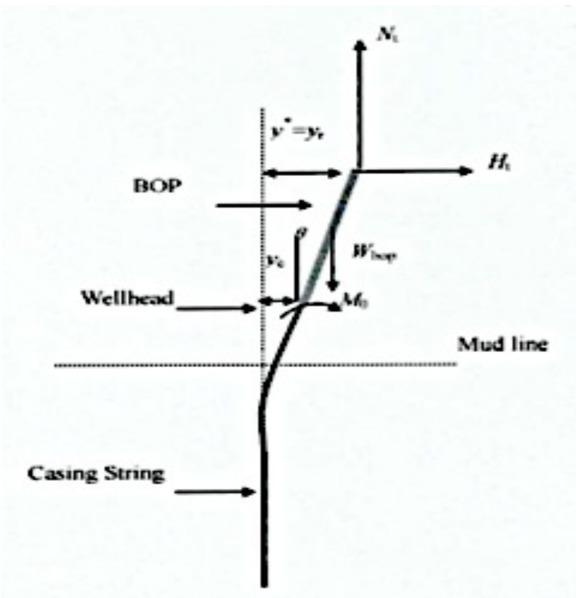


Figure 3-3: Forces acting on a wellhead [Guan, Su & Su, 2010]

4 Drilling With The NeoDrill CAN™ Foundation Concept Development

4.1 General

The CAN™ (Conductor Anchor Node) is a new well foundation developed by NeoDrill. It eliminates the "weak link" in current well design by providing a stable and reliable foundation, which mitigates the risks of conductor damage caused by bending and fatigue. Up to this date the CAN™ technology has been applied for conventional as well as more technical challenging wells in various fields on the NCS. It can be used for both exploration and single production wells. When the CAN™ is used for production it is left in the well and when it is used for exploration the CAN™ will be retrieved with the same vessel that installed it [Sivertsen & Strand, 2011].

The CAN™ will mitigate the risks of the well becoming over-loaded by undesired, accidental loads, e.g.: as a result of a rig drive off/drift off situation. To achieve this substantial carrying capacity is mobilised through the CAN's large cross-sectional area. The CAN™ provides sufficient load capacity for carrying the BOP as well as X-mas trees [Sivertsen & Strand, 2011].

The concept will reduce the rig time as it enables pre-rig conductor installation. This will reduce the top-hole construction costs and rig failure risk exposure [NeoDrill 2014].

The CAN™ supports all conductors: driven, jetted, drilled and cemented and leave the conductor motionless and supported during set-up [NeoDrill 2014].

Figure 4-1: NeoDrill CAN™ development [NeoDrill 2014]

4.2 CAN™ technology

The design is a special anchor type of structure. It consists of an open ended (down) cylindrical outer shell with a strong lid section and a concentric centre pipe/conductor guide, which extends as deep as the CAN™ skirt [Sivertsen & Strand, 2011].

According to [NeoDrill, 2014] a typical CAN™ weight will be about 60-80 tons, with a diameter from 5 to 6 meters and a height from 8 to 12 meters, giving a soil penetration capacity from 10 to 11 meters.

The CAN™ will be pre-installed by a fit vessel that need to have a "Dynamic Positioning" system and a crane that is heave compensated. The same vessel may also be used for the conductor installation. The conductor installation gets shorten by that the needed joints are reduced from six (or more) to three (or two) and they can be assembled (welded) onshore, [Sivertsen & Strand, 2011].

4.3 Comparison with conventional drilling method

According to [NeoDrill, 2014] the CAN™ foundation provides:

- Less rig time: this means a cost efficient solution compared to conventional drilling methods
- Extended well fatigue life: mitigation of risk regarded to fatigue problems because of the bending moment getting transferred down to the stiff CAN™ instead of hitting the conductor or wellhead connector
- Proven technology: used on fields on the NCS
- Increased axial and lateral load capacity: because of the big dimensions of the CAN™
- Increased bending, fatigue and accidental load capacity
- "Fast track" field development: accelerated production enabled i.e. earn money faster
- Reduced environmental footprint: smaller vessels in addition to reduced cuttings and cement disposal
- HSE – improvement: less manual handling of heavy equipment
- Overall risk mitigation: according to ALARP

4.4 Typical CAN/BOP force distribution

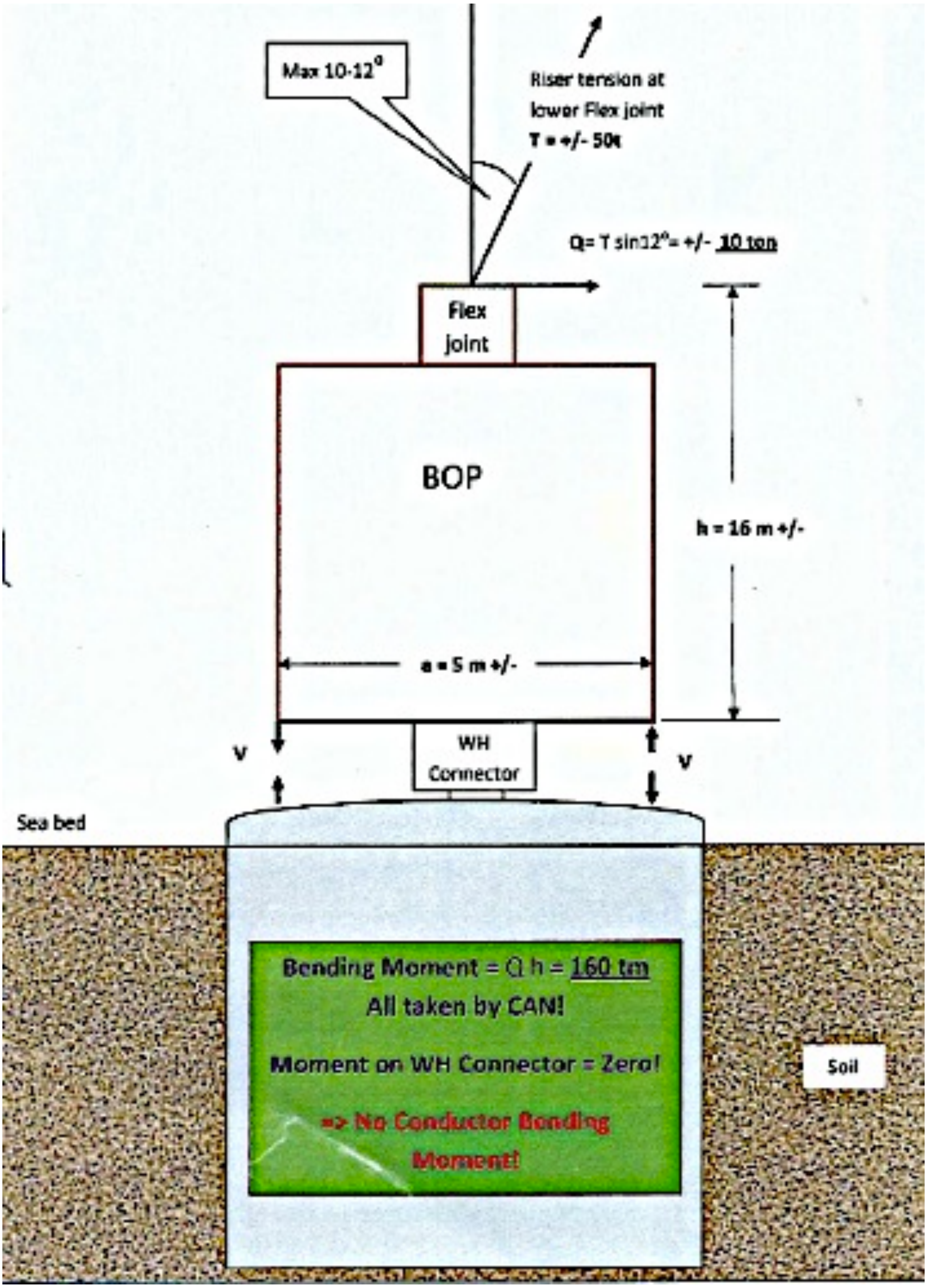


Figure 4-2: Typical CAN/BOP force distribution [Nergaard, 2014]

5 Background Information For Analysis

5.1 General

To do calculations and analysis on marine riser forces and the wellhead connector some background information need to be understood and explained.

The riser analysis will be done in an analysis program called OrcaFlex. The results in OrcaFlex will give the moment and the shear force (calculated from the riser tension) in the lower flex joint. Simplified mechanical models will be used to investigate the forces in the wellhead connector and the computer software Mathcad will be used for these calculations. It will be preformed a top-down analysis that will start at the drilling rig and end up with obtaining the forces in the wellhead/wellhead connector based on the results obtained from the model in OrcaFlex and the mechanical models developed in chapter **5.6 page 24**.

To understand the global analysis it will in this chapter be given some background information for marine riser mechanics, wellhead boundary conditions and simple beam theory to be used for calculations.

The mechanical models and boundary conditions for obtaining the forces in the wellhead connector will be presented after the background information given from chapter **5.2** to chapter **5.4**.

5.2 Riser mechanics: effective tension

The influence of tension, pressure and weight on pipe and risers is widely discussed and a misunderstanding of the subject has led to expensive mistakes in the past. The effective tension equation can be derived different ways. In this chapter Sparks, C.P method will be presented. This equation is the same as the equation used for calculating effective tension in OrcaFlex.

Sparks, C.P, 2007, calculates the effective tension as:

$$T_e = T_{tw} - p_i A_i + p_e A_e \qquad (5.1)$$

$where$

$T_{tw} = wall\ tension$

$p_i, A_i = internal\ pressure\ and\ internal\ cross\ sectional\ stress\ area$

$p_e, A_e = external\ pressure\ and\ external\ cross\ sectional\ stress\ area$

At any point in the riser the effective tension can be obtained by considering the top tension and the apparent weight of the intervening riser segment, see figure 5-1 below.

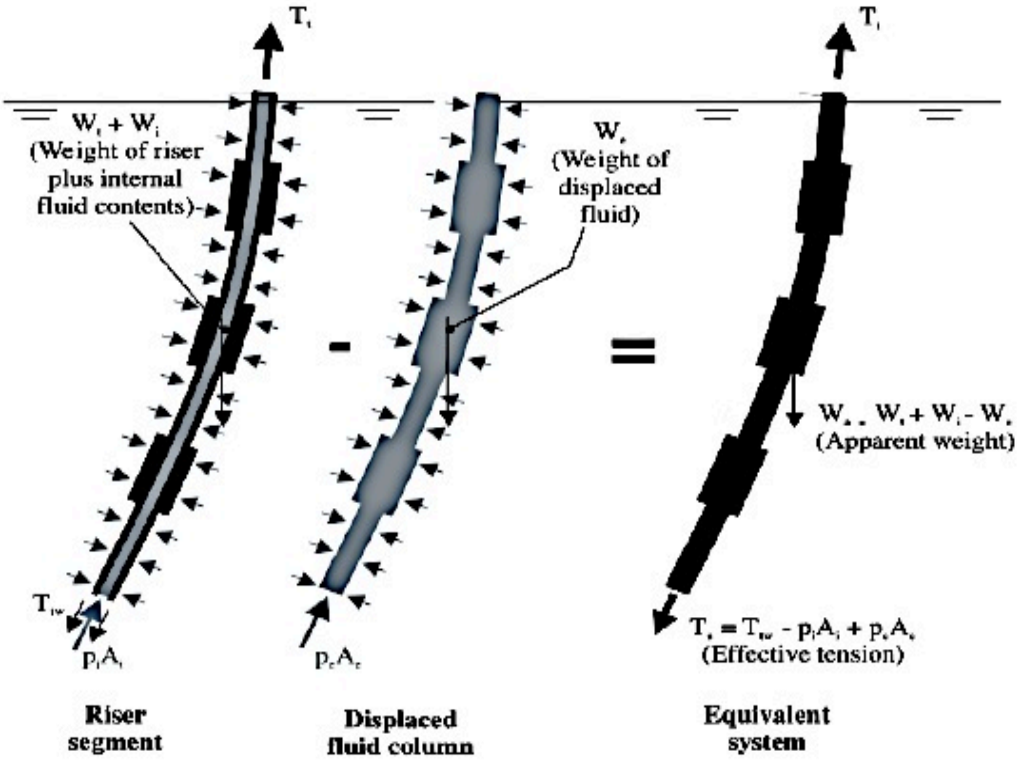


Figure 5-1: Forces and pressures acting on a long riser segment (Sparks, 2007)

5.3 Riser mechanics: stress

Combination of stresses in the riser cause yielding and a limit stress criteria needs to be decided for a riser analysis. Most codes require the Von Mises stress failure criterion to be checked. This is considered to be the most accurate criterion for ductile materials (Sparks, 2007).

For the general cases of triaxial stresses the Von Mises' equivalent stress, σ_{vm}, is given by (Sparks, 2007):

$$2\sigma_{vm}^2 = (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 + 6(\tau_{12}^2 + \tau_{23}^2 + \tau_{31}^2) \qquad (5.2)$$

where

$\sigma_1, \sigma_2, \sigma_3 = axial\ stresses\ in\ the\ three\ directions\ (x, y, z)$

$\tau_{12}, \tau_{23}, \tau_{31} = shear\ stresses$

Yielding will occur when the equivalent Von Mises' stress equals the yield stress of the material.

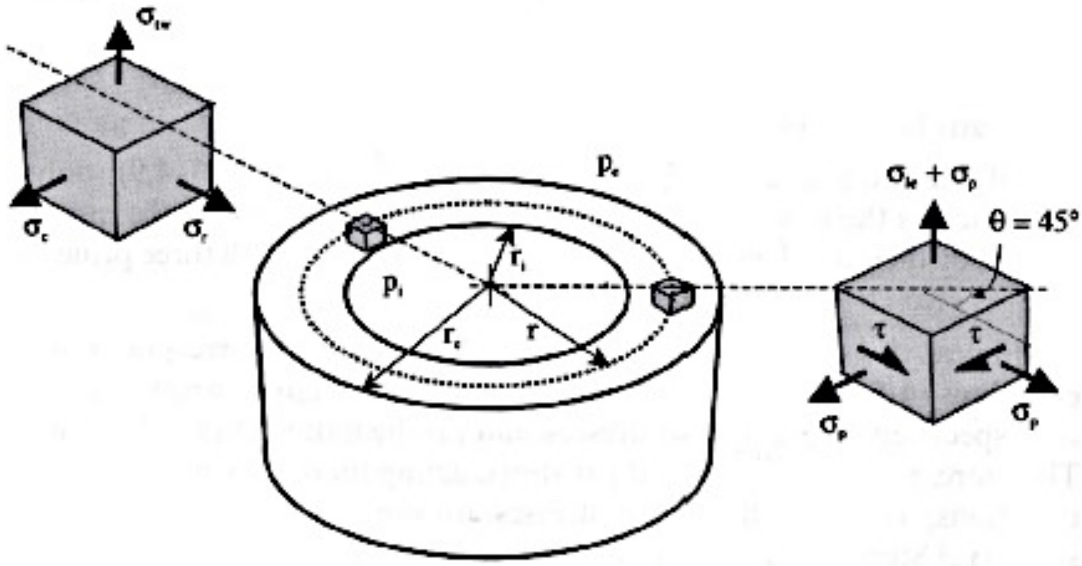


Figure 5-2: Pipe in-wall stresses (Sparks, 2007)

If equation (5.2) is applied to the principal stresses in the left stress cube in figure 5-2 (shear stresses are zero) the equation reduces to (Sparks, 2007):

$$2\sigma_{vm}^2 = (\sigma_{tw} - \sigma_c)^2 + (\sigma_c - \sigma_r)^2 + (\sigma_r - \sigma_{tw})^2 \qquad (5.3)$$

where

$\sigma_{tw} = axial\ stress$

$\sigma_c = circumferential\ stress$

$\sigma_r = radial\ stress$

5.4 Riser mechanics: strain

Axial strains are important to different riser problem and correct calculations are vital. For nearly vertical risers, the axial strains influence the required stroke of the tensioners. When the riser does not have tensioners, the strains between adjacent risers could affect riser performance. If the stability of drilling riser kill and choke lines were to be analysed, the axial strains would need to be considered (Sparks, 2007).

The principal strains are related to the principal stresses by the Young's modulus E and Poisson's ratio ν, relationship for an elastic isotropic pipe (anisotropic pipes is not considered in this project). The axial strain, ε_a, is given by (Sparks, 2007):

$$\varepsilon_a = \frac{1}{E}(\sigma_{tw} - \nu\sigma_c - \nu\sigma_r) \qquad (5.4)$$

$where$

$\sigma_{tw} = axial\ stress$

$\sigma_c = circumferential\ stress$

$\sigma_r = radial\ stress$

$E = Young'modulus$

$\nu = Poisson's\ ratio$

5.5 Wellhead boundary conditions

Normally a local response model needs to be developed for validation when mechanical models are used to evaluate lateral support and stiffness, load, displacement and rotation at the wellhead-datum. In this report the forces in the flex joint from the OrcaFlex model will be used to obtain displacement and rotation curves as a function of moment and shear forces at the wellhead datum as the focus is a global analysis of wellhead forces. In [Reinås, 2012] it is shown that wellhead and conductor housing behaves as a composite beam.

It would require more detail information about several parameters to make a local response model before the global analysis; such resources are not available to the writer of this project. From the example model in OrcaFlex and from the developed mechanical models it should be possible to come up with results for comparisons.

As explained in the beginning of this chapter the OrcaFlex model will be used for obtaining the forces in the flex joint area.

The moment will be transferred down to the wellhead datum from the flex joint by the following formula:

$$M_{WHdatum} = M_{flex\ joint} + (F_{shear} \times H_{BOP}) \qquad (5.5)$$

$where$

$M_{flex\ joint} = Bending\ moment\ at\ flex\ joint$

$H_{BOP} = Height\ of\ BOP$

The shear force, F_{shear} is obtained by taking the riser effective tension at the flex joint and multiply it with sine to the flex joint angle of rotation with the following formula:

$$F_{shear} = F_{effective\ tension} \times \sin\varphi \qquad (5.6)$$

$where$

$\varphi = flex\ joint\ rotation\ angle$

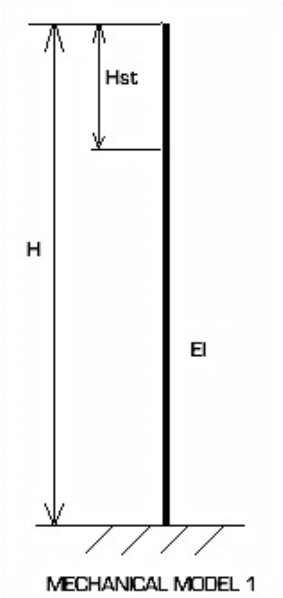
$F_{effective\ tension} = the\ effective\ tension\ in\ the\ riser\ at\ flex\ joint\ location$

5.6 Development Of Mechanical Models

The soil investigation to obtain values for different soil types is a comprehensive topic and not affordable in the time perspective of this project. In these sections three different simplified models for modelling wellhead stiffness and lateral displacement will be presented.

In this chapter the following mechanical models are developed:

Mechanical model 1:



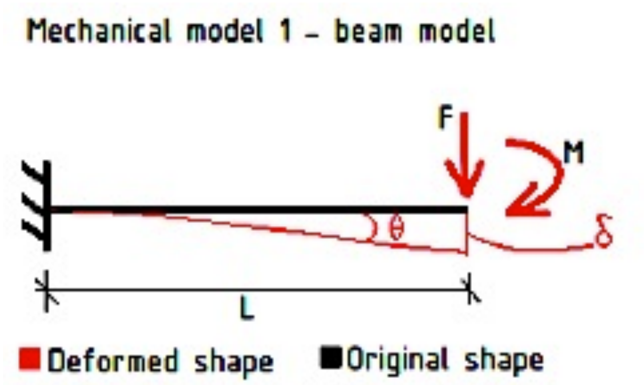


Figure 5-3: Mechanical model 1

- Wellhead datum at top of beam
- Fixed support
- Beam stiffness, EI
- Beam length, H
- “Stickup” height, H*st*

This model is made for comparison with the additional deflection when using the spring stiffness in the other two models.

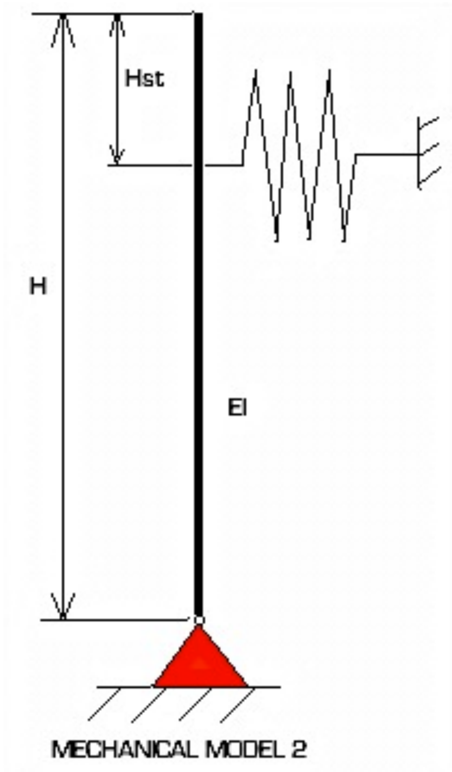
Formulas for rotation, θ and deflection, δ:

Only with shear force: Only with moment:

$$\theta = \frac{FL^2}{2EI} \qquad \delta = \frac{FL^3}{3EI} \qquad \theta = \frac{ML}{EI} \qquad \delta = \frac{ML^2}{2EI} \tag{5.7}$$

For combined load you simply add either the two rotations or deflections.

Mechanical model 2 (Hørte, 2011):



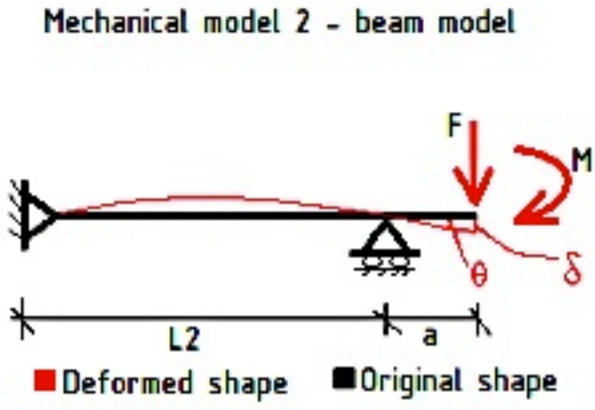


Figure 5-4: Mechanical model 2 [Hørte, 2011]

- Wellhead datum at top of beam
- Pinned support
- Beam stiffness, EI
- Beam length, H
- Non-linear spring to represent lateral stiffness
- "Stickup" height, H*st*

This model represents the lateral support of the wellhead by a spring and the pinned end support allows the end to rotate about x- and y-axis. This problem can be calculated statically with replacing the spring with a roller support. After the forces are obtained the additional deflection and rotation because of the spring can be obtained.

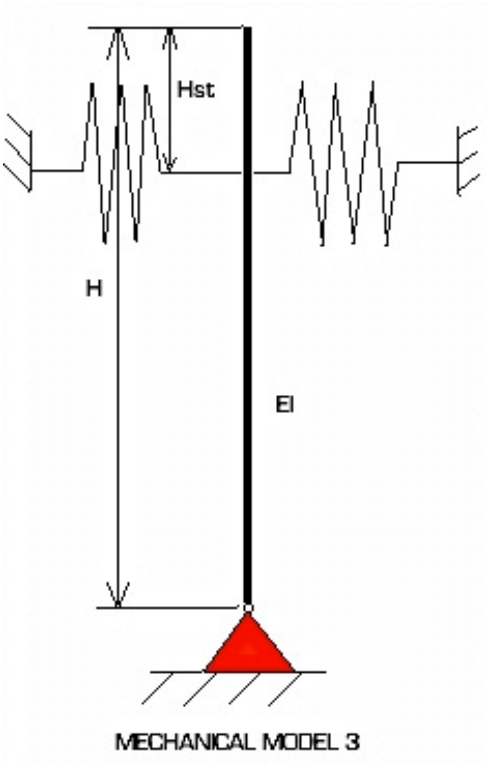
Formulas for rotation, θ and deflection, δ:

Only with shear force: Only with moment:

$$\theta = \frac{F(L2)a}{3EI} + \frac{Fa^2}{2EI} \quad \delta = \frac{Fa^2(L2)}{3EI} + \frac{Fa^3}{3EI} \quad \theta = \frac{M(L2)}{3EI} + \frac{Ma}{EI} \quad \delta = \frac{Ma(L2)}{3EI} + \frac{Ma^2}{2EI} \quad (5.8)$$

For combined load you simply add either the two rotations or deflections.

Mechanical model 3:



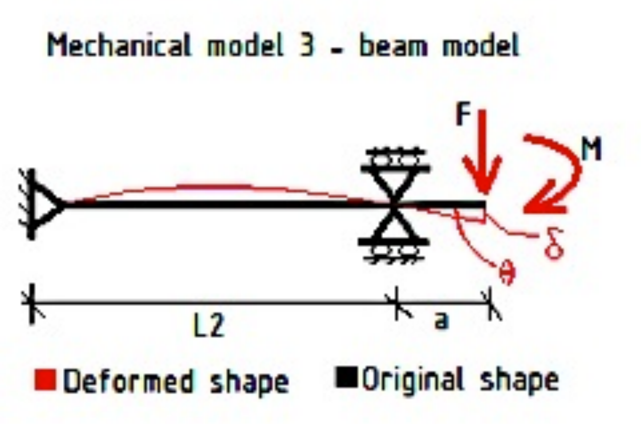


Figure 5-5: Mechanical model 3

- Wellhead datum at top of beam
- Pinned support
- Beam stiffness, EI
- Beam length, H
- Two non-linear springs to represent lateral stiffness when using the CAN developed by NeoDrill
- "Stickup" height, H*st*

Model 3 can be calculated the same way as model 2. The only difference is that it exits a higher stiffness of the spring (i.e. smaller deformation at roller support) because of the CAN. Investigation of forces when using the CAN is interesting for comparison.

Formulas for rotation, θ and deflection, δ:

Only with shear force: Only with moment:

$$\theta = \frac{F(L2)a}{3EI} + \frac{Fa^2}{2EI} \quad \delta = \frac{Fa^2(L2)}{3EI} + \frac{Fa^3}{3EI} \quad \theta = \frac{M(L2)}{3EI} + \frac{Ma}{EI} \quad \delta = \frac{Ma(L2)}{3EI} + \frac{Ma^2}{2EI} \quad (5.9)$$

For combined load you simply add either the two rotations or deflections.

For the two last presented models, model 2 and 3, the spring will cause further deformation and rotation of the wellhead datum. To find the result deformation and rotation the reaction force at the roller support needs to be divided by the stiffness of the spring, before further transformation to the wellhead datum. For model 3 there are to springs to model the extra stiffness from the CAN.

The method of doing this is the following:

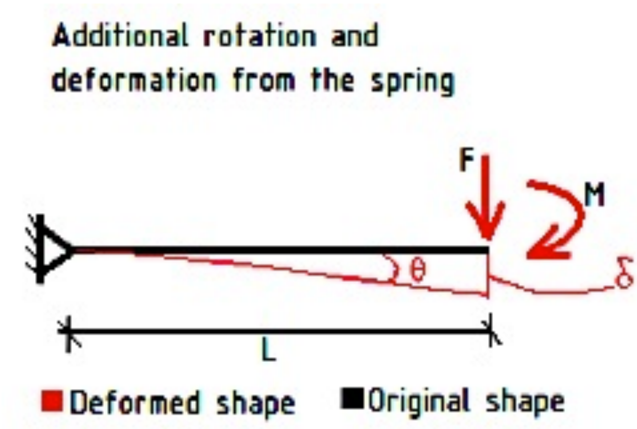


Figure 5-6: Additional deformation and rotation from spring

Reaction force: Deformation:

$$R_{roller} = \frac{P(a+(L2))}{L2} + \frac{M}{L2} \qquad \delta_{spring} = \frac{R_{roller}}{K_{spring}} \qquad (5.10)$$

Transferred to wellhead datum:

$$\delta_{springWH} = \delta_{spring}\frac{((L2)+a)}{L2} \qquad (5.11)$$

Further it is used simple Pythagoras to obtain the additional rotation:

$$\theta_{spring} = \cos(\delta_{spring}) \qquad (5.12)$$

Total rotation and deflection of the wellhead datum becomes:

Total rotation: Total deformation:

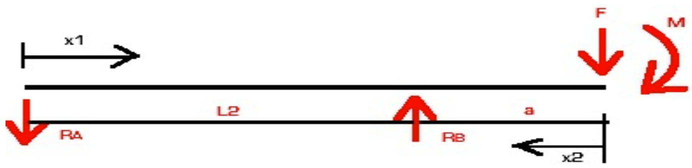
$$\theta_{TOTAL} = \theta_{spring} + \theta \qquad \delta_{TOTAL} = \delta_{springWH} + \delta \qquad (5.13)$$

5.7 Beam Theory

The mechanical models in chapter 5.6 are modelled as simple beam and spring systems with the use of elastic beam theory.

The models are obtained by using the *method of superposition*. This is method is a practical and simple way of obtained deflection and angle of rotation of beams. The way to do it is to calculate deflection and angle of rotation for each loading (shear, moment or uniform load) separately and then add them together to obtain the combined loading.

The calculation of mechanical model 2 will be shown as an example for this project:



$$+\downarrow \sum M_B = -R_A * (L2) - P * a - M$$

$$R_A = -\frac{P * a}{(L2)} - \frac{M}{(L2)} \quad (downward)$$

$$+\downarrow \sum M_A = R_B * (L2) - P * \big(a + (L2)\big) - M$$

$$R_B = \frac{P * \big(a + (L2)\big)}{(L2)} + \frac{M}{(L2)} \quad (upward)$$

$$M_{AB} = R_A * x_1 = -\frac{P * a}{(L2)} x_1 - \frac{M}{(L2)} x_1 \quad 0 \le x_1 \le L2$$

$$M_{BC} = -Px_2 - M \quad\quad 0 \le x_2 \le a$$

Integrate moments and multiply by moment partial derivative of $M_{AB}\ and\ M_{BC}$for $x_1\ and\ x_2$ and add them together to obtain the angle of rotation:

$$\theta_C = \int_0^{L2} \left(\frac{M_{AB}}{EI}\right)\left(\frac{\partial M_{AB}}{\partial M}\right) dx_1 + \int_0^{a} \left(\frac{M_{BC}}{EI}\right)\left(\frac{\partial M_{BC}}{\partial M}\right) dx_2$$

$$\theta_C = \frac{Pa(L2)}{3EI} + \frac{ML}{3EI} + \frac{Pa^2}{2EI} + \frac{Ma}{EI} \quad (5.14)$$

Integrate moments and multiply by shear force partial derivate of $M_{AB}\ and\ M_{BC}$for $x_1\ and\ x_2$ and add them together to obtain the deflection:

$$\delta_C = \int_0^{L2} \left(\frac{M_{AB}}{EI}\right)\left(\frac{\partial M_{AB}}{\partial P}\right) dx_1 + \int_0^{a} \left(\frac{M_{BC}}{EI}\right)\left(\frac{\partial M_{BC}}{\partial P}\right) dx_2$$

$$\delta_C = \frac{Pa^2(L2)}{3EI} + \frac{MaL}{3EI} + \frac{Pa^3}{3EI} + \frac{Ma^2}{2EI} \quad (5.15)$$

6 Analysis Values And Parameters

6.1 General

The analysis values for vessel motions, marine riser properties and environmental data in this project are taken from relevant standards and recommended practices, as the writer doesn't have any particular area for the analysis. OrcaFlex implements some of the standards and this will make it easier to obtain good results from the OrcaFlex model. In chapter 7 the model build-up in OrcaFlex is explained in more detail.

6.2 Material Properties

Steel data obtained from DNV report on wellhead fatigue [Hørte, 2011]:

E (Young's modulus)	ν (Poisson's ratio)	ρ (density)
210 GPa	0.3	7850 kg/m^3

Table 6-1: Material properties for steel

6.3 Vessel motions - RAOs

The motion of the vessel is important in predicting the expected riser response. Not only the magnitude but also it's phasing with respect to the wave. Response Amplitude Operator (RAO) values for this analysis are given in "Appendix B: RAO Data", page number xxi.

An 8-column semi-submersible with a 24,4 m draught is modelled in OrcaFlex.

6.4 Environmental data

The environmental data with sea states and currents are usually obtained from metocean data from the specific area of operation. This project is not deducted for a specific area and therefore the environmental data is obtained from recommendations from standards and recommended practices.

6.4.1 Water information

Parameter	Value
Water density	1025 kg/m^3
Water depth	1000 m
Sea temperature	10 °C

Table 6-2: Water information

6.4.2 Waves

The significant wave height is defined as the average of the highest 1/3 waves in the indicated time period. For this analysis the JONSWAP spectrum is used in the OrcaFlex model that is a wave model representing irregular waves. This project is not executed for a specific location so the significant wave height and period values are taken from OrcaFlex and verified to be in the reasonable JONSWAP model. Normally the wind and wave condition would be obtained from metocean data for the specific location; previously measured wind and wave data from the drilling site can also be used. The more information available, the more accurate the predicted climatology will be.

NORSOK N-003 is a Norwegian standard and a simplified approach for obtaining reasonable waves using the relevant design wave height H_{100} is presented. The H_{100} corresponds to a wave with annual probability excess of 10^{-2} (the 100-year wave) and it may be taken as 1.9 times the significant wave height H_s. The H_s should then be obtained from long-run statistics when the sea-state duration is 3 hours. For the OrcaFlex simulation the simulation time is 1000 s and a simulation time of 3 hours is not affordable in this project but need to be applied for real-life wave estimations. The wave period to be used together with the H_s and the design wave H_{100} are suggested to be in the range:

$$\sqrt{6.5H_{100}} \leq T_p \leq \sqrt{11H_{100}}$$

The Pierson-Moskowitz (PM) is a wave spectrum that originally was proposed for a fully developed sea. The JONSWAP spectrum is an extension of the PM spectrum that in addition to the fully developed sea includes fetch-limited seas, describing developing sea states. A wave spectrum is simply the power spectral density function of the vertical sea surface displacement [DNV-RP-C205].

The JONSWAP spectrum is expected to be a reasonable model for

$$3.6 < \frac{T_p}{\sqrt{H_s}} < 5$$

Where T_p is seconds and H_s is in meters. The effect of the peak shape parameter, γ (non-dimensional) for H_s = 4.0 m and T_p = 8.0 s is shown in figure 6-1 below.

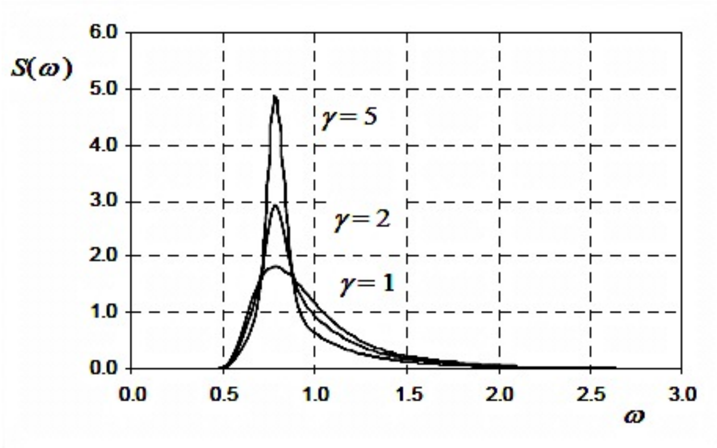


Figure 6-1: JONSWAP spectrum [DNV-RP-C205]

When setting the parameters in OrcaFlex it exists an automatic functions were the H_s and T_z are specified, then the program calculates the rest of the parameters.

Sea states used in this analysis are both within the reasonable JONSWAP model and reasonable for connected operability analysis. The sea states are also within the recommended period using the design wave, H_{100}.

Sea State	H_s (m)	T_p (s)	T_z (s)
1	7	11.2	9
2	10	11.7	9.5

Table 6-3: Significant wave height and corresponding peak and zero-crossing periods

6.4.3 Current

The current profile is randomly selected in OrcaFlex. As an example it could range from 1.0 m/s at the sea surface and decrease to approximately 0 m/s for a depth of 1000m. In a real situation the current should be considered in detail as it can cause slow drift motions to moored platforms, give rise to drag and lift forces on submerges structures, lead to VIV (vortex induced vibrations) of slender and large volume structures and have an impact on the waves that could lead to change in wave height and wave periods [DNV-RP-205].

6.4.4 Wind

Wind is not included in the analyses because of the fact that the wind magnitude is assumed to be negligible compared to the sea environment when analysis of what happens at 1000 m below mean sea level is the main task for this project.

6.5 Marine riser properties

For a real riser analysis all riser component joint weights and dimensions would be provided for the analyst. A riser configuration from top to bottom would also be provided. A typical riser configuration (Ormen Lange project) is shown in table 6-2.

No.	Description
1	Upper flex joint
2	Telescopic joint
3	Keel transition joint
4	Intermediate flex joint
5	Termination spool and split ring
6	Pup joints, as needed
7	Buoyed joints, as needed
8	Bare joints, as needed
9	LMRP + Lower flex joint
10	BOP
11	Tree (If used)
12	Wellhead
13	Foundation – template structure

Table 6-4: Riser configuration overview for the Ormen Lange project (SES, 2006)

In this project the riser used in OrcaFlex are modelled as a 24" drilling riser consisting of 204 segments with a length of 5 m.

6.6 BOP and wellhead connector

The BOP and Wellhead dimensions are obtained from a previous thesis [Harildstad, E. & Haukanes, A., 2013] written at NTNU. The properties used in this thesis are obtained from Statoil for both the BOP and the wellhead connector (high pressure wellhead connector).

6.6.1 BOP properties

The BOP is calculated as infinitely stiff and is also modelled like this in OrcaFlex.

BOP parameter	Value
Mass (dry weight)	190×10^3 kg
Mass (in water)	162×10^3 kg
Height	12.4 m

Table 6-5: BOP parameters

6.6.2 Wellhead connector and properties (conventional drilling)

WH connector parameter	Value
Stickup height, H_{st}	1.5 m
Length, H	4.6 m
Bending stiffness, EI	1.4 x 10^6 kNm^2
Stiffness lateral spring, K	35 x 10^3 kN/m
Position lateral spring	1 m below wellhead datum
ID (inner diameter)	18 ¾" = 0,476 m
OD (outer diameter)	26.8" = 0,689 m

Table 6-6: Wellhead connector parameters

6.7 CAN stiffness

The stiffness of the CAN is non-linear i.e. it is usually represented by P-Y curves. P-Y curves are a relationship between the forces applied to soil to the lateral deflection of the soil, so the curves will vary with the soil type.

The springs can be represented by the equation:

$$p = ky$$

$$where$$

$$k = non-linear\ spring\ stiffness\ defined\ by\ P-Y\ curve$$

$$y = deflection\ of\ the\ spring$$

$$p = force\ applied\ to\ the\ spring$$

There is no particular soil type for this project. But using the linear models developed it can be assumed that the stiffness of the CAN "springs" are significantly higher than the one for the wellhead during conventional drilling.

6.8 Rules and standards

In the oil and gas industry it exists numerous standards. In Norway it is used NORSOK standards, in America it is used API standards, internationally it is used ISO standards and it also exists recommended practices and standards from DNV. It is impossible to follow every standard and the most important thing is to follow the standard applicable for the company you work for and the country you work in.

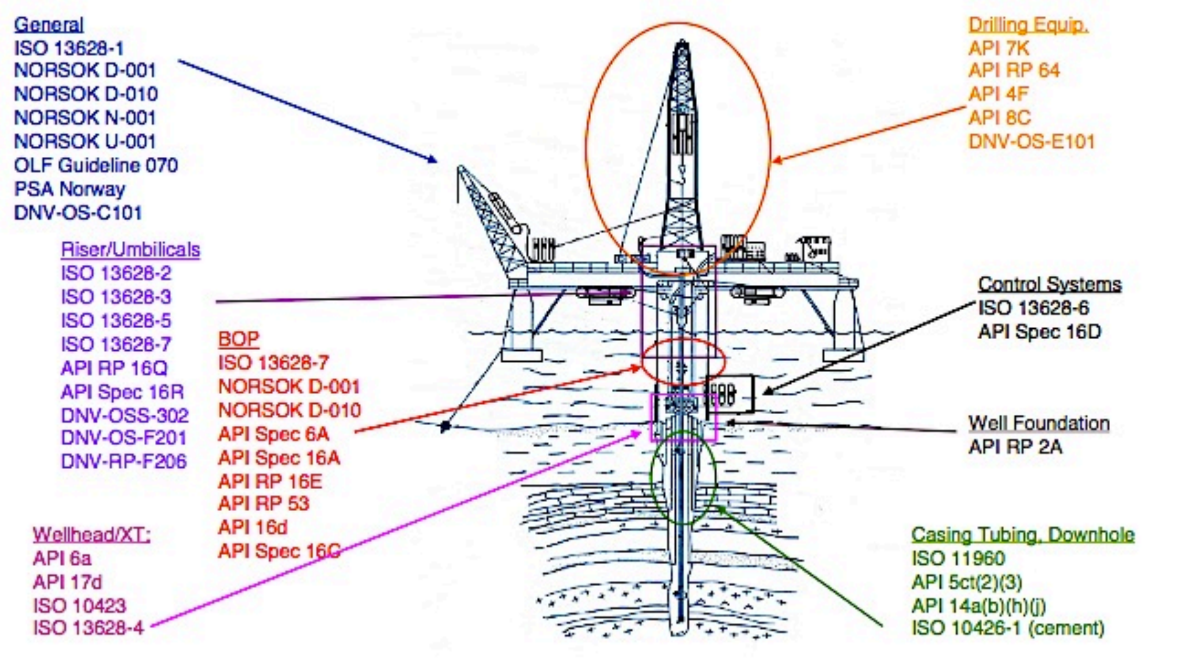


Figure 6-2: Relevant codes for drilling and well systems (Reinås, 2012)

For Marine drilling risers there are three design codes that is relevant:

1. ISO 13624-1 Design and operation of marine drilling riser equipment (based on API RP 16Q).

Design parameter	Riser connected		Riser disconnected
	Drilling	Non-drilling	
Mean upper flex/ball jt. angle [a]	1° to 1,5°	N/A	N/A
Max. upper flex/ball jt. angle [a]	5,0°	90 % available (or contact with moonpool structure)	90 % available (or contact with moonpool structure)
Mean lower flex jt. angle [b]	2,0°	N/A	N/A
Max. lower flex jt. angle [b]	5,0°	90 % available	N/A
Stress criteria: [c,d]			
– Method "A" - allowable stress [e]	$0,40\ \sigma_y$ [f]	$0,67\ \sigma_y$ [f]	$0,67\ \sigma_y$ [f]
– Method "B" - allowable stress [e]	$0,67\ \sigma_y$ [f]	$0,67\ \sigma_y$ [f]	$0,67\ \sigma_y$ [f]
– Sign. dyn. stress range [e]			
@ SAF $\leqslant$ 1,5 [g]	69 MPa (10 ksi)	N/A	N/A
@ SAF > 1,5 [g]	$15/F_{SA}$	N/A	N/A
Minimum top tension [h]	T_{min}	T_{min}	N/A
Dynamic tension limit [i]	F_{DTL}	F_{DTL}	N/A
Maximum tension setting	90 % F_{DTL}	90 % F_{DTL}	N/A

Figure 6-3: ISO13624-1: Maximum design guidelines

From figure 6-3 above it is given that the max lower flex joint angle are 5 degrees while drilling and 90% of available when non-drilling i.e. with a 10 degrees available flex joint rotation the maximum allowable angle would be 9 degrees.

2. ISO 13624-2 Deepwater drilling methodologies, operations and integrity technical report.

3. API RP 16Q Recommended Practice for Design, Selection, Operation and Maintenance of Marine Drilling Riser Systems.

For wellhead design there are two main design codes:

1. ISO 10423:2009 Petroleum and natural gas industries – Drilling and production equipment – Wellhead and christmas tree equipment
2. API Spec 6A Specification for Wellhead and Christmas tree equipment

7 Model BOP And Riser For Load Cases

7.1 OrcaFlex introduction

This chapter is in large extent written according to the OrcaFlex user manual developed by Orcina, which is an engineering software and consultancy company located in Cumbria, United Kingdom. Their homepage for other analysis software is: http://www.orcina.com.

OrcaFlex is a dynamic analysis programme used for offshore marine systems. It is user friendly and has technical breadth. The static and dynamic analysis extend to a large range of systems, including:

- All types of marine risers (rigid and flexible)
- Global analysis
- Moorings
- Installation
- Towed systems

In this project the programme will be used for analysing a tensioned marine drilling riser descended from a semi-submersible drilling vessel to a BOP on the seabed. A drill string is modelled running inside the riser down to the BOP. The model is an example from the Orcina homepage and fits good to the model needed to obtain the forces in the lower flex joint for further analysis of the forces in the wellhead. This example will give a more accurate analysis than one modelled by the student writing this project as engineer employees with long experience with the programme made the example. The model is fully editable so it is possible to add new values for all input parameters and change the model to fit this project or other projects.

7.2 OrcaFlex theory

7.2.1 Coordinate system

OrcaFlex uses one global coordinate system GXYZ, where G is the global origin and GX, GY, GZ are the global axes directions. In addition, there are a number of local coordinate systems, generally one for each object in the model. All the coordinate systems are right-handed, as shown in figure 7-1, which shows the global axes and a vessel with its own local vessel axes Vxyz. Positive rotations are clockwise when looking in the direction of the axis rotation.

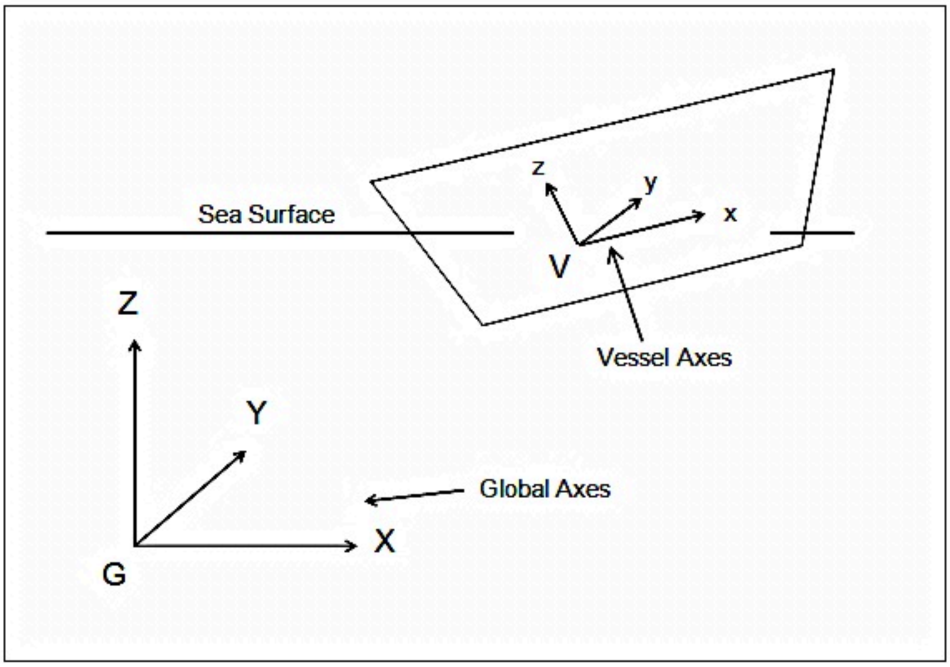


Figure 7-1: Orcaflex coordinate system (Orcina, 2014)

7.2.2 Static analysis

The static analysis has two main objectives. The first objective is to determine the equilibrium configuration of the system analysed under weight, buoyancy, hydrodynamic drag, etc. The second objective is to provide a starting configuration for dynamic simulation of the model. The static equilibrium configuration is usually the best starting point for dynamic simulation and these two objectives become one.

Static equilibrium is determined in a series of iterative stages:

1. At the start of the calculation, the initial positions of the vessel and buoys are defined by the data: these in turn define the initial positions of the ends of any lines connected to them.
2. The equilibrium definition of each line is then calculated; assume the line ends are fixed.
3. The out of balance load acting on each free body (node, buoy, etc.) is then calculated and a new position is estimated for the body. This process is repeated until the out of balance load on each free body is zero (up to the specified tolerance).

7.2.3 Dynamic analysis

The dynamic analysis is a time simulation of the motions of the model over a specific period of time, starting from the position derived from the static analysis. The period of simulation is defined as a number of consecutive stages, whose durations are specified in the data.

OrcaFlex implements two complementary dynamic integration schemes: explicit and implicit, as described below.

The equation of motion that OrcaFlex solves is as follows:

$$\boldsymbol{M}(\boldsymbol{p},\boldsymbol{a}) + \boldsymbol{C}(\boldsymbol{p},\boldsymbol{v}) + \boldsymbol{K}(\boldsymbol{p}) = \boldsymbol{F}(\boldsymbol{p},\boldsymbol{v},\boldsymbol{t})$$

$where$

$\boldsymbol{M}(\boldsymbol{p},\boldsymbol{a}) = system\ inertia\ load$

$\boldsymbol{C}(\boldsymbol{p},\boldsymbol{v}) = system\ damping\ load$

$\boldsymbol{K}(\boldsymbol{p}) = system\ stiffness\ load$

$\boldsymbol{F}(\boldsymbol{p},\boldsymbol{v},\boldsymbol{t}) = external\ load$

$\boldsymbol{P},\boldsymbol{v}\ and\ \boldsymbol{a} = position, velocity\ and\ acceleration\ vectors\ respectively$

$\boldsymbol{t} = simulation\ time$

The **explicit** integration is forward Euler integration with a constant time step. At the start of the time simulation, the initial positions and orientations of all objects in the model, including all nodes in all line, are known from the static analysis. The forces and moments acting on the free body and node are then calculated.

The equation of motion (Newton's law) is then formed for each free body and each line node:

$$\boldsymbol{M}(\boldsymbol{p},\boldsymbol{a}) = \boldsymbol{F}(\boldsymbol{p},\boldsymbol{v},\boldsymbol{t}) - \boldsymbol{C}(\boldsymbol{p},\boldsymbol{v}) - \boldsymbol{K}(\boldsymbol{p})$$

The equation is solved for the acceleration vector at the beginning of each time-step, for each free body and each line node. It is then integrated using forward Euler integration. At the end of each time step, the positions and orientations of all nodes and free bodies are again known and the process is repeated.

When **implicit** integration is used, OrcaFlex uses the Generalised-α integration scheme as described by Chung and Hulbert (book discussing the Generalised-α integration). The forces, moments, damping, mass etc. are calculated the same way as for the explicit scheme. Then the system equation of motion is solved at the end of each time step.

7.2.4 Line theory

OrcaFlex uses a finite element model for a line shown in figure 7-2. The line is divided into segments that are modelled by straight massless model segments with a node at each end. The segments only model the axial and torsional properties of the line. The other properties (mass, weight, buoyancy, etc.) are all lumped to the nodes, as indicated by the arrows in figure 7-2. Nodes and segments are numbered (1,2,3...) from end A of the line to end B. Segment n joins nodes n and (n+1).

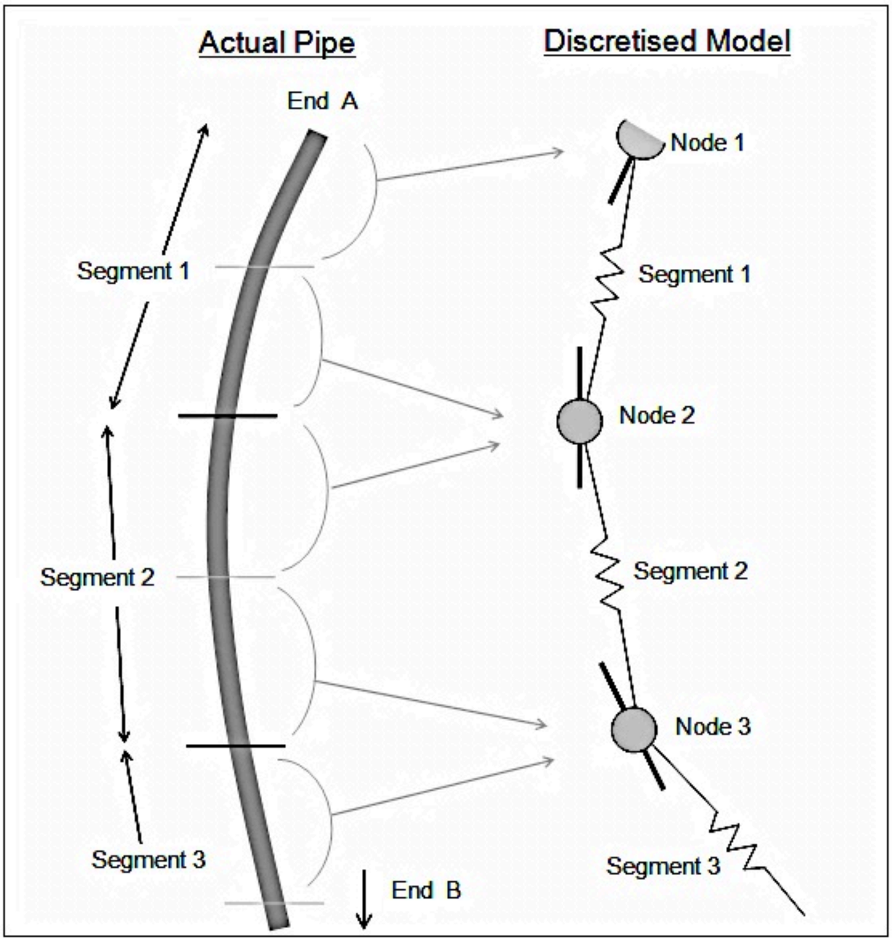


Figure 7-2: Orcaflex line model [Orcina, 2014]

OrcaFlex does the calculation on a mid-node (for example node – in figure 7-2) in 5 steps:

1. Tension forces.
2. Bending moments.
3. Shear forces.
4. Torsion moments.
5. Total load.

Figure 7-3 below shows a more detailed line model, including various spring + dampers that model the structural properties of the line. The figure also shows the xyz-directions of reference and the angles from node to segment.

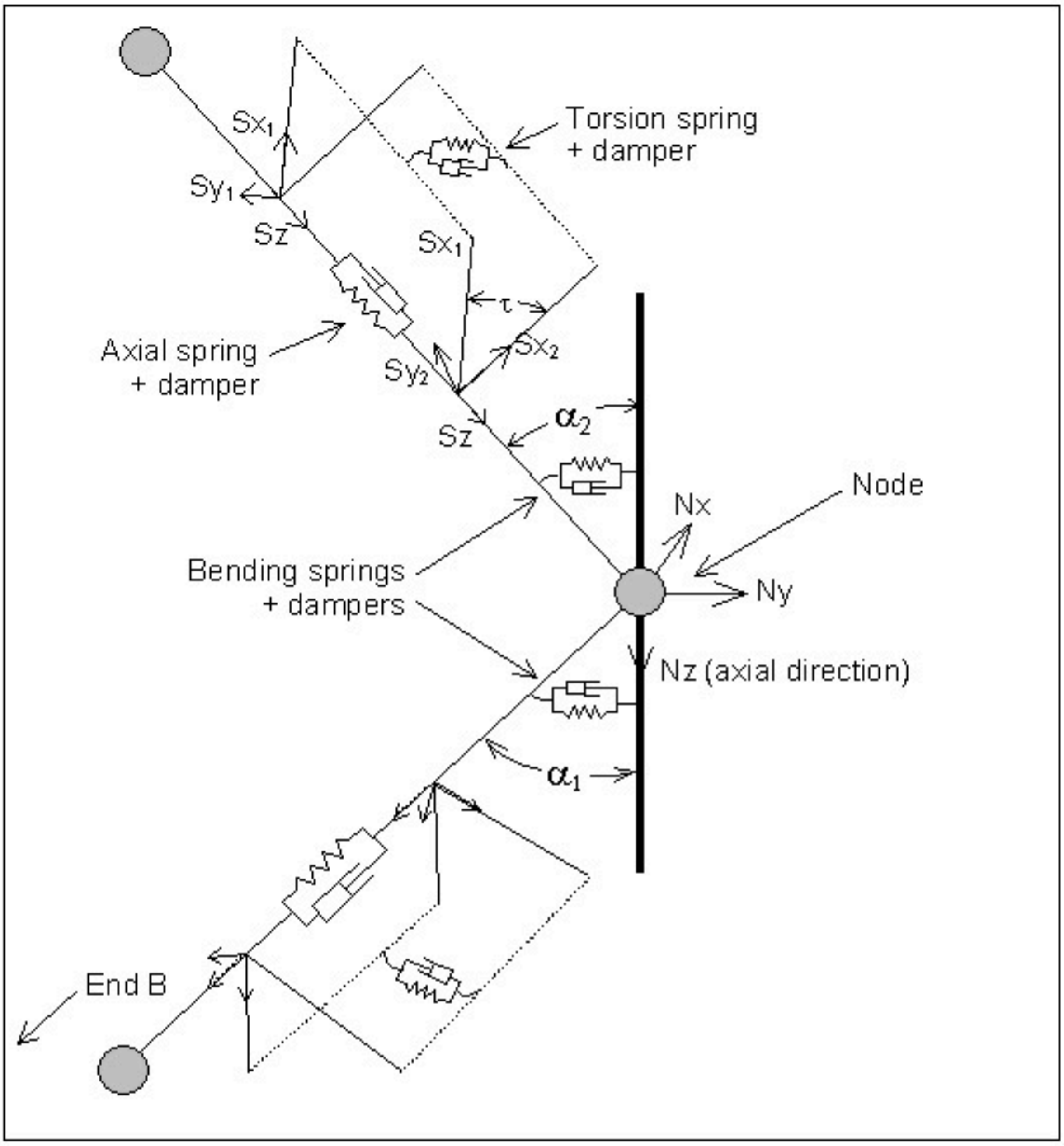


Figure 7-3: Orcaflex detailed line model [Orcina, 2014]

7.2.5 Directions conventions

In OrcaFlex the headings and directions are specified by the angle of direction, azimuth, measured from the x-axis towards the y-axis to get a positive measurement. The directions are shown in the figure below.

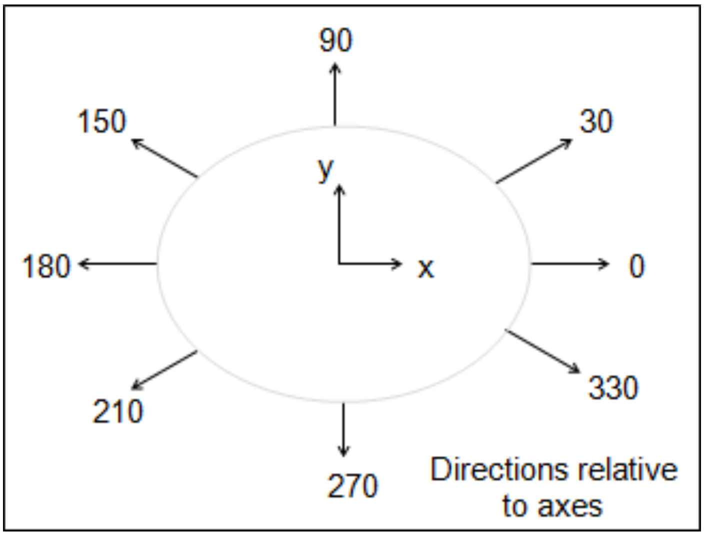


Figure 7-4: Directions and headings [Orcina, 2014]

When it comes to the vessel response to the wave it depend on the wave direction relative to the vessel. RAOs are therefore given as a wave direction relative to vessel axes. The x-axis in the figure above becomes the vessel heading direction. A relative wave direction of zero degrees means a wave coming from astern and a relative direction of 90 degrees means one coming from starboard.

7.2.6 Hydrodynamics

To obtain hydrodynamic loads on the various line, 3D- and 6D buoys OrcaFlex use an extended form of the Morrison's equation. This formula was originally made for calculating wave loads on fixed cylinders. The equation have two force components, one related to the water particle acceleration, called the inertia force, and one related to the water particle velocity, called the drag force.

The extended form of the Morison's equation is (with inertia force in the parentheses):

$$F_w = (\Delta * a_w + C_a * \Delta * \Delta a_r) + \frac{1}{2} * \rho * C_d * A * V_r * [V_r]$$

$where$

$F_w = the\ fluid\ force$

$\Delta = the\ mass\ of\ fluid\ dispaced\ by\ the\ body$

$a_w = the\ fluid\ acceleration\ relative\ to\ earth$

$C_a = added\ mass\ coefficient\ for\ the\ body$

$a_r = fluid\ acceleration\ relative\ to\ the\ body$

$\rho = density\ of\ water$

$V_r = fluid\ velocity\ relative\ to\ the\ body$

$A = drag\ area$

7.3 OrcaFlex model build-up

For the OrcaFlex analysis an example from the Orcina homepage will be used (example B01 Drilling Riser, Orcina, 2013) and modified to fit the purpose of this project.

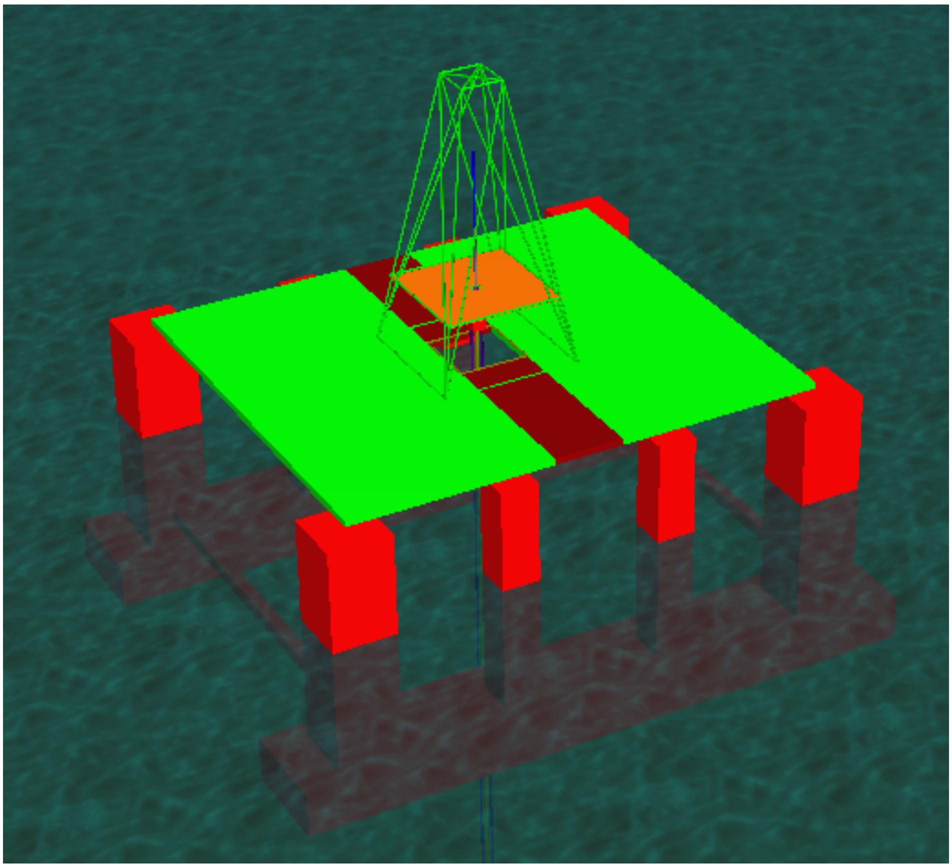


Figure 7-5: Semi-submersible modelled in OrcaFlex

The vessel is a semi-submersible drilling vessel. The vessel is connected to a BOP on the seabed via the drilling riser. A drill string is modelled running inside the riser down to the BOP and carries on into the casing below the seabed. The bottom of the BOP is located at 1020 meters below the semi-submersible.

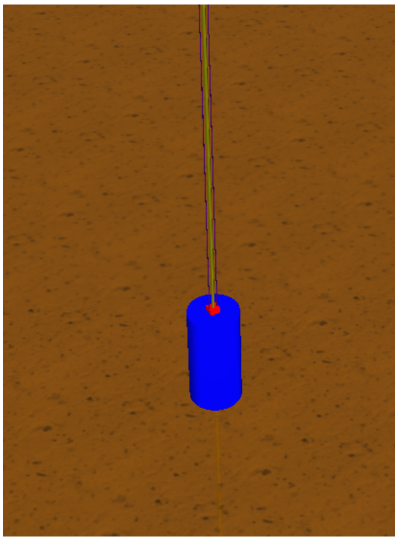


Figure 7-6: BOP (blue) connected to the flex joint (red)

The vessel is modelled in detail with upper and lower deck, rotary table and tensioners. The riser has a slip-joint at the tensioners to allow movement axially but not laterally. It is four tensioners in the model and they are modelled using four links of spring/damper type. Their stiffness is non-linear but their damping is linear with velocity.

The tensioners and the slip joint are connected to a tensioner ring. The tensioner ring is modelled as a 6D buoy and its only intention is to act as a connection point.

The capacity of the tensioners is modified (higher) to avoid compression (that could lead to buckling) in the riser. When the waves were changed from regular (originally modelled) to irregular (JONSWAP) it caused the effective tension in the riser to reach compression at a water depth of approximately 900m. The results for the effective tension are presented in "Appendix C: Orcaflex results", page number xxv. The tensioners have been kept under a capacity of 270 Mt per tensioner (which is the capacity of the riser tensioners on the drilling rig West Hercules) to be realistic.

In addition to the riser the model also have kill and choke lines. These are connected to the tensioner ring at the top and to a flex joint modelled at the top of the BOP. This ensures that the BOP sees appropriate total moments instead of individual ones.

The vessel was set up with a prescribed motion making it move with a constant speed in the Global X direction. Simulation time was 42 seconds. In this project there is no prescribed motion assuming the rig is anchored and neglecting drift-off. The simulation time is changed to 1000 s to get wider aspects of the movements and the forces in the different parts.

This model will be used for obtaining bending moment and shear force at the wellhead datum and use this for local analysis in the mechanical models.

8 Analysis Results For Wellhead Forces

8.1 General

The results will be represented in the following order:

1. Calculated shear force as a function of flex joint angle. This force is obtained from the formula (5.6) given in chapter "5.5 wellhead boundary conditions".
2. Calculated moment at wellhead datum as a function of shear force. This force is obtained from the formula (5.5) given in chapter "5.5 wellhead boundary conditions".
3. Bending moment at wellhead datum as a function of rotation and displacement. Obtained from mechanical models calculations shown in "Appendix A: Mathcad calculations".
4. Shear force at wellhead datum as a function of rotation and displacement. Obtained from mechanical model calculations shown in "Appendix A: Mathcad calculations".
5. Total displacement and rotation after adding the contribution from the spring.

Graphical results from OrcaFlex that is used for calculations can be found in "Appendix C: Orcaflex results", page number xxv. Calculations done in Mathcad is found in "Appendix A: Mathcad Calculations", page xii.

The shear force will be presented as a function of the maximum flex joint angle (obtained from the conditions implemented in OrcaFlex). The effective tension as a function of arc length is shown in figure C-1 in "Appendix C: Orcaflex results", page number xxv. All the results are from calculations done with mechanical model 2. Mechanical model 1 will be used for a comparison with the additional deformation calculation and mechanical model 3 is an illustration of the wellhead with the use of a CAN. The CAN will be discussed in chapter **8.4**, page 51.

8.2 Results sea state 1: $H_s = 7m$ and $T_p = 11.2s$

1. Shear force with maximum flex joint angle (OrcaFlex):

$$\boldsymbol{F_{shear}} = 2490\text{kN} * \sin(0.60) = \boldsymbol{26kN}$$

2. Moment at wellhead datum with maximum flex joint angle (OrcaFlex):

$$\boldsymbol{M_{WHdatum}} = 70\text{kNm} + (26\text{kN} * 12.4m) = \boldsymbol{392\ kNm}$$

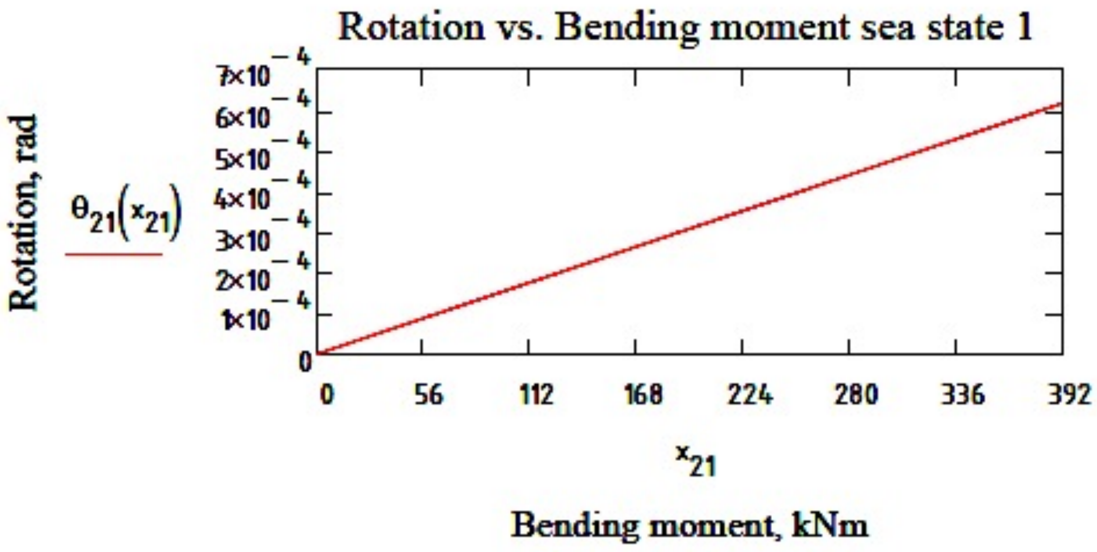
Description	OrcaFlex
Shear force, F_{shear}	26 kN
Bending moment at wellhead datum, $M_{WHdatum}$	392 kNm

Table 8-1: Result: shear force and bending moment at wellhead datum, Hs=7m

Comment on results: Because of the small flex joint angle (0.60°) from sea state 1 the results shows a small shear force and a small bending moment at wellhead datum.

By varying the rotation in the wellhead the effect on the moment at the wellhead datum can be shown for the different mechanical models.

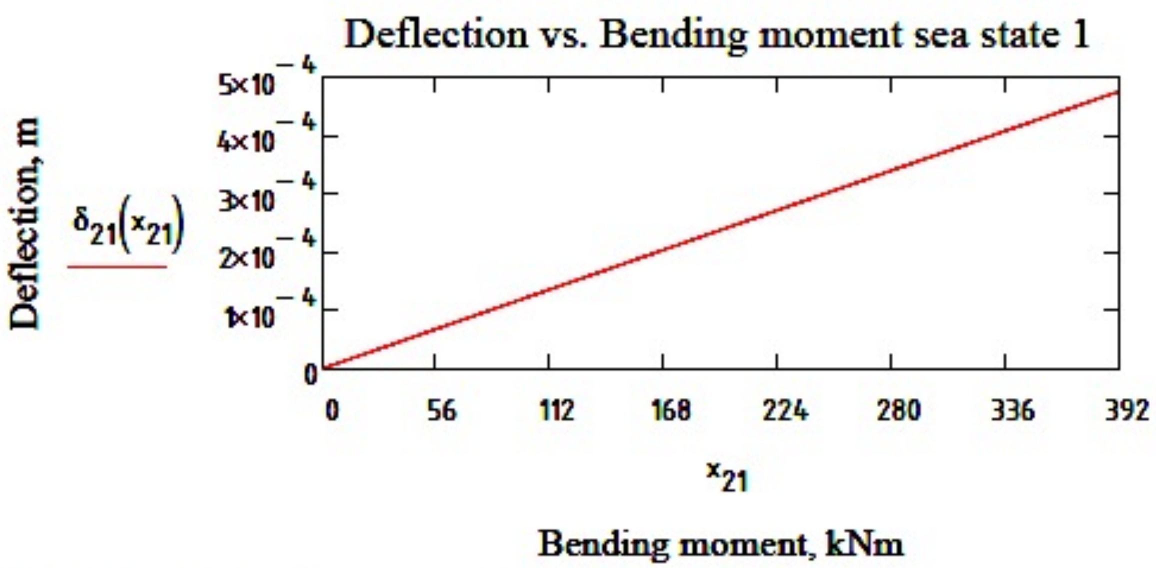
3. Bending moment at wellhead datum as a function of rotation and displacement:



Rotation with maximum moment from sea state 1:

$\theta_{21}(392) = 0.001$ rad

Figure 8-1: Results: rotation vs. bending moment sea state 1

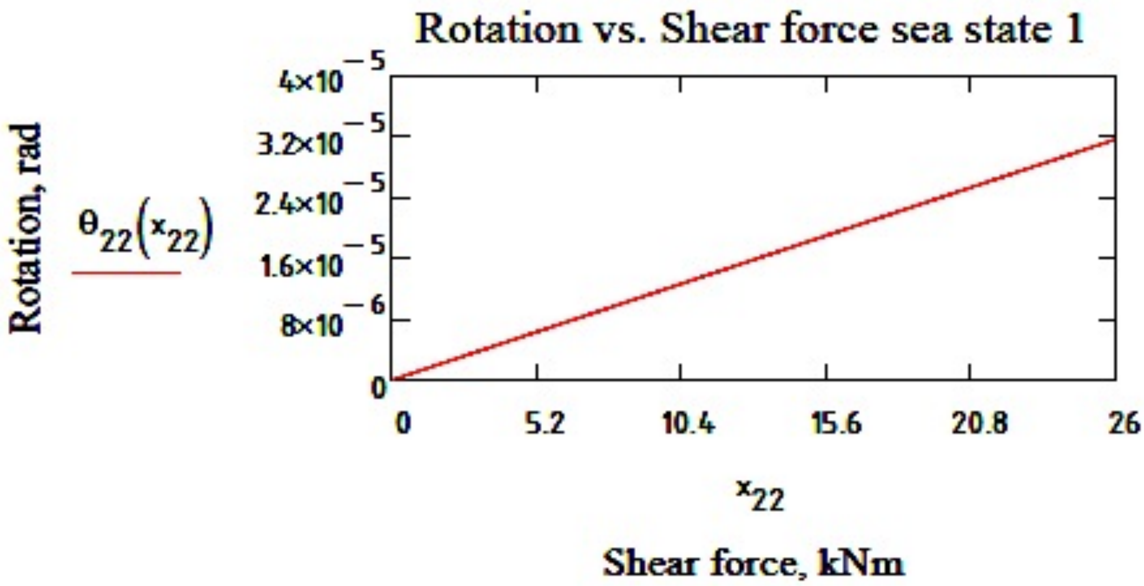


Deflection with maximum moment from sea state 1:

$\delta_{21}(392) = 0.0005$ m

Figure 8-2: Results: deflection vs. bending moment sea state 1

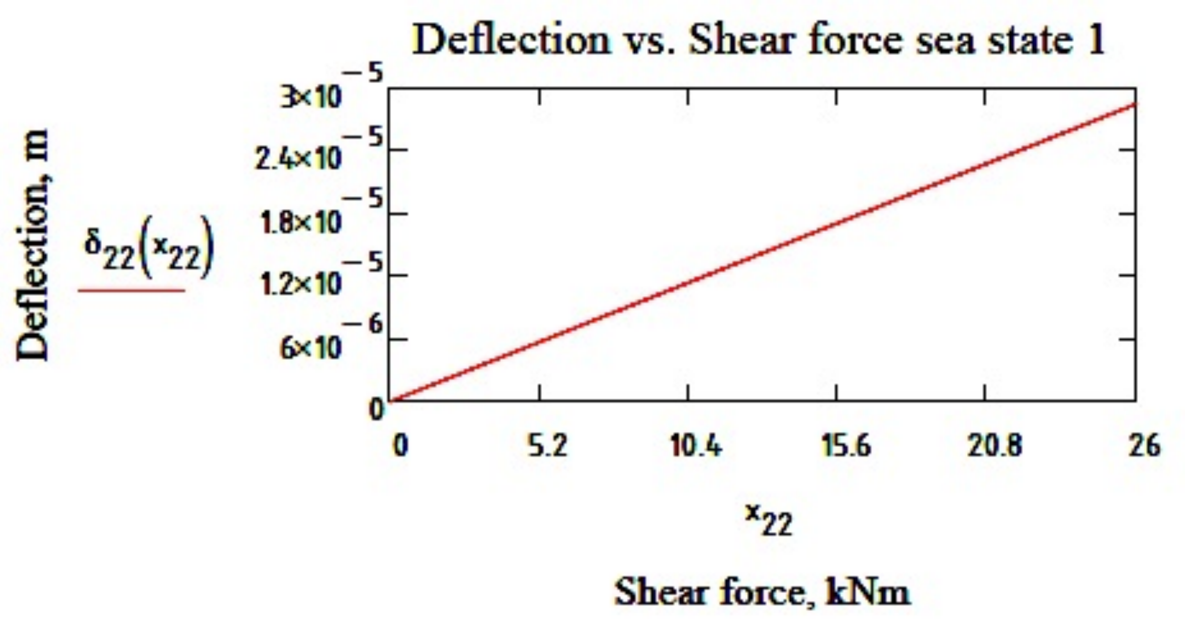
4. Shear force at wellhead datum as a function of rotation and displacement:



Rotation with maximum shear force from sea state 1:

$\theta_{22}(26) = 0.00003$ rad

Figure 8-3: Results: rotation vs. shear force sea state 1



Deflection with maximum shear force from sea state 1:

$$\delta_{22}(26) = 0.00003\,\mathrm{m}$$

Figure 8-4: Results: deflection vs. shear force sea state 1

5. Total angle of rotation and deflection/displacement, sea state 1:

Description	**Values**
Combined deflection (moment and shear force)	0.0005 m
Added deflection (spring)	0.004 m
Total deflection (moment, shear force and spring)	0.0045 m
Combined rotation (moment and shear force)	0.00065 rad
Added rotation (spring)	0.001 rad
Total rotation (moment, shear force and spring)	0.00164 rad 0.094°C

Table 8-2: Results: Total angle of rotation and deflection sea state 1

Comment on results: The results show a small deflection of 4.5 mm and an angle of rotation of 0.094°C witch is very small. From mechanical model 1 in "Appendix A: Mathcad results", page xiii it is shown that rotation and deflection at wellhead datum are the same as the calculated added rotation and deflection.

8.3 Results sea state 2: H_s = 10m and T_p = 12.4s

1. Shear force with maximum flex joint angle (OrcaFlex):

$$F_{shear} = 3060\text{kN} * \sin(2.2) = \mathbf{117\ kN}$$

2. Moment at wellhead datum with maximum flex joint angle (OrcaFlex):

$$M_{WHdatum} = 250\text{kNm} + (117\text{kN} * 12.4m) = \mathbf{1700kNm}$$

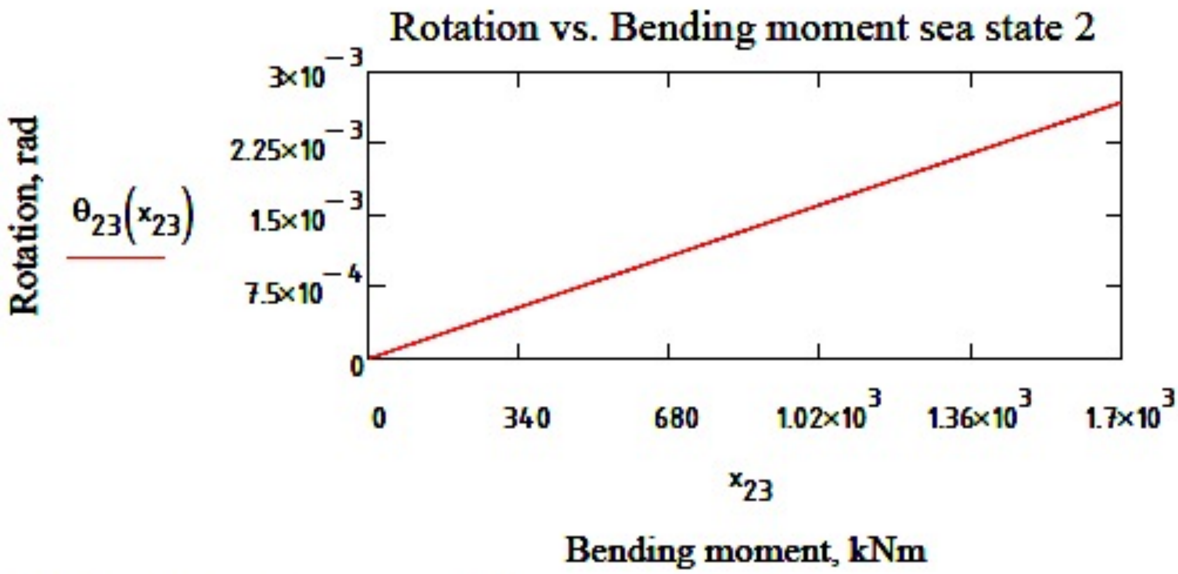
Description	OrcaFlex
Shear force, F_{shear}	117 kN
Bending moment at wellhead datum, $M_{WHdatum}$	1700 kNm

Table 8-3: Results: shear force and bending moment at wellhead datum, Hs=10m

Comment on results: In sea state 2 we have bigger waves and hence the results shows a bigger shear force and a bigger bending moment at wellhead datum.

By varying the rotation in the wellhead the effect on the moment at the wellhead datum can be shown for the different mechanical models.

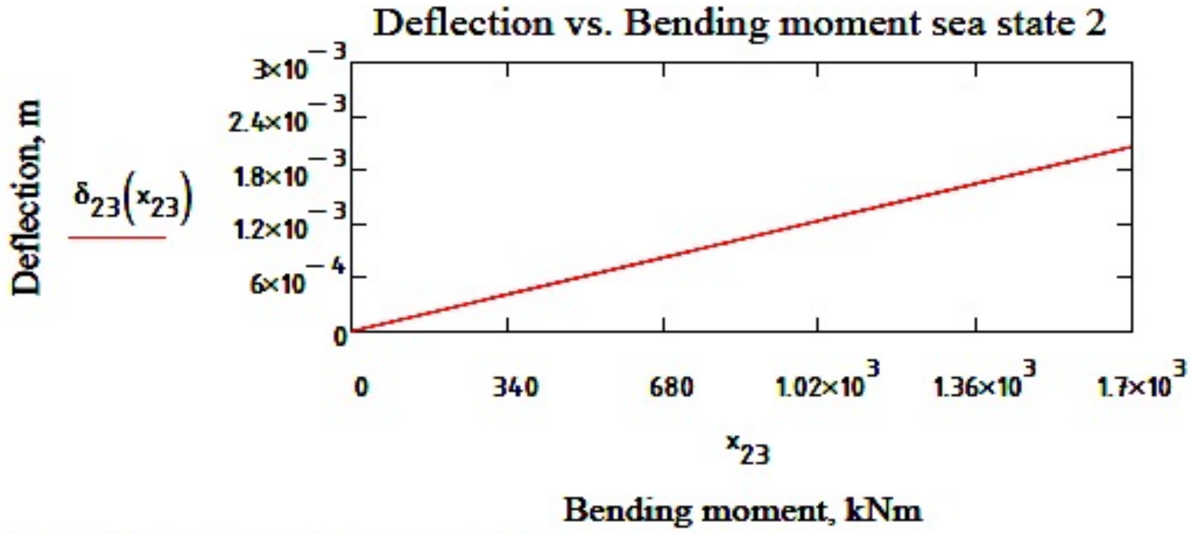
3. Bending moment at wellhead datum as a function of rotation and displacement:



Rotation with maximum moment from sea state 2:

$$\theta_{23}(1700) = 0.003 \text{ rad}$$

Figure 8-5: Results: rotation vs. bending moment sea state 2

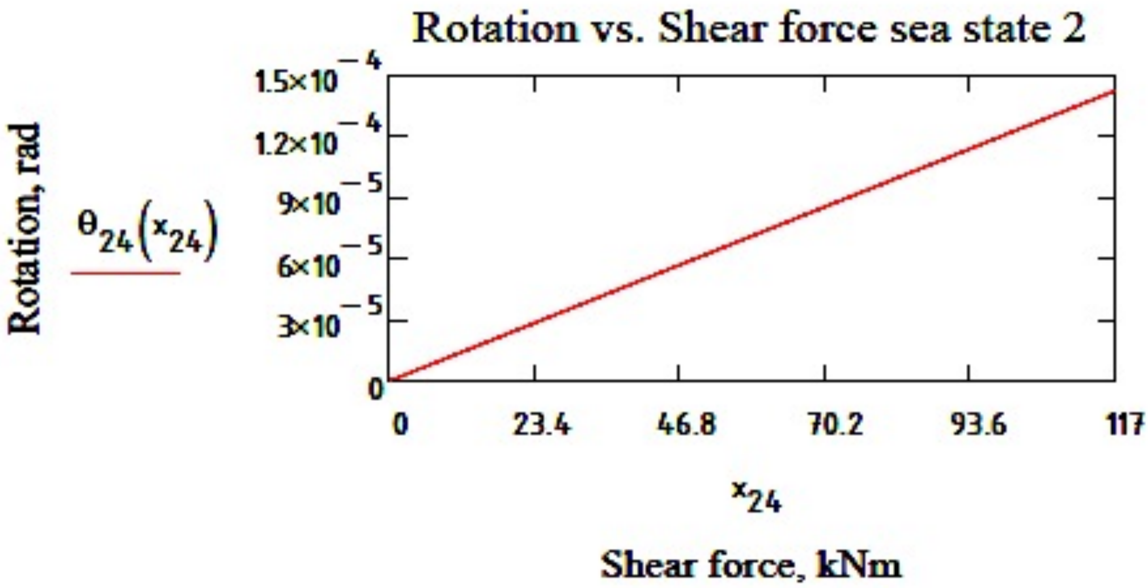


Deflection with maximum moment from sea state 2:

$\delta_{23}(1700) = 0.0021\,\mathrm{m}$

Figure 8-6: Results: deflection vs. bending moment sea state 2

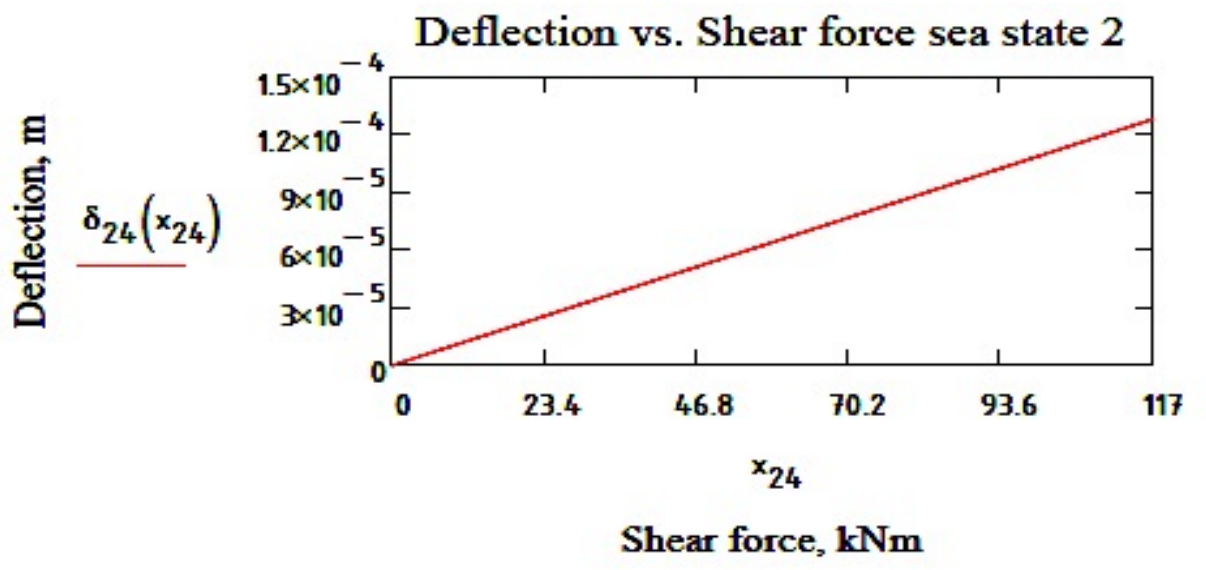
4. Shear force at wellhead datum as a function of rotation and displacement:



Rotation with maximum shear force from sea state 2:

$\theta_{24}(117) = 0.00014\ \mathrm{rad}$

Figure 8-7: Results: rotation vs. shear force sea state 2



Deflection with maximum shear force from sea state 2:

$\delta_{24}(117) = 0.00013\ \text{m}$

Figure 8-8: Results: deflection vs. shear force sea state 2

5. Total angle of rotation and deflection/displacement, sea state 2:

Description	Values
Combined deflection (moment and shear force)	0.00219 m
Added deflection (spring)	0.018 m
Total deflection (moment, shear force and spring)	0.02 m
Combined rotation (moment and shear force)	0.00281 rad
Added rotation (spring)	0.004 rad
Total rotation (moment, shear force and spring)	0.00715 rad 0.41°C

Table 8-4: Results: Total angle of rotation and deflection sea state 2

Comment on results: From this sea state the shear force and moment creates a displacement of 20 mm and an angle of rotation of 0.41°C at the wellhead datum. It is still small numbers because of the large stiffness. From mechanical model 1 in "Appendix A: Mathcad calculations", page xiii it is shown that the model gives approximately the same rotation and deflection as the calculated added rotation and deflection.

8.4 Discussion of results with the use of CAN

If the drilling is done with the CAN installed it would be an even higher stiffness in the spring and this will cause smaller rotations and deflections. As the bending moment on the wellhead connector also becomes zero (ref. fig 4-2, p.19) when the CAN is used, the rotation and deflection become even smaller since the bending moment is the main contribution to the wellhead deflection and rotation. If it is possible to make the stiffness smaller i.e. create a space for the wellhead to move within the stiffness would get smaller and maybe allow the CAN to be used in a wider matter. It could also be evaluated not to transfer the whole bending moment down to the CAN from the BOP to make the situation in the wellhead connector less stiff.

In figure 8-9 below, the deformation of the spring is shown as a function of the spring stiffness:

Deformation of spring as a function of spring stiffness, where x is the spring stiffness:

$$\delta_{spring11}(x_1) := \frac{R_{roller1}}{x_1 \cdot \frac{kN}{m}} \cdot \frac{1}{m}$$

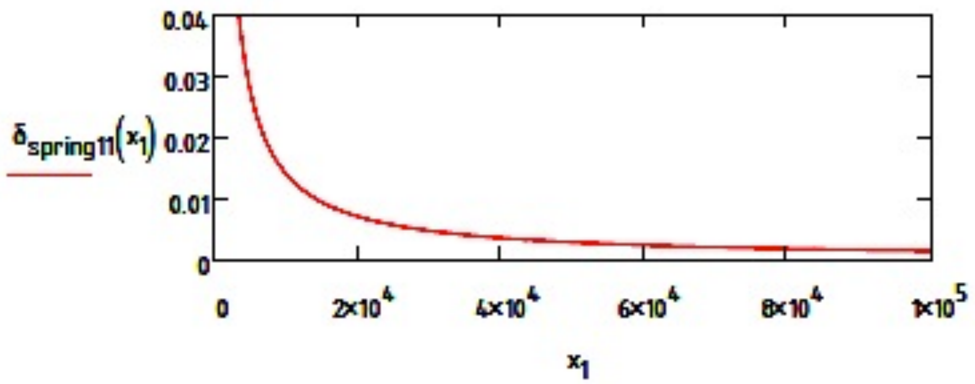


Figure 8-9: Deformation of spring as a function of spring stiffness

In figure 8-9 it is shown that with higher stiffness the deformation gets smaller and smaller.

9 Conclusion And Recommendation Of Further Work

9.1 Conclusion

It is done a riser analysis OrcaFlex to obtain values for the bending moment and the shear force at the flex joint location. From here the forces are transferred and applied on the wellhead datum of simplified mechanical models to see the effect on the angle of rotation and deflection at the wellhead datum.

From the results in chapter 8 it is shown that the whole wellhead system is in fact very stiff and with installation with the CAN it would become even stiffer. The parameter that would stop the deflection capacity is the bending moment at the wellhead reaching the design limit.

However the models only consider linear deflection and rotation (simplified models) witch would not be the case in a real wellhead system. The models provide some uncertainty regarding the soil parameters i.e. more soil models could have been assessed together with the wellhead system to obtain more stiffness properties for the spring. This is not done in this thesis because of missing information about the wellhead connector and soil properties.

Letting the wellhead move within a small area of the CAN could provide a less stiff system and make the CAN applicable for more situations. To do this the wellhead connector fatigue criteria also need to be checked. This can be done according to the "Wellhead fatigue analysis method" written by DNV in 2011.

From the two sea states modelled in OrcaFlex it is shown that there is big differences between applying higher waves to the riser system. The bending moments and shear forces get significantly higher with a significant wave height of 10 m compared with the significant wave height of 7 m, witch is not very surprising.

The results in this project could not be used for any real situations as all the input parameters are worked out from standards or obtained from previously written master theses.

9.2 Recommendation of further work

In this project it is made a lot of assumptions regarding analysis values and simplifications of the calculations because of the missing detailed information about location and parts.

To get a wider applicability of the study the following further work could be recommended:

- Include damping in the spring in the mechanical models (will not have much effect but it is worth to look into). Ref. Hørte, T, 2011.
- Assess soil more properly. Create local models including non-linear soil models and obtain real stiffness of the CAN if considering this type of drilling.
- Create local model with BOP and wellhead connector with real properties to get exact stiffness for actual parts that are going to be used in an drilling operation.
- Implement real analysis values from a real offshore location and situation and compare with obtained results.
- Get real vessel, environmental and riser stack up data for the riser analysis.
- Assess the possibility of letting the wellhead move and not calculate the stiffness of the CAN as infinitely stiff (even if it is tempting) considering the drilling method when evaluating using the CAN from NeoDrill.

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APPENDIX A: Mathcad calculations

Calculation variables:

Wellhead bending stiffness: $EI := 1.4 \cdot 10^{6} kN \cdot m^{2}$

Wellhead length: $H_{WH} := 4.6m$

Positions lateral springs: $x_{spring} := 1m$ (below wellhead datum)

Length notation in Mathcad: $L := 4.6m$ $L2 := 3.6m$ $a := 1m$

Spring stiffness (model 2): $K_{2} := 35 \cdot 10^{3} \frac{kN}{m}$

BOP height: $H_{BOP} := 12.4m$

Symbols used in the Mathcad calculations:

θ = angle of rotation (at wellhead datum)

δ = deflection/displacement (at wellhead datum)

R_{roller} = reaction force at roller (spring)

F = shear force applied at wellhead datum

M = moment applied at wellhead datum

L = total length of wellhead

L2 = length between the supports

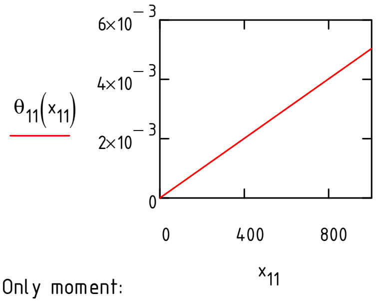
a = length from wellhead datum to spring location (roller support)

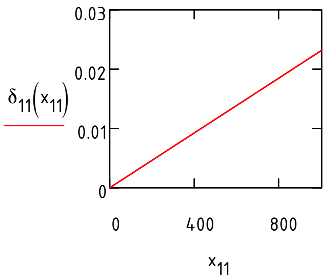
Mechanical model 1:

Only shear force:

$$\theta_{11}(x_{11}) := \frac{x_{11}\cdot L^2\cdot kN}{3EI} \qquad \delta_{11}(x_{11}) := \frac{x_{11}\cdot L^3\cdot kN}{3EI}$$

where x_{11} = shear force at wellhead datum, F

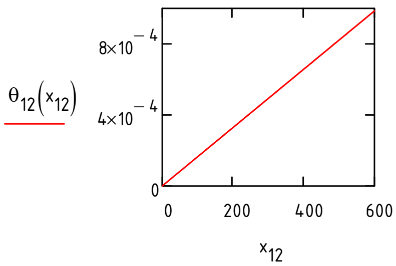


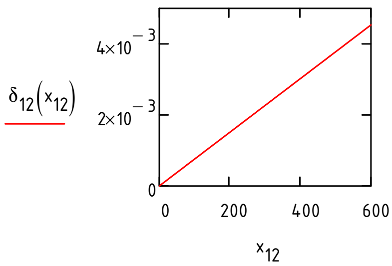


Only moment:

$$\theta_{12}(x_{12}) := \frac{x_{12}\cdot L\cdot kN\cdot m}{2EI} \qquad \delta_{12}(x_{12}) := \frac{x_{12}\cdot L^2\cdot kN\cdot m}{2EI}$$

where x_{12} = moment at wellhead datum, M





Combined maximum total rotation and deformation:

Sea state 1:

$\theta_{13} := \theta_{11}(26) + \theta_{12}(392) = 0.001$ rad

$\delta_{13} := \delta_{11}(26) + \delta_{12}(392) = 0.004\ m$

Sea state 2:

$\theta_{14} := \theta_{11}(117) + \theta_{12}(1700) = 0.003$ rad

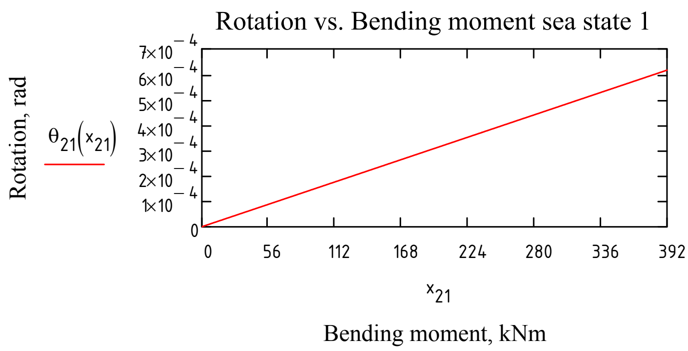
$\delta_{14} := \delta_{11}(117) + \delta_{12}(1700) = 0.016\ m$

Mechanical model 2 – Sea state 1:

Only moment:

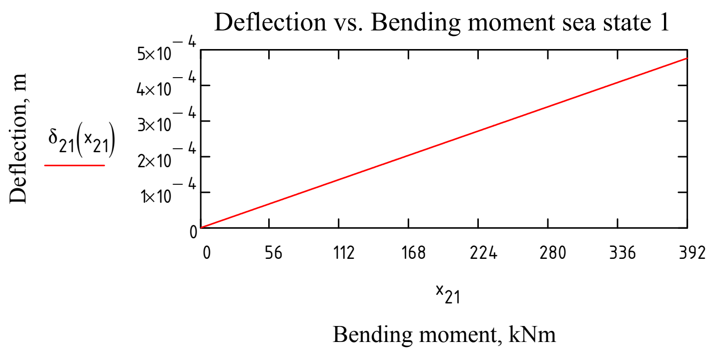
$$\theta_{21}(x_{21}) := \frac{x_{21}\cdot L2\cdot kN\cdot m}{3EI} + \frac{x_{21}\cdot a\cdot kN\cdot m}{EI} \qquad \delta_{21}(x_{21}) := \frac{x_{21}\cdot L2\cdot a\cdot kN\cdot m}{3EI} + \frac{x_{21}\cdot a^2\cdot kN\cdot m}{2EI}$$

where x_{21} = moment at wellhead datum, M



Rotation with maximum moment from sea state 1:

$\theta_{21}(392) = 0.001 \ \ rad$



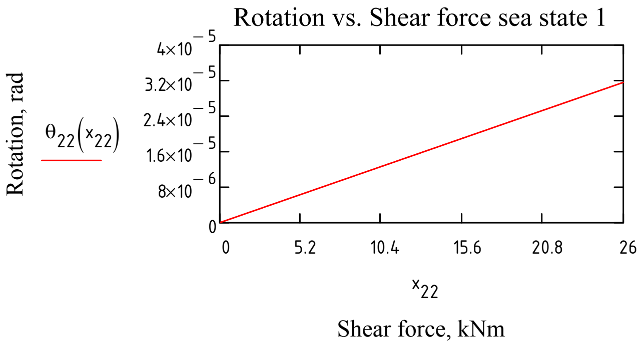
Deflection with maximum moment from sea state 1:

$\delta_{21}(392) = 0.0005\,m$

Only shear force:

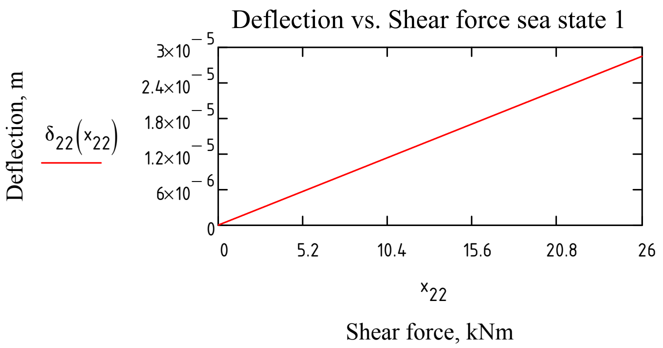
$$\theta_{22}(x_{22}) := \frac{x_{22} \cdot L2 \cdot a \cdot kN}{3EI} + \frac{x_{22} \cdot a^2 \cdot kN}{2EI} \qquad \delta_{22}(x_{22}) := \frac{x_{22} \cdot L2 \cdot a^2 \cdot kN}{3EI} + \frac{x_{22} \cdot a^3 \cdot kN}{3EI}$$

where x_{31} = shear force at wellhead datum, M



Rotation with maximum shear force from sea state 1:

$\theta_{22}(26) = 0.00003$ rad



Deflection with maximum shear force from sea state 1:

$\delta_{22}(26) = 0.00003\,m$

Combined deflection, δ_{21max} and rotation, θ_{21max} model 2 - sea state 1:

$$\theta_{21max} := \theta_{21}(392) + \theta_{22}(26) = 0.00065 \quad \text{rad}$$

$$\delta_{21max} := \delta_{21}(392) + \delta_{22}(26) = 0.0005\,\text{m}$$

Total deflection, δ_{TOT1} and rotation, θ_{TOT1} sea state 1:

Max. values from Orcaflex:

$F_1 := 26\text{kN}$ $\quad M_1 := 392\text{kN}\cdot\text{m}$ — forces sea state 1

$$R_{roller1} := \frac{F_1\cdot(a + L2)}{L2} + \frac{M_1}{L2}$$ reaction force at roller (spring)

$$\delta_{spring1} := \frac{R_{roller1}}{K_2}\cdot\frac{1}{m} = 0.004$$ added deflection

$$\delta_{TOT1} := \delta_{21max} + \delta_{spring1}\cdot m = 0.005\,\text{m}$$ total deflection

Length of "hypotenuse":

$$L_{hyp1} := \sqrt{\delta_{TOT1}^{\;2} + L^2}$$ length from pin support to total deflection

$$\theta_{spring1} := \operatorname{asin}\left(\frac{\delta_{TOT1}}{L_{hyp1}}\right) = 0.001$$ added rotation

$$\theta_{TOT1} := \theta_{21max} + \theta_{spring1} = 0.00164\ \text{rad}$$ total rotation in rad

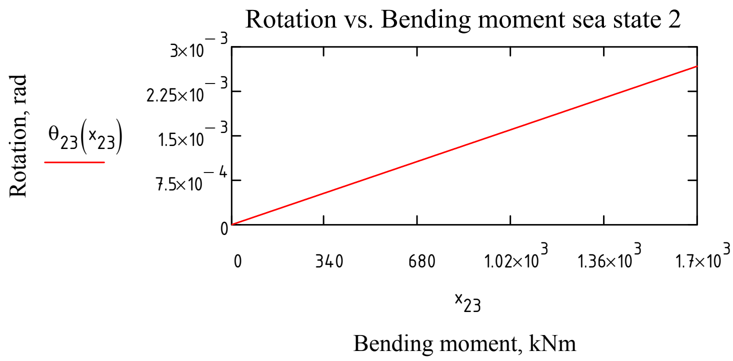
$$\theta_{TOTdegrees1} := \frac{180}{\pi}\cdot\theta_{TOT1} = 0.094\,\text{degrees}$$ total rotation in degrees

Mechanical model 2 - Sea state 2:

Only moment:

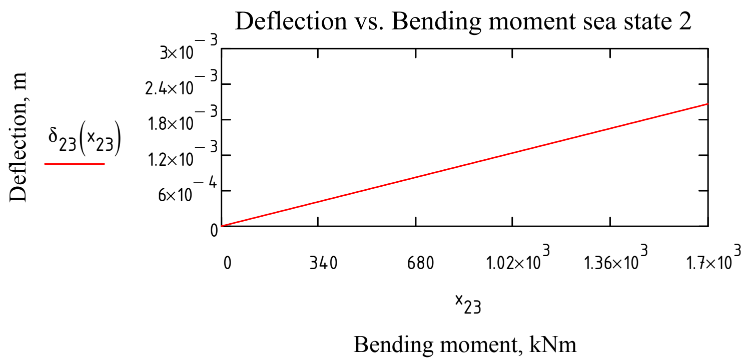
$$\theta_{23}(x_{23}) := \frac{x_{23}\cdot L2\cdot kN\cdot m}{3EI} + \frac{x_{23}\cdot a\cdot kN\cdot m}{EI} \qquad \delta_{23}(x_{23}) := \frac{x_{23}\cdot L2\cdot a\cdot kN\cdot m}{3EI} + \frac{x_{23}\cdot a^2\cdot kN\cdot m}{2EI}$$

where x_{21} = moment at wellhead datum, M



Rotation with maximum moment from sea state 2:

$\theta_{23}(1700) = 0.003\ rad$



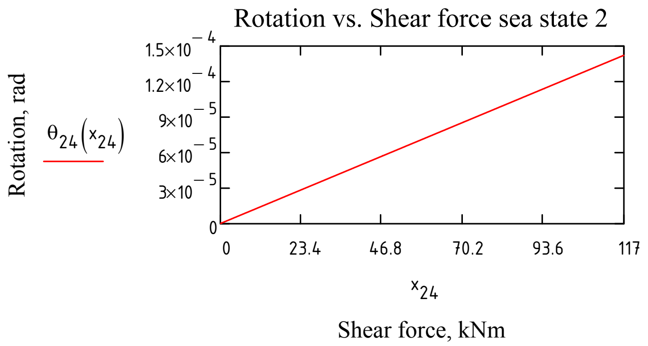
Deflection with maximum moment from sea state 2:

$\delta_{23}(1700) = 0.0021\, m$

Only shear force:

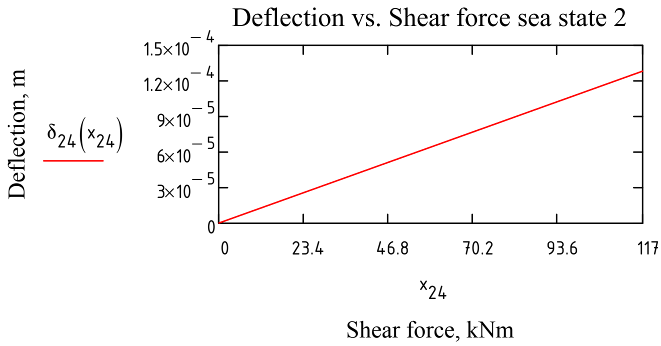
$$\theta_{24}(x_{24}) := \frac{x_{24} \cdot L2 \cdot a \cdot kN}{3EI} + \frac{x_{24} \cdot a^2 \cdot kN}{2EI} \qquad \delta_{24}(x_{24}) := \frac{x_{24} \cdot L2 \cdot a^2 \cdot kN}{3EI} + \frac{x_{24} \cdot a^3 \cdot kN}{3EI}$$

where x_{31} = shear force at wellhead datum, M



Rotation with maximum shear force from sea state 2:

$\theta_{24}(117) = 0.00014$ rad



Deflection with maximum shear force from sea state 2:

$\delta_{24}(117) = 0.00013\,\mathrm{m}$

Combined deflection, δ_{22max} and rotation, θ_{22max} model 2 - sea state 2:

$$\theta_{22max} := \theta_{23}(1700) + \theta_{24}(117) = 0.00281 \quad \text{rad}$$

$$\delta_{22max} := \delta_{23}(1700) + \delta_{24}(117) = 0.00219\,\text{m}$$

Total deflection, δ_{TOT2} and rotation, θ_{TOT2} sea state 2:

Max. values from Orcaflex:

$F_2 := 117\text{kN}$ $\quad M_2 := 1700\text{kN}\cdot\text{m}$ — forces sea state 2

$$R_{roller2} := \frac{F_2\cdot(a + L2)}{L2} + \frac{M_2}{L2}$$ reaction force at roller (spring)

$$\delta_{spring2} := \frac{R_{roller2}}{K_2}\cdot\frac{1}{m} = 0.018$$ added deflection

$$\delta_{TOT2} := \delta_{22max} + \delta_{spring2}\cdot m = 0.02\,\text{m}$$ total deflection

Length of "hypotenuse":

$$L_{hyp2} := \sqrt{\delta_{TOT2}^{\ 2} + L^2}$$ length from pin support to total deflection

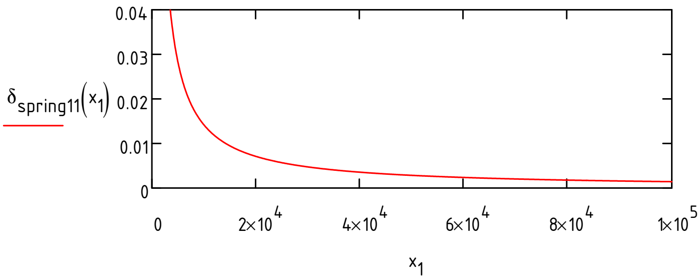
$$\theta_{spring2} := \text{asin}\left(\frac{\delta_{TOT2}}{L_{hyp2}}\right) = 0.004$$ added rotation

$$\theta_{TOT2} := \theta_{22max} + \theta_{spring2} = 0.00715 \quad \text{rad}$$ total rotation in rad

$$\theta_{TOTdegrees2} := \frac{180}{\pi}\cdot\theta_{TOT2} = 0.41 \quad \text{degrees}$$ total rotation in degrees

Deformation of spring as a function of spring stiffness, where x is the spring stiffness:

$$\delta_{spring11}(x_1) := \frac{R_{roller1}}{x_1 \cdot \frac{kN}{m}} \cdot \frac{1}{m}$$



APPENDIX B: RAO Data

The Response amplitude operator explains or shows (graphically) the behaviour of a floating vessel in each of the 6 degrees of freedom of motion.

OrcaFlex uses “Displacement RAOs” and “Wave Load RAOs”. The “Displacement RAOs” define the first order motion of the vessel and depend on the given period and amplitude of the waves. The “Wave Load RAOs” define the first order wave force and moment on the vessel due to the waves of given period and amplitude. The results for the 6 degrees of freedom: surge, sway, heave (translations) and roll, pitch, yaw (rotations) the RAO data consist of 6 amplitude and phase pairs for each wave period and direction. The “phase” defines the timing of the vessel motion relative to the wave [Orcina, 2014]. Some of the RAOs will be presented as a function of the period and phase for the first sea state for the vessel (semi-submersible) used in this project in the following figures.

RAO example values for sea state 1, H_s = 7m, T_p = 11,2s:

Displacement RAOs

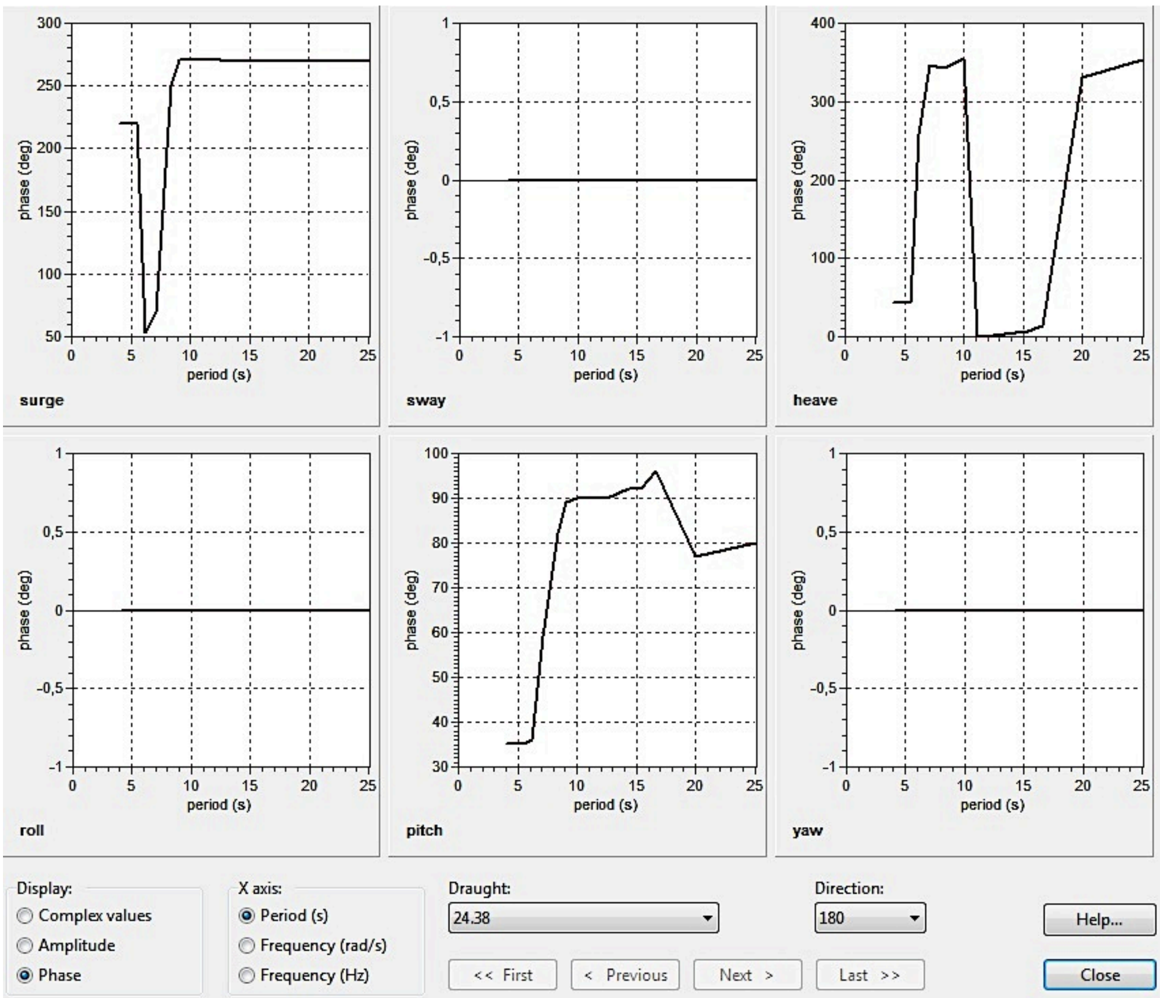


Figure A-1: Displacement RAOs sea state 1

Wave Load RAOs [0°C]

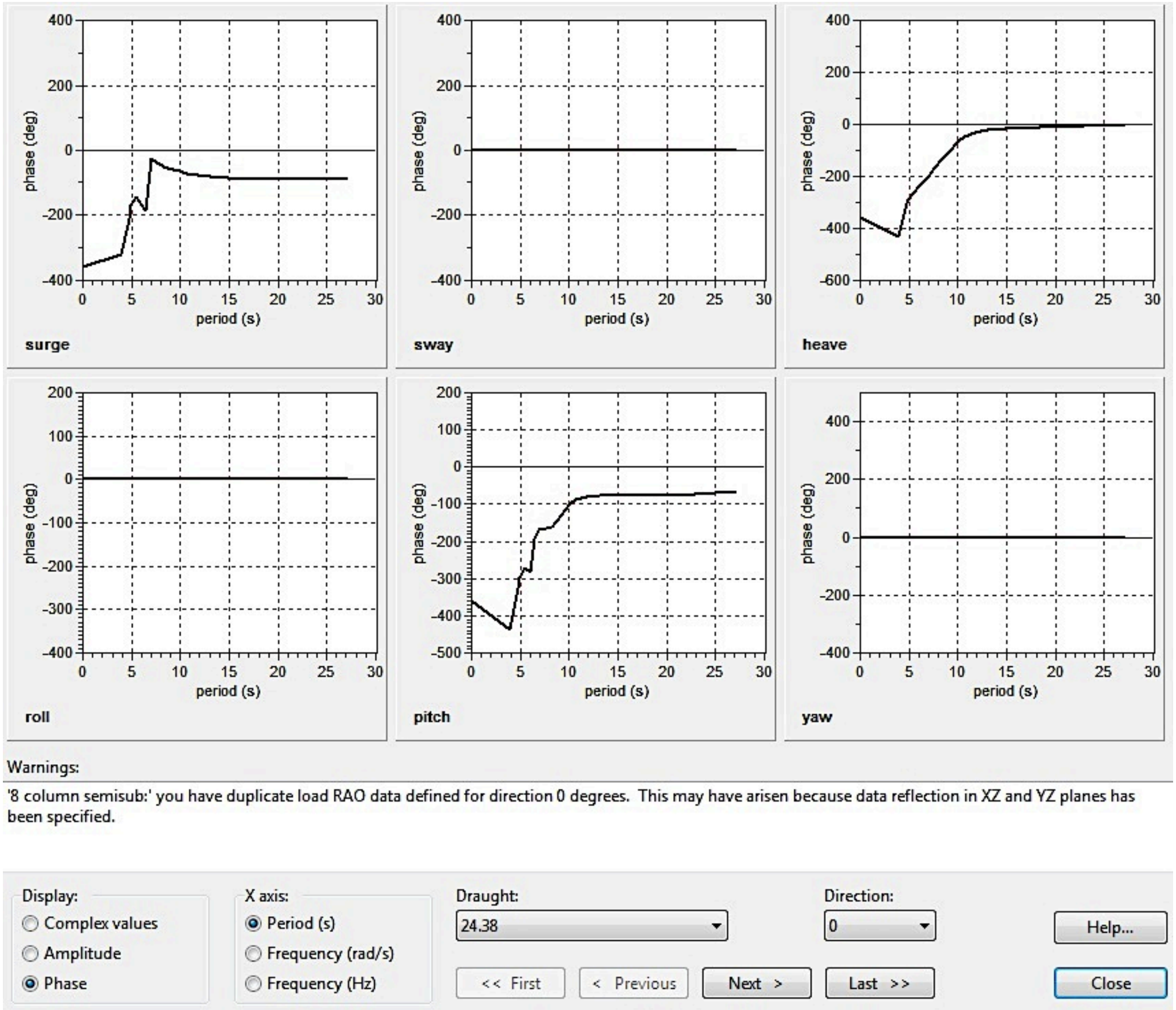


Figure A-2: Wave load RAOs sea state 1 [0°C]

Wave Load RAOs [90°C]

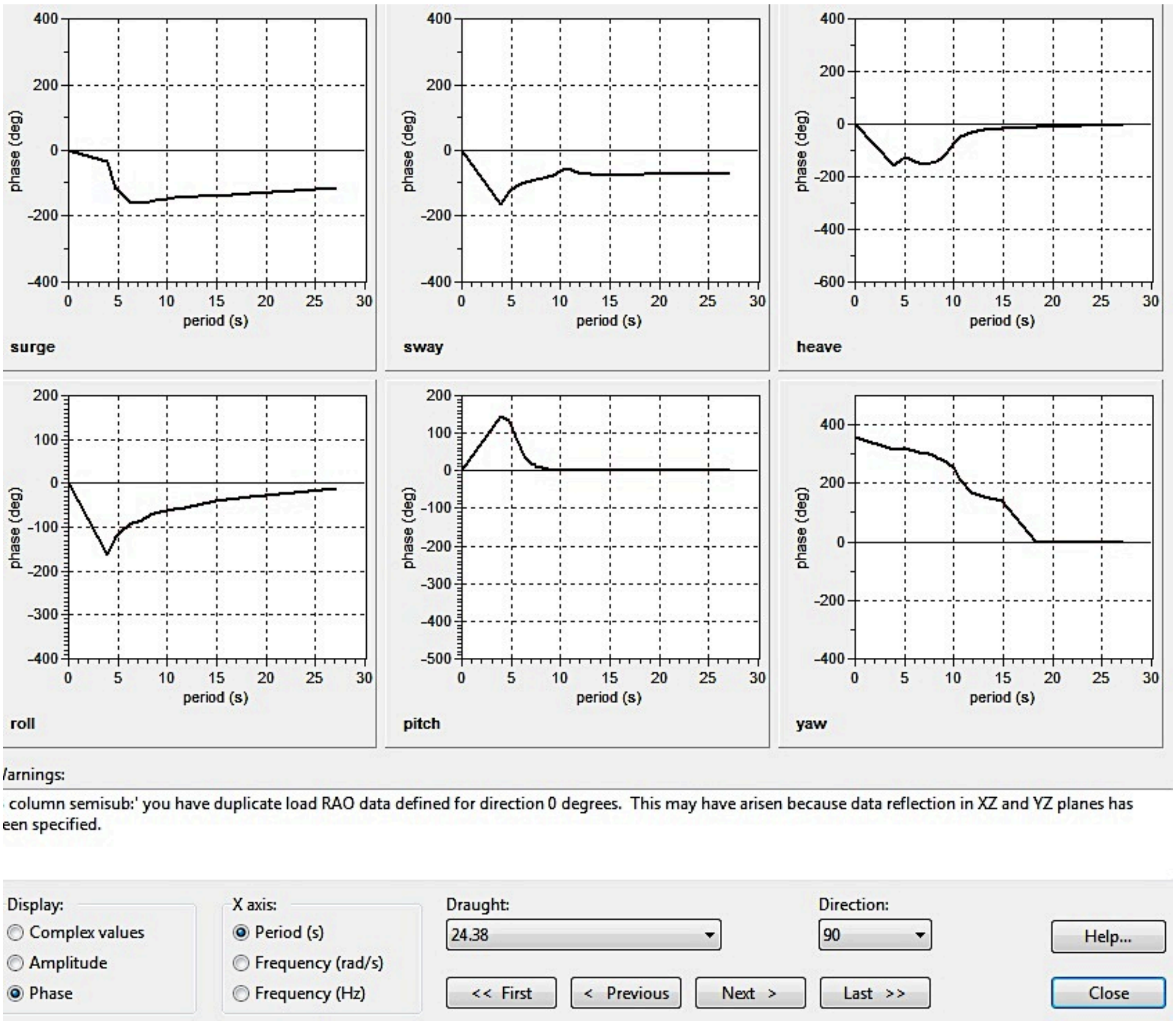


Figure A-3: Wave load RAOs sea state 1 [90°C]

Wave Load RAOs [180°C]

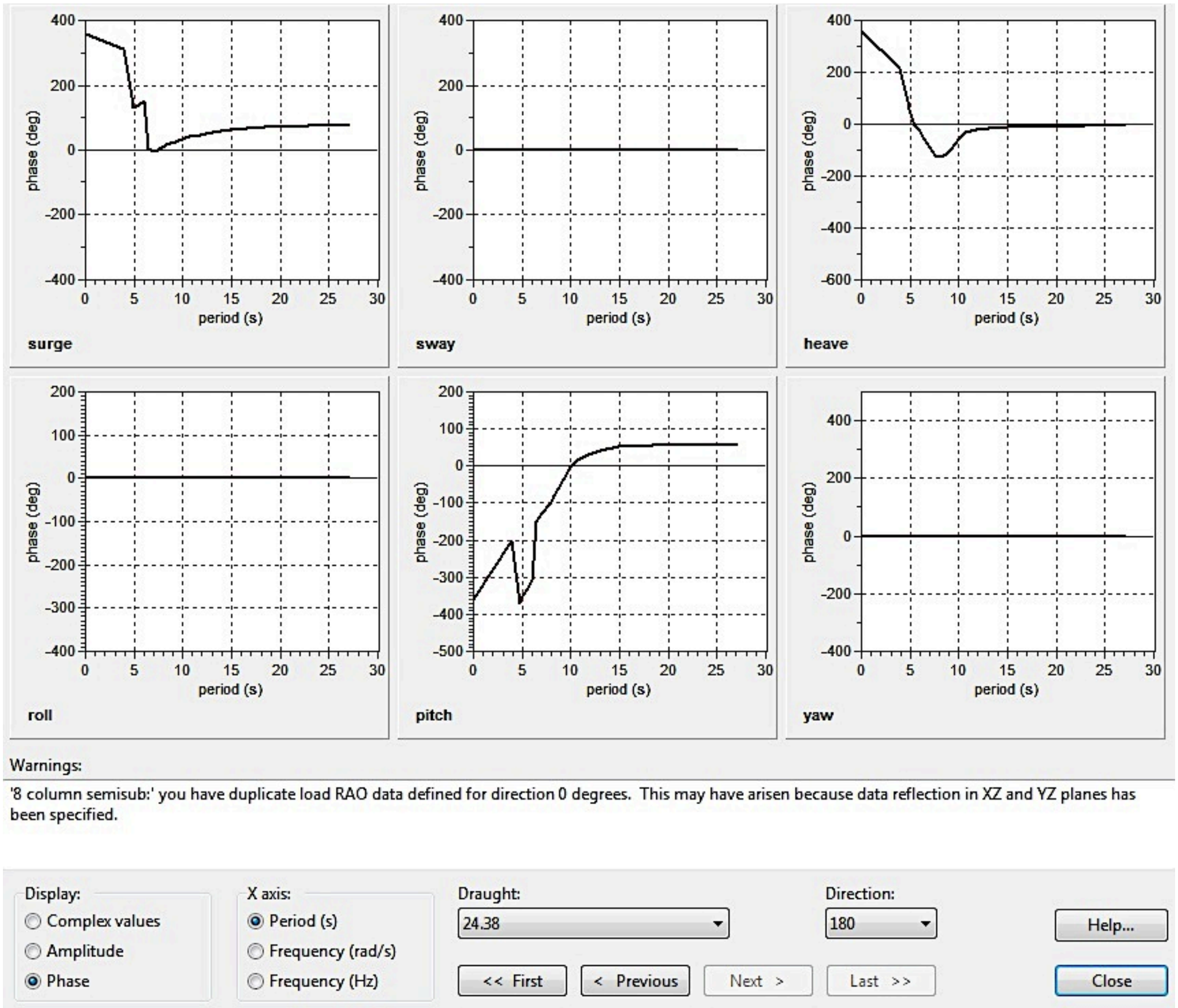


Figure A-4: Wave load RAOs sea state 1 [180°C]

APPENDIX C: Orcaflex Results

Sea state 1, H_s = 7m, MSL = 0, flex joint located at arc length = 1000m:

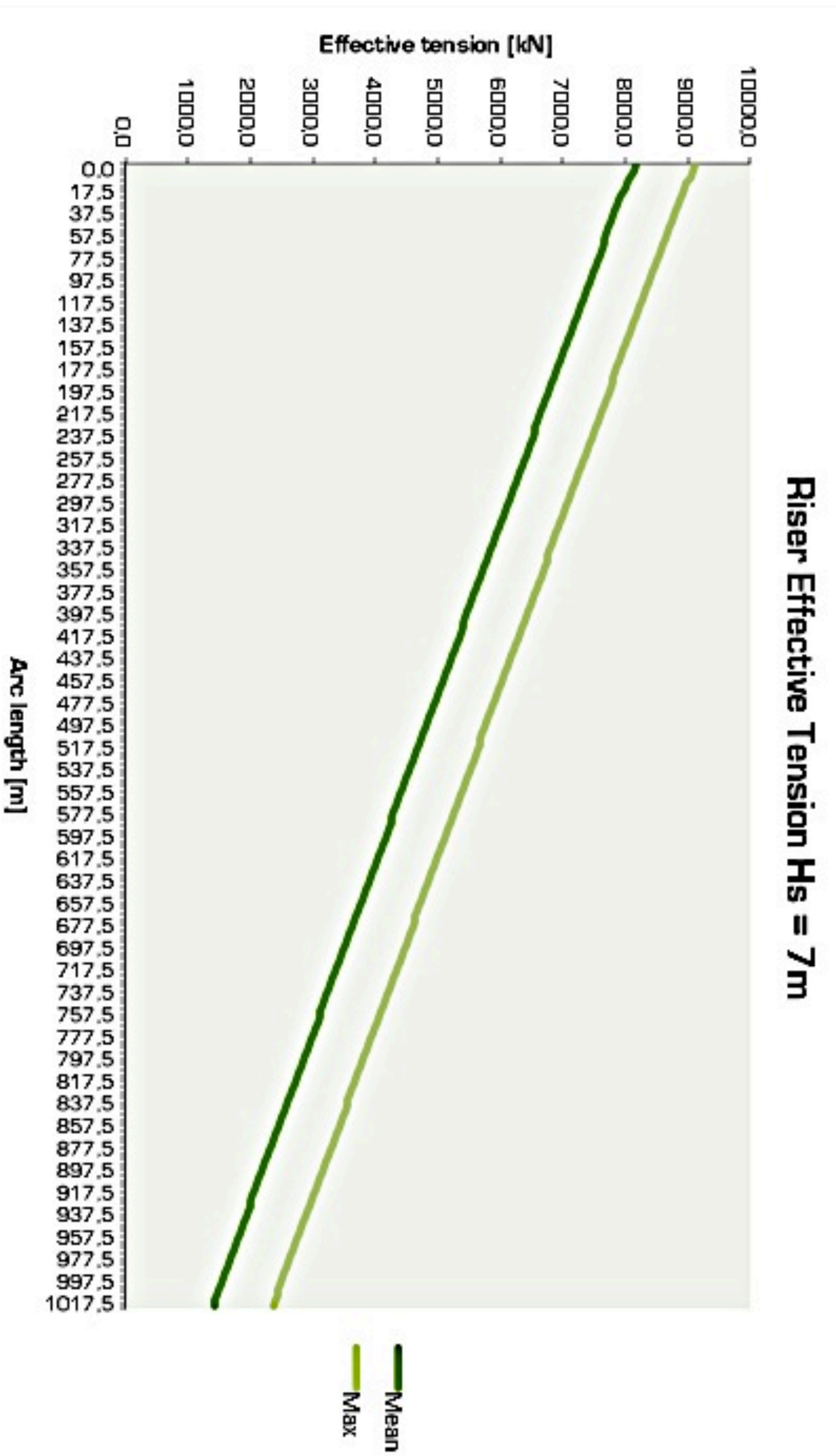


Figure C-1: Riser mean and max effective tension as a function of arc length, Hs=7m

Figure C-2: Flex joint rotation as a function of simulation time (1000s), Hs=7m

Description	Value [deg]
Maximum angle (positive x-direction)	+ 0.58
Maximum angle (negative x-direction)	- 0.60
Allowable during drilling	+/- 5.00

Table C-1: Minimum and maximum flex joint angles, Hs=7m

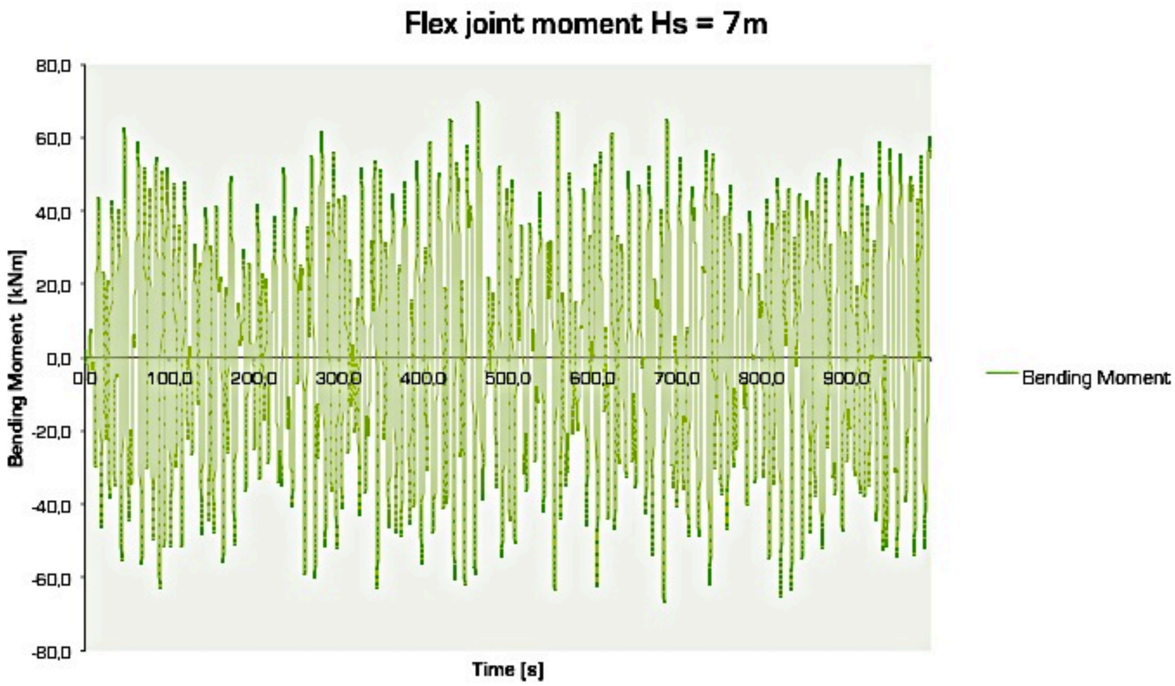


Figure C-3: Flex Joint Bending Moment (y-direction) as a function of time, Hs=7m

Description	Value [kNm]
Maximum BM	+/- 70

Table C-2: Maximum values for flex joint bending moment, Hs=7m

Sea state 2, H_s = 10m, MSL = 0, flex joint located at arc length = 1000m:

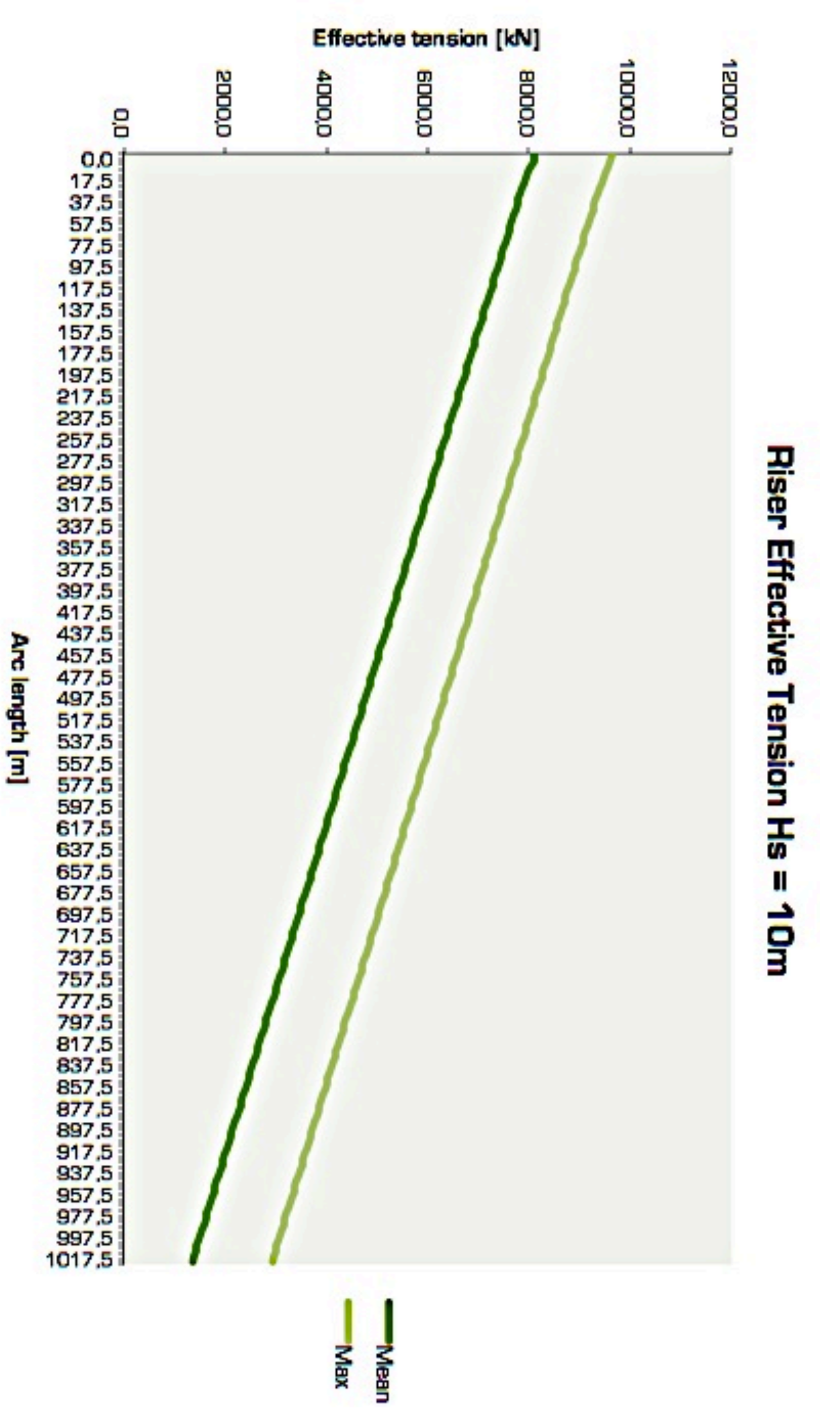


Figure C-4: Riser mean and max effective tension as a function of arc length, Hs=10m

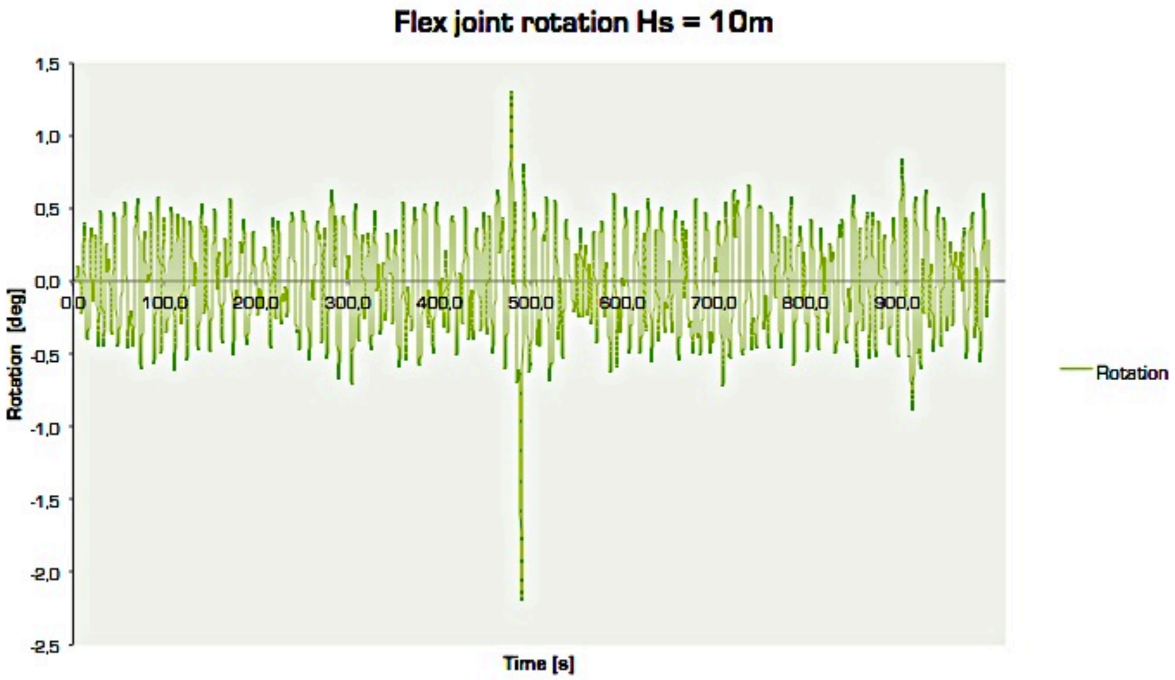


Figure C-5: Flex joint rotation as a function of simulation time (1000s), Hs=10m

Description	**Value [deg]**
Maximum angle (positive x-direction)	+ 1.3
Maximum angle (negative x-direction)	- 2.2
Allowable during drilling	+/- 5.00

Table C-3: Minimum and maximum flex joint angles, Hs=10m

Figure C-6: Flex Joint Bending Moment (y-direction) as a function of time, Hs=10m

Description	Value [kNm]
Maximum BM	+/- 250

Table C-4: Maximum values for flex joint bending moment, Hs=10m