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Design of a Riser Equipment Handling System for a Well Intervention Unit

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for

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Well intervention is an important activity for offshore intervention vessels at oil and gas fields located on the Norwegian Continental Shelf. During a change of operations between two wells at different water depths the length of the riser must be adjusted. This procedure is time consuming because the equipment mounted on the upper section of the riser must be dismounted before the riser can be removed. In order to minimise the turnaround time it is desirable to remove the riser without dismounting the top components. Accordingly, it could be beneficial to establish a method to carry out this operation. This will involve the design and development of any new handling equipment that is necessary, together with its integration into the existing derrick and module handling procedures.

The main objective will be to design a riser handling system that can achieve the goals mentioned above. In order to design and implement a new riser handling system the following points will be addressed in the thesis:

- Define the scope of work.
- Set up a 3D model representing the existing derrick and handling equipment, and establish a new handling sequence within the available space and weight restrictions.
- Identify possible solutions for handling the riser in and out of the tower.
- Check for existing solutions on the market.
- Review advantages/disadvantages with the alternatives.
- Write a design specification for the new handling system, including relevant rules and definition of the load case.
- Design the unit, carry out initial strength calculations and document them during the design process.
- Once the initial design is complete run a Finite Element Analysis of the equipment and check it against the initial calculations. Verify that governing Rules and code safety factors are met.
- Conclude on the overall feasibility of the design.

Within 14 days of starting the assignment the candidate shall send the department a detailed plan for carrying out the work, for evaluation and discussion with the supervisor/contact persons. The thesis should be formulated as much as possible as a research report, with abstract, conclusions, reference list, contents. etc.

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The work often forms part of a larger investigation at the department, which reserves itself the right to use all results in the masters assignment in connection with teaching, publications or other activities.

The thesis is to be submitted in 2 examples. Additional copies to co-supervisors/contact persons from cooperating companies shall be agreed with and delivered directly to them. A complete copy of the thesis shall be delivered to the department on a CD-ROM in Word-format.

This masters assignment is being carried out in cooperation with Aker Solutions ASA, and the contact person is Henrik Vedeld.

Assignment handed out:

Assignment to be handed in:

Assignment handed in:

Maurice F. White
Academic Supervisor

Preface

This final master thesis is written together with Aker Oilfield Services and Norwegian University of Technology and Science (NTNU), for the master degree program at the department of Marine Technology.

This thesis has been a great opportunity to gain knowledge of many different aspects around all the technical solutions in the tower on board Skandi Aker. It has been challenging, but also very inspiring, to be allowed to develop my own design, especially considering the adapter component. It is a great advantage to have had the time to learn the SolidWorks analysis tool to the extent that I have during this project.

I would like to take the opportunity to thank my supervisor at Aker Oilfield Services, Henrik Vedeld, and my academic supervisor at NTNU, Maurice F. White, for all the guidance and support. I would also thank my colleagues, among them Magnus and Sasha, for many valuable discussions.

Oslo

December 22th 2012

Frøydi Røe Flobakk

Abstract

A Well Intervention Unit performs subsea intervention on different water depths ranging from approximately 800-3000 meters. During a change of operations sites, between two neighbouring wells at different water depths, the length of the riser must be adjusted. This procedure is time consuming, and excessive cost-rates are the motivation for evaluating a new solution that may reduce the rigging time.

In this thesis, it is established a new handling sequence reducing the turnaround time by approximately 20-22 hours per trip. Estimated savings is thus 50-60 million NOK on a yearly basis. The suggested handling sequence is to elevate the Upper Riser Pack (URP) higher in the tower, leaving enough handling space underneath to retrieve riser elements. This eliminates the need of rigging down the surface stack, hence the turnaround time is reduced.

To be able to disconnect riser elements efficiently, while the URP is in the tower, a second yoke is introduced to the handling system. A yoke is a passive lifting appliance, and has to be connected to the main hoisting system, which already holds the Upper Riser Pack. The most convenient solution is therefore to connect the yoke to the Surface Flow Tree, the lowest component in the riser stack (UPR). This indicates that the entire Upper Riser Pack has to be elevated together with the new yoke system during riser retrieval. Suggested commercial solution is a hydraulic lifting yoke provided by National Oilwell Varco. This yoke consists of two weld-less BJ links and a BX 5 elevator.

Design specification together with relevant rules and regulations is used to define the load case. Normal operation, accidental heel, and an API-load is used when calculating and simulating the equipment. The API-load is the conservative estimate taken from the API 8C[1] standard of the rated load, i.e. the design load, multiplied with a safety factor of 2.25. This is more conservative than DNV[2] and will therefore be used as an additional test case.

The new riser handling system were drawn and simulated in SolidWorks. The yoke consist of the elevator assembly and two links. The elevator is a component designed to hold riser elements. The links are connection arms that give the necessary handling space between the Surface Flow Tree and the risers. The final component is the adapter; this component is designed to connect the yoke and the Surface Flow Tree. This component is tailor made for the operation sequence and the existing equipment. It is designed so that no additional modifications have to be made to the Upper Riser Pack.

Finally, a strength analysis of the new riser handling equipment is computed, and the results are presented. Some local yielding occurs, especially in the contact regions, but the overall impression is that the equipment satisfies the rules and safety requirements provided by both DNV and API.

[1] American Petroleum Institute
[2] Det Norske Veritas

Sammendrag

En brønnintervensjonsenhet utfører subsea-intervensjon på brønner plassert på 800 til 3000 meters dyp. Rørlengden må justeres når enheten bytter lokasjon mellom to nabobrønner plassert på forskjellig dyp. Denne prosedyren er tidskrevende og høye dagsrater gir motivasjon til å evaluere nye løsninger som kan redusere tidsforbruket.

I denne masteroppgaven blir det presentert en løsning som reduserer riggetid med ca. 20-22 timer per tur. Den totale besparingsevnen er estimert til 50-60 millioner kroner per år. Det nye håndteringsforslaget er å løfte den øvre riser-pakken høyere opp i tårnet, slik at det frigis håndteringsrom til riser-håndtering. Det er dermed ikke nødvendig å demontere den øvre riser-pakken og mye riggetid blir derfor spart.

For å være i stand til å fjerne riser-elementer på en effektiv måte, mens den øvre riser-pakken fortsatt er i tårnet, må det introduseres ett åk. Et åk er ett passivt løfteutstyr, og må kobles til det eksisterende håndteringssystemet, som er opptatt med å holde riser-pakken oppe. Den enkleste løsningen er derfor å koble åket til det øvrige riser systemet, slik at både åket og riser pakken blir løftet når man tar bort riser elementer. Anbefalt kommersielt utstyr er et åk bestående av *BJ links* og *BX 5 elevator* som leveres fra National Oilwell Varco.

Designspesifikasjonen, sammen med regler og anbefalinger, er lagt til grunn for de lastetilfellene som er satt opp. Normal last, krengelast og API-last er brukt når utstyret er kontrollert og simulert. API-lasten er et konservativt estimat tatt fra standarden API 8C[3]. Den består av en designlast ganget med en sikkerhetsfaktor på 2,25. Dette er ett mer konservativt lasttilfelle en de man finner i DNV[4], og blir derfor tatt med som testtilfelle.

Det nye riser-håndteringsutstyret er tegnet og simulert i SolidWorks. Åket består av en elevator og to armer (links). Elevatoren er en komponent designet for å holde riser-elementer. Armene er brukt til å skaffe den nødvendige håndteringsrommet mellom Surface Flow Tree og riser-elementet. Den siste komponenten er en adapter. Denne komponenten er spesiallaget for denne operasjonssekvensen og passer det eksisterende utstyret. Den er designet slik at ingen modifikasjoner av det eksisterende utstyret er nødvendig.

Til slutt blir det utført en styrkeanalyse av det nye riser-håndteringsutstyret. Det er lokal flyting på noen steder, spesielt i kontakt flater, men generelt ser det ut til at utstyret møter de krav til sikkerhet som blir gitt av både DNV og API.

[3] American Petroleum Institute
[4] Det Norske Veritas

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Nomenclature

a-1	*Total height of Dynaplex hook, CTTF and SFT combined*
a-2	*Length of new handling tools*
a-3	*Riser element length.*
A equipment	*Overall equipment length.*
C	*Geometric stiffness i.e. spring constant.*
C clearance	*Total tower clearances.*
c-top clearance	*Clearance at the top beam (Including a safety margin).*
c-WF clearance	*Clearance at the work floor (Including a safety margin).*
F	*Force*
g	*Acceleration of gravity (9.81 m/s2)*
L_D	*Design Load*
L_d	*Dynamic load*
L_s	*Static load*
M-margin	*Extra in-tower margin (* $y^*{}_{tower} > A_{equipment}$ *).*
R	*Rated Load*
S_F	*Safety factor, DNV*
SF_D	*Design Safety Factor, API*
VR	*Relative velocity*
W	*Working load (static weight of load lifted plus weight of accessories)*
*y*tower*	*In-tower handling space.*
Y tower	*Total tower length.*
γ_f	*Load factor*
γ_m	*Material safety factor*
$\sigma_1, \sigma_2, \sigma_3$	*True principal stresses in combined loading*
σ_{allow}	*Allowable stress level MPa*
σ_{cyc}	*Critical amplitude of alternating stresses defined as the value corresponding to 90% probability of survival*
σ_e	*Flow stress in tension*
σ_y	*Material Yield strength MPa*
τ	Shear stress
ψ	*Dynamic Amplification Factor = DAF*

List of Abbreviations

In this document the following abbreviations are to be understood as:

AHC:	Active Hive Compensator
AKOFS:	Aker Oilfield Services
API:	American Petroleum Institute
CT:	Coil Tubing
CTTF:	Coil Tubing Tension Frame
DAF:	Dynamic Amplification Factor
DNV:	Det Norske Veritas
DP:	Dynamic Positioning
EDP:	Emergency Disconnect Package
Ft:	Feet (1 Imperial unit foot = 30.48 cm)
LRP:	Lower Riser Package
LV:	Lubricator Valve
LWI:	Light Well Intervention
MHS:	Module Handling System
MHW	Main Hoisting Winch
MPD	Moonpool Door
mT:	Metric Ton
NOV:	National Oilwell Varco
OS:	Offshore Standard
ROV:	Remotely Operated Vehicles
SFT:	Surface Flow Tree
sT:	Short Ton
UCF:	Upper Cursor Frame
URP:	Upper Riser Pack
WIU	Well Intervention Unit
WP:	Work Package
XT/XMT:	X-mas tree

Definitions

Deepwater:	Water depths ranging from 150 to 800 meters
SWL:	DNV defines Safe Working Load (SWL) to be the maximum available mass to be lifted Maximum static lifting load.
Ultra-Deepwater:	Water depths deeper than 800 meters
Vessel:	AKOFS purpose built unit, Skandi Aker, for Conventional Rigid Riser Light Well Intervention (CRRLWI) operations

Chapter 1: Introduction

During the past years the offshore industry has boomed. Oil and gas wells are many, and installed on ever-increasing water depths. The field production rate is important and the market for deepwater and ultra-deepwater well interventions is growing. Well intervention covers a variety of maintenance operations performed on producing wells to restore or increase production. Traditionally this type of operation has been conducted by drilling rigs, but the recent growth in the offshore industry has affected the rig availability. This results in excessive cost-rates for drilling rigs, which leads to increasing demand for time and cost efficient offshore solution. Aker Solutions has developed alternative technology that meets today's challenges on efficiency, cost, and water depths. A well intervention vessel operates faster and at lower rates than a drilling rig and it covers a wide range of intervention services.

1.1 Well intervention

The Offshore Technology Conference (1) defines well intervention by the following term:

"Well intervention is defined as a variety of remedial operations performed on producing wells with the intention of restoring or increasing production"

Well intervention is basically different maintenance operations conducted on wells to increase the recovery rate and gather information on the production condition. Services that are provided could include logging, stimulation, installation, remedial actions, or retrieval of damaged or stuck equipment.

Wells may require intervention if it experience flow restrictions, sand in production stream, mechanical failures, stuck equipment, and/or changes in the reservoir characteristic. Intervention could also be conducted to seal off existing regions or to enter new zones. The most common reason for well interventions is to address issues due to changes in the reservoir characteristics. The regularity depends on a number of aspects that include economic considerations, infrastructure and the reservoir characteristics. (2)

The main motivation for intervention is to increase production and the field recovery rate. A field's recovery rate indicates how much oil that can be extracted from a field. Due to accessibility of the drilling equipment and maintenance frequency, the extraction rate differs greatly from the shallow-water to deepwater facilities. The overall deepwater recovery rate *without* intervention is approximately 25% of the total oil amount in that area. With well intervention this could be increased to the same recovery rate as shallow water wells of approximately 40%. (3)

There are three major categories of subsea well interventions; light-, medium- and heavy- well interventions. An operation is categorized after operation complexity and time consumption. The complexity gives an indication of vessel type, equipment, and top side facilities.

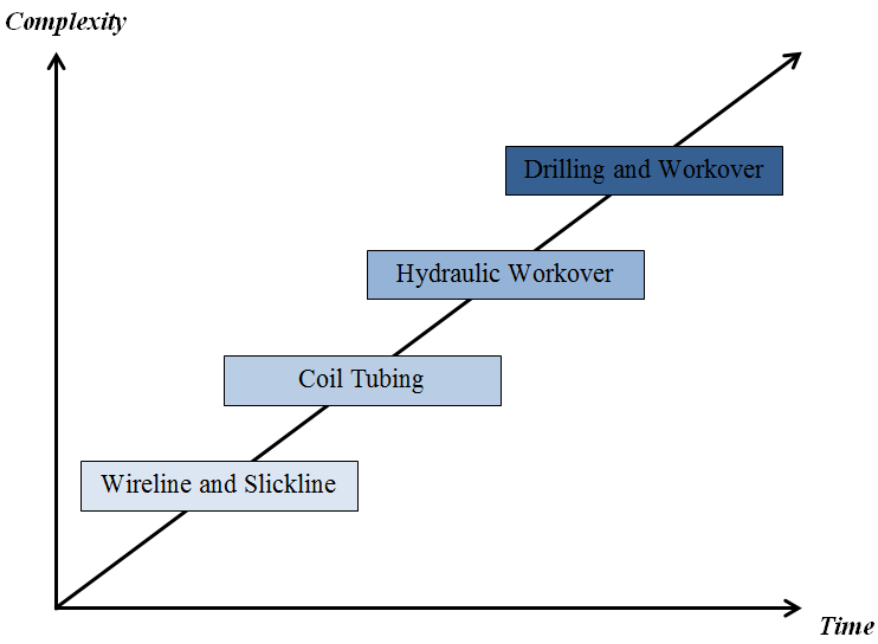


Figure 1: Well intervention classes divided according to complexity and time consumption

Light well interventions include *wireline services* using a subsea wireline lubrication system. Wireline could be electric line or slickline. *Electric line* (E-line) is a steel armoured electric cable that transmits logging data to the surface. It may also perform perforation operations. The *Slickline* is a mechanical wireline that does not supply electricity to the downhole tools. It is used for cleaning, fishing[5] and plug removal as well as to operate or place gas lift valves.

Medium well intervention services include Coil Tubing (CT) services and Hydraulic Workover (HWO). These types of intervention are normally conducted from a semisubmersible and need a workover riser package installed on the wellhead.

Heavy well intervention often needs semisubmersible or large monohull vessels with derrick, rotary table and marine drilling and hoisting equipment. Operations conducted could require pull tubing strings, re-entry drilling equipment, and side tracking equipment.

Well interventions may also be classified by riser-less or riser-based well interventions. **Riser-less** interventions is light well interventions. Riser-less intervention lowers tools down to the wells using wire. **Riser-based** well intervention has riser connection between the well and the operation platform. This is more complex and time consuming than the riser less system, and covers therefore medium and heavy well interventions. Riser-less intervention are conducted on wells down to approximately 800 meters, while wells on deeper waters need riser-based systems.

[5] Removal of downhole equipment as for example stuck wireline tools, packers, liners, or screen pipe

1.2 Skandi Aker

Skandi Aker is a multi-purpose monohull vessel specially designed and equipped for deepwater well intervention. Her main operation is deepwater and ultra-deepwater intervention, but she may also perform shallow water riser-less intervention. She is the fourth unit in a series of sister vessels, and the design has been modified to meet requirements for conventional rigid riser well intervention operations. The combination of tailor-made design and high-tech equipment packages made Skandi Aker *Ship of the year 2010.* (4)

Skandi Akers main capabilities are:

- Rigid riser well intervention operations
- Slickline, Electrical-line, Tractor and Coiled Tubing services
- Well testing and clean-up
- Subsea construction and installation work
- Subsea maintenance operations

Figure 2: The Well Intervention Unit; Skandi Aker

There exist cost efficient alternatives for light well intervention on water depths down to 800 meters. When wells are located deeper than 800 meters, a riser based system is needed. There have so far not been many alternatives to conventional drilling rigs for light well intervention on deepwater wells. Skandi Aker is the first of her kind offering light end medium interventions to the deepwater and ultra-deepwater market segment. Her goal is to operate on water depths down to 3000 meters. Skandi Aker uses a Dynamic Positioning System (DP-system) to maintain her position and can relocate much faster than a drilling rig. She uses ROVs to control the subsea equipment instead of the conventional umbilical line system. This solution makes the equipment lighter and faster to install than conventional equipment. This weight reduction also reduces the wellhead fatigue which is a problem in the offshore industry. The combination of high-tech equipment and fast operation makes Skandi Aker a cost effective intervention alternative. Aker Solution has estimated that she can operate to approximately half of the cost of a conventional drilling rig (5).

1.3 Scope of Work

The well intervention unit performs subsea intervention on several neighbouring wells on different water depths. To access these deepwater wells, the riser systems length has to be adjusted according to the depth of new locations. This is of relevance for Aker Solutions, as their major goal is to provide first-class and efficient solutions for the offshore market.

The objective of this thesis is to establish a riser equipment handling system that enables the vessel to change operation sight more time efficient. The method must be in accordance with existing equipment and meet the constraining factors on-board Skandi Aker. This includes a study of the existing top side equipment, handling sequences and design of new handling tools. When a favoured design is selected a Finite Element Analysis is conducted to ensure that the equipment meets rules, regulations and capacity constrains for this operation. The work will give a complete equipment design.

The report has the following chapters:

Chapter 1 Introduction: Brief introduction on the ship, well intervention and scope of work.

Chapter 2 Background: This chapter gives a technical description of some relevant equipment onboard Skandi Aker. It also includes an introduction of SolidWorks as a design tool and the idea behind Finite Element Method analysis (FEM analysis).

Chapter 3 Commercial Solutions: Brief market assessment including advantages and disadvantages between different alternatives and a suggestion of new equipment are presented.

Chapter 4 Handling Sequence: Evaluation of the handling sequence for the riser retrieval / deployment.

Chapter 5 Design Specification: Design specification including relevant rules and load case.

Chapter 6 Equipment Design: Initial design strength calculations.

Chapter 7 Strength Analysis: SolidWorks simulation evaluating displacements and stress levels in the new handling equipment.

Chapter 8 Conclusion: Conclusion on the overall feasibility of the design.

Chapter 2: Background

This thesis is written without any pre study of the vessel, equipment or problem at hand. A short introduction of the equipment is therefore included. The technical description is a selected summary that will provide some necessary information to understand the problem and solution presented in this report. Illustration of the tower system, normal stack-up and main lifting appliances can be found in Appendix 1 Illustrations.

In addition to the Technical Description, chapter 3 also include a short introduction to SolidWorks as a design tool and Finite Element Method. Only the idea behind Computer Aided Design (CAD) and Finite Element Methods (FEM) are presented. Details of how to use SolidWorks are included in *Chapter 7-Strength Analysis*.

2.1 Technical Description

This chapter covers a brief technical description of some of the equipment on Skandi Aker. The intention with this chapter is not to give a complete description of the equipment, but to provide an introductory account which can assist in the reading of this report. The focus is the equipment found in the tower.

2.1.1 Vessel

Skandi Aker is designed and equipped to meet specific requirements related to deepwater subsea operation as conventional rigid riser well intervention. Her STX[6] monohull design gives her good sea-keeping abilities and a Dynamic Positioning system, provided by Kongsberg Marine, gives her excellent station keeping performance. The Well Intervention Unit (WIU) is environmental friendly, with low fuel consumption, and follows the precautions and requirements stated by DNV's *Clean Design*[7].

2.1.2 Operation

As mentioned before, Skandi Aker uses a riser-based system and performs well intervention operations using rigid riser. Her main capabilities are:

- *Slickline*
- *Electrical-line*
- *Tractor and Coiled tubing services*
- *Well testing*
- *Clean-up*
- *Subsea construction*
- *Installation work*
- *Subsea maintenance work*

[6] STX is the company providing the hull design
[7] DNV is an independent foundation with the purpose of safeguarding life, property, and the environment.

2.1.3 Riser Handling System

Aker Oilfield Services (AKOFS) and National Oilwell Varco (NOV) has optimally furnish a handling system with a riser tensioning system. The main handling system is the derrick located near the mid-ship on Skandi Aker. This section is based on an internal document (6).

Main Hoisting Tower (MHT): is a tower with several lifting appliances mounted above the moonpool. The derrick has an *inside height* of approximately 41.5 metres measured between work floor and underside of tower crown. The *net lifting height* is approximately 30 metres, measured from work floor to underneath elevator block. A guiding rail system is integrated into the tower structure to ensure efficient and safe handling of heavy equipment and subsea modules. The MHT system is designed to work in cooperation with the Active Hive Compensated (AHC) main hoisting winch, riser tensioners, Coiled Tubing Tensioners and Coiled Tubing Tension Frame, Cursor systems, utility winches and Rucker deck. The tower is designed and rated 450 metric tonnes (mT) Safe Working Load (SWL).

Moonpool Door (MPD): The tower features a moonpool door assembly (MPD). The MPD is designed for normal and storm hang-off of the riser stack, and has 450 mT static capacity. The MPD contains skidding rails with rating of 100 mT when the door is secured closed. These are used when lower riser packages are skidded to the moonpool centre for deployment.

Rucker deck: Designed together with National Oilwell Varco. The rucker deck is a substructure underneath the derrick. The rucker deck builds 5.5 meters and houses the riser tensioners, idler sheaves and the work floor (drill floor). The rucker deck can be run up and down into the derrick to provide a stack-up height [8] on the moonpool door of 20 meters, making it easier to launch long stacks.

Spider Jaw: The spider is located on the work floor, and is designed to grips and holds the riser assembly. It builds less than 1 meter and is a part of the work floor/rucker deck.

Skidding system: The deck skidding system is rails covering the entire deck of the vessel. It is designed for safe handling of equipment and cargo. The minimum rating of the skidding system is 60 metric tonnes. An area adjacent to the tower has a 100 metric tonnes rating for handling and storage heavy stacks.

Catwalk Machine (CWM): The CWM is a trolley used to transport riser joints between the riser bay and the tower. The CWM has two arms that guide the riser elements while connecting to the main lifting tool. It also has a slide that supports the lower part of the riser element when the riser is shifted from horizontal to vertical direction into the tower.

Main Hoisting Winch (MHW): The Main Hoisting Winch (MHW) is located on the port side of the tower. The capacities are:

- Single fall reeve 125 mT @ 2000 m, AHC hook speed approximately 2,6 m/s
- Four fall reeve 450mT inside the tower only, AHC hook speed approximately 0,65 m/s

[8] Free height inside the tower available when assembling the subsea stack.

Cursor Frame System: consisting of a frame structure and a hook connected to the main winch. The frame structure is in place to make sure that the hook is centred over the moonpool. The hook has a capacity of 450 mT with a rated load of 550 mT.

Hydraulic lifting yoke: The main riser element handling equipment is the hydraulic lifting yoke provided by NOV. It consists of a hydraulic driven elevator (grab) and two links (arms). The yoke can rotate riser elements from vertical to horizontal position. It has a capacity of 450 mT. The link arms are 6 feet long to ensure enough space while connected to the Coiled Tubing Tension Frame.

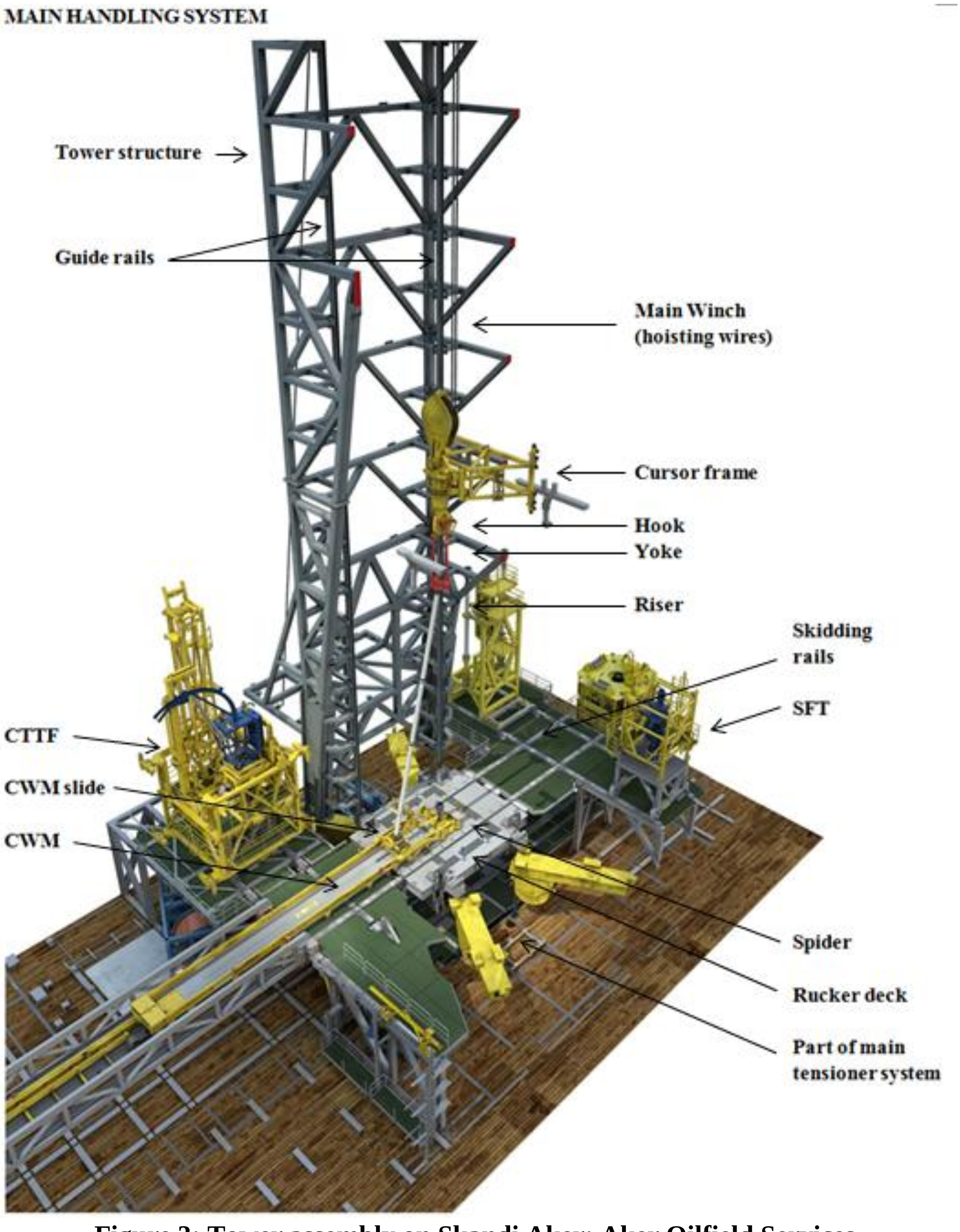


Figure 3: Tower assembly on Skandi Aker; Aker Oilfield Services

2.1.4 Subsea equipment

Coil Tubing Tension Frame (CTTF): The CTTF is a part of the upper surface equipment pack. It is a work platform for the downhole applications and consists of a frame structure, gimbal, and workover package equipment. The frame structure is connected to the tower guide railing. The CTTF is lifted by the main winch via the cursor frame hydraulic lifting yoke which is connected to the CTTF yoke. Coiled Tubing Tensioners help carry the load during operation.

Surface Flow Tree (SFT): The SFT is a control module consisting of a combination of valves. It has an outer frame that is connected to the CT tensioner frame. This frame is not connected to the derrick guide rails.

Landing Joint (LJ): The LJ is the riser element connected to the SFT. This is the first element in the riser stack. It has a special riser pin connection that locks to the SFT, allowing equipment to enter the riser stack and oil to flow to the test plant through the SFT.

Riser Joint (RJ): Riser elements are rigid pipes used in the well intervention. They function as a tube or as an outer shell where equipment can be launched to the seabed. There are two types of standard riser joints used on Skandi Aker; heavy wall and light wall. The outer diameter range from 9.5 to 10.5 inches (241– 267 mm). The length of a standard element is approximately 50 feet (~15 m).

Lubrication valve (LV): The lubrication valve is located in the riser stack. It lubricates downhole tools to ease injection into the well.

Lower Workover Riser Package (LWRP): The LWRP consist of a Lower Riser Package (LRP) and an Emergency Disconnect Package (EDP). LWRP are connected to the X-mas tree (XT) hub on the seabed (well-head).

Figure 4: Coil Tubing Tension Frame (left) and Surface Flow Tree (right)

2.2 SolidWorks as a design tool

CAD programs are relatively easy use. The first step is to gather relevant information and find restraining factors. Input which are important during the design process using CAD and FEM are; *Geometry, Forces,* and *Boundary conditions.* Other aspects are the mesh, number of elements and how big / detailed the component is. These factors determine the simulation time needed and the accuracy of the results. The reason for using CAD as a part of the design process is often more than only strength analysis. SolidWorks has a broad spectre of functionalities that would aid the designer.

SolidWorks provide a tool that can give an accurate 3D representation of mechanical component. The assembly mode can be used as aid while evaluate contact areas, space, complex assemblies, and how to select assembly procedure. Exploded views, cut views and assembly animations are tools that can be used in the evaluation process. Conflicting and colliding components can easily be adjusted or modified. SolidWorks can also evaluate moving components. When a design is selected, technical and manufacturing drawings can easily be produced according to many different pre-set standards.

In addition to the graphical tool SolidWorks provide an analytic tool package. SolidWorks provide a FEM analysis tool that can be used efficiently on complex geometries including both 2D and 3D problems. It can be used to calculate deformations, stresses, strains, temperatures and flows. It works on parts, multi-body parts and assemblies.

Using these functionalities the designer is provided with a good optimizing tool. It delimits the need of manufacturing many different test models before making an actual design. It is also possible to analyse existing equipment. If for instance a component differs from earlier design, FEM can be used to analyse if this particular change in design is tolerable or if it is possible to do modifications that would make it acceptable. An example is manufacturing errors on expensive components or components with long delivery time.

The main disadvantages using an analysis tool as SolidWorks, is that it is time consuming to get the needed accuracy. Implementation of geometry, forces, boundary conditions, and mesh may affect the result greatly. An understanding of the system limitation and functionality are essential to interpret the results correctly. Some hand calculations to validate the results are therefore recommended. In addition to the time needed to implement the components into the program, good computer equipment is needed when running a simulation on reasonable time. Big assemblies are a challenge and the big assembly mode has proven to be slow.

When selecting a CAD program, it is important to select a program that delivers the solution package that you need. There are many good alternatives, but SolidWorks is selected since it has the required functionality, has good tutorials and help-functions, and is well proven.

2.3 Finite Element Method

Today there are several computational methods used for design and manufacture. Finite Element Method (FEM), Finite Difference Method (FDM) and Boundary Element Method (BEM) are three examples, where the FEM is the most commonly used method today. These methods can be used to solve many practical problems found in the industry, as for example the problem presented in this report. The FEM analysis is numerical and is a great alternative for implementation in computer programs. A continuous problem can only be solved mathematically by approximation, creating a discrete system. The fact that the FEM method uses a discrete number of elements makes it possible to solve the problem accurately using numerical methods. Computers in combination with the FEM analysis are therefore a great aid for the engineer. SolidWorks is only one of many commercial softwares that uses FEM analysis as a part of their solution package.

Finite Element Method (FEM), or Finite Element Analysis (FEA), is a numerical method to solve complex problems where the analytic methods have failed. It is difficult to determine the origin of the method, but it started back in the early 1950's. Clough seems to be the first using the term "*finite element"* implying direct *use of standard methodology valid to discreet systems* (7). The idea behind the method is the commonly used technique of breaking a complex problem with unknown behaviour down into small individual components with known physical properties and behaviour. These known property elements are then used to rebuild the system. The idea stands on the assumption *that small elements in a continuum behave in a simplified manner* (7). Analysing these elements one by one, and then assembling them to the original system, would give a description of the behaviour of the complex system which in other cases would not be possible to analyse. The behaviour of each single part gives the behaviour of the complete system due to the assumption of analogue relation.

The term "finite element" implies a *direct use of a standard methodology applicable to discrete systems* (7). The *standard methodology* state that there exist a unified approach for the problems and standard computational procedures can be used. The *discrete system* implies that there are a countable number of elements (finite). A *standard discrete system* is systems where a standard pattern or computational procedure can be applied. The *direct analogy* view between single element and complex system is the reason behind the term "*finite element". *(7)

The method itself is not that difficult to understand, but the theory behind can be more challenging. A common term used is the "divide and concur". The first step is to divide the system into known geometrical elements with known properties, then calculate the properties for each element. The last part in the method is to assemble the components back into the original system.

The methodology can be summarized by:

- *Divide a continuum into a finite number of elements where the behaviour and properties is described by a finite number of parameters.*
- *The behaviour of the complete system assembly follows the same rules as those applicable for the standard discrete problems.*

The FEM analysis gives a detailed view on the complex system. As mentioned earlier there are no analytic methods that can give the same result. The FEM is a natural choice considering the available tools (SolidWorks) and information. The method is highly acknowledged all over the world. It has the advantage that it is relatively easy to implement and interpret. This makes it the engineers' number one tool.

Chapter 3: Handling sequence

Evaluation of the existing system and the problem at hand is important so that best possible solution can be found. Operation, handling sequence, system restrictions and limiting factors should be accounted for in an early design phase. It is very important to make sure that efficient and safe operation is maintained. Critical situations during operation should therefore be in focus while evaluating the system.

Chapter 3 evaluates the handling sequence of the new riser handling system in three parts. In section 3.1 the overall operation sequence for changing operating site is presented, in addition to some critical situations during the transit. Section 3.2 is a suggested riser retrieval sequence – a section which gives a detailed view on the operation where the new handling equipment is used. The next section, 3.3, covers an evaluation of the space limitation in the derrick. This section is of note, since space limitations is considered to be the most critical constraint for the new handling system. The chapter's final part, section 3.4, is an evaluation of potential time savings.

3.1 Main Operation Sequence

A Well Intervention Unit performs subsea intervention on different water depths ranging from approximately 800-3000 meters. The Intervention Unit access several wells during one trip. These wells are often neighbouring wells on same field where the water depth variation are relatively small. To access the wells, the riser assembly length has to be adjusted according to the new depth of the sites. Previously this was done by disconnecting the Upper Riser Pack (URP), consisting of the Coil Tubing Tension Frame (CTTF) and the Surface Flow Tree (SFT), together with riser elements until the desired length was reached. When the vessel is at the new location the URP is reassembled into the tower and the riser stack are landed on the new well. A new suggestion is to elevate the URP higher up in the tower, leaving enough handling space to retrieve riser elements. This eliminates the need of rigging down the URP, hence the turnaround time is reduced.

To be able to disconnect riser elements efficiently while the URP is in the tower, an additional handling tool has to be connected under the URP. The design of this tool is evaluated later in this thesis. A suggested solution is to introduce a second yoke to the handling system. A yoke is a passive lifting appliance, and have to be connected to the main hoisting system, which already holds the Upper Riser Pack. The most convenient solution is therefore to connect the yoke to the Surface Flow Tree, the lowest component in the riser stack (UPR). This indicates that the entire Upper Riser Pack has to be elevated together with the new yoke system during riser retrieval.

On the next page a flowchart describing the suggested vessel operation is presented. Detailed procedure can be found in Appendix 2. The XT-hub is the Christmas-tree, i.e. permanent mounted subsea equipment on the well head. The Lower Workover Riser Package (LWRP) is the subsea equipment landed on the well. The UTA is the umbilical line, which are connected to the LWRP. (This component might not need to be disconnected, but analysis of this problem is not conducted in this thesis).

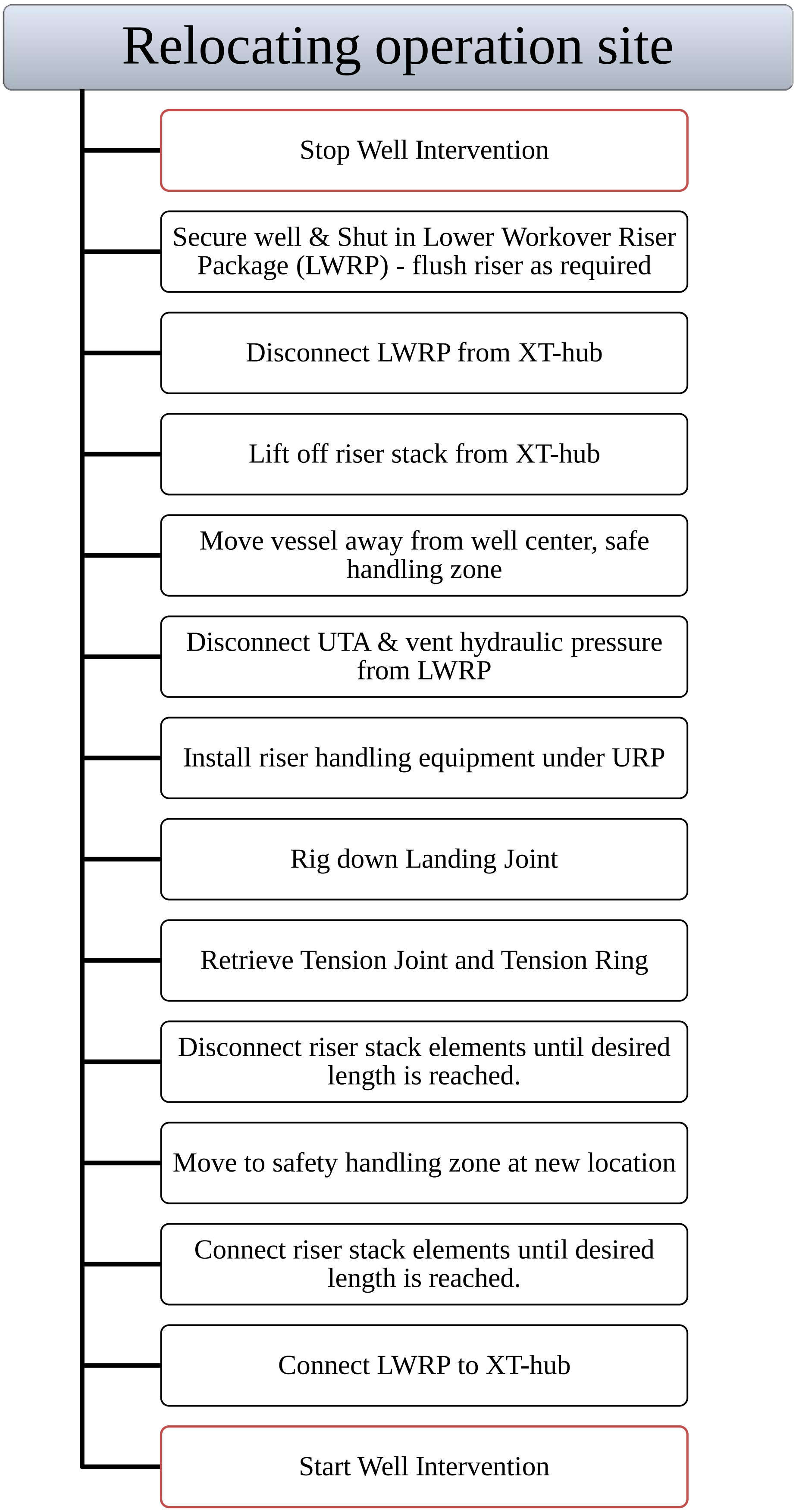


Figure 5: Flowchart of suggested new procedure when relocating

3.1.1 Critical situations, main operation

Crew safety and safe operation is a main focus in Aker Solutions. Down time cost due to damaged equipment in the oil segment are very large and detailed handling-procedures are put in action to ensure safe operation.

Disconnect from well head

The disconnection envelope from the wellhead is set to 4 meters significant wave height (Hs). Collision with the well head may occur if the wave heights exceed this wave height due to tensioner lift-off velocity. The new handling system will not be used during lift-off, so this does not imply any limitations.

Transit after lift-off

There is an increase of stresses when the LWRP is lifted off the XT-hub. The riser tensioners are in fully stroke-out position. This means that only a few tensioner wires support the entire riser load. The elevator should therefore be connected as soon as possible after lift-off due to the extensive stress in the tensioner wires. During transit large bending moments due to vessel roll/pitch are expected in the riser. To reduce moments to acceptable levels the landing joint shoulder should hang in the elevator approximately 1 meter above the closed spider. The riser stack should also follow the heave motion of the vessel to overcome dynamic loads in the riser ((8) page 24). The new handling system will be used during transit. This indicates that dynamic loads from the riser system and bending moments should be evaluated.

It is also important that the vessel and the riser system (LWRP) have a safe path to the safety zone. This to ensure that there is a safe distance between the LWRP and other neighbouring subsea facilities in case of falling object and collision.

Riser retrieval

Riser retrieval is the main operation for the new handling tool. Running riser elements through moonpool has an operation limit of 2 meters Hs due to wave forces in moonpool. Riser retrieval and deployment is the main operation for the new handling system, and 2 meter Hs will be the operation limit.

Additional preparations to enable new operation

In addition to handle riser elements, other components that may be handled using the new equipment are the Landing Joint, Tensioner Joint, and the Tensioner Ring (see stack up in Appendix 1). The next component on the riser stack is the Lubrication Valve (LV). The Lubrication Valve assembly should be installed earlier i.e. further down on the riser system. This alteration is put in action so that it would not be necessary to disconnect this section when changing operation site. This would enable more riser joints to be removed time efficiently. An estimate is to install the LV 150 m under the intervention unit[9].

[9] Dialogue with Henrik Vedeld

3.2 Handling sequence for riser retrieval

The handling sequence is preformed while the upper subsea pack is located in the tower. The equipment involved would be the Coil Tubing Tension Frame (CTTF) and the Surface Flow Tree (SFT) located in the tower. The SFT is connected underneath the CTTF, creating what is here called the Upper Riser Pack (UPR). The CTTF is connected to the cursor frame hook via the gimbal yoke (mechanical) and a hydraulic lifting yoke (NOV). The cursor frame is connected to the main winch [10]. New procedure is based on (8) and (9).

Riser retrieval

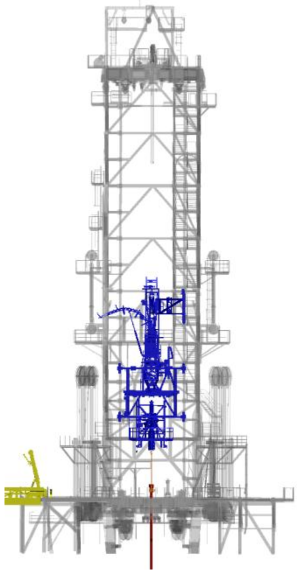
The starting point for the riser retrieval is when the vessel is located in a safe handling zone. The Landing Joint is disconnected from the SFT and hung-off in the spider. Riser retrieval can begin when the new yoke system is installed under the SFT, and safety routines are preformed to ensure safe handling. The Landing Joint is retrieved in the same manner as a standard riser element. The following description gives a simplified retrieval procedure for riser elements:

1. Connect the yoke to the riser that are currently hung off in the spider. The upper part of the riser element is stronger and has two hang-off shoulders. The riser is hung off on the lower shoulder leaving room for the yoke clamp. See Appendix 1 for illustration.
2. Elevate the in-tower equipment[11] until next riser joint is approximately 0.5-1.0 meters above Work Floor. The entire riser stack weight is now carried by the new handling tool. Lock and secure the riser element in the spider jaw, so that it is resting on the lower shoulder, reliving the load from the yoke.
3. Disconnect riser joint from the main riser stack in the spider jaw. Skid the Catwalk Machine (CWM) to the moonpool area (MP-area). Use the CWM Pipe Trail-in Arm (PTA) to secure the disconnected riser element when it is elevated to get handling clearance (Approximately 0.5 meters above riser stack).
4. Guide the riser element onto the Catwalk Machine trolley using the PTA, and land it on the CWM slide. This slide is used in step 5 when the element is lowered to horizontal position. The riser pin head is checked for damages before a protection cap are fitted.
5. After the pin is controlled the riser joint is laid down in horizontal position on the trolley. In this step the elevator and riser are rotated from vertical to horizontal direction.
6. The last step is to disconnect the riser form the elevator. When a riser element is disconnected, it is skidded to the storage area by the Catwalk Machine. A Cargo Riser Crane lifts up each element and stores them in the riser bay. Note that there is strict regulation as regards where each element should be placed to ensure rotation of the riser stack element order.

Steps 1 to 6 are repeated until the riser stack has the desired length. Each riser element is approximately 15.6 meters. Figure 6 shows the handling sequence illustrated by a SolidWorks model. The tower has a grey colour, the upper riser package is blue, riser elements are red and the Catwalk Machine is yellow. Detained procedure can be found in Appendix 2.

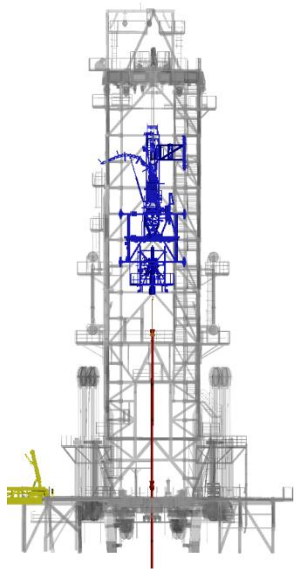
[10] Equipment order from top: Main Winch – Cursor Frame – Hook – Hydraulic yoke – Gimbal yoke – CTTF – SFT.
[11] Main handling tools + Upper Riser Package + new yoke system

Step-1:



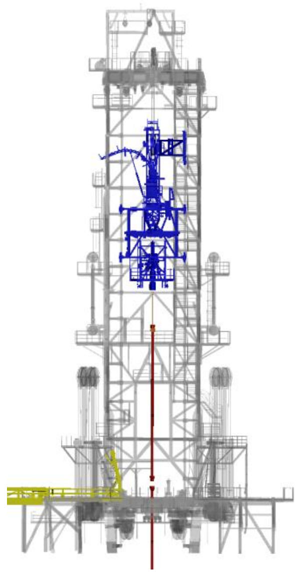
Connect yoke to riser

Step-2:



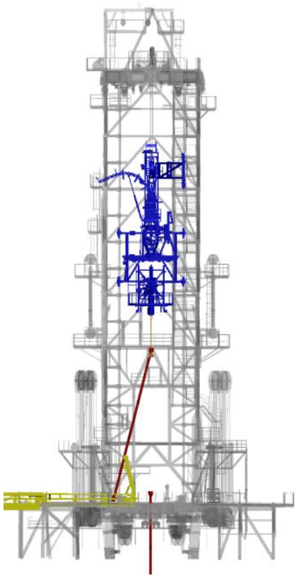
Lift riser stack until next riser joint is above woork floor and lock spider

Step-3:



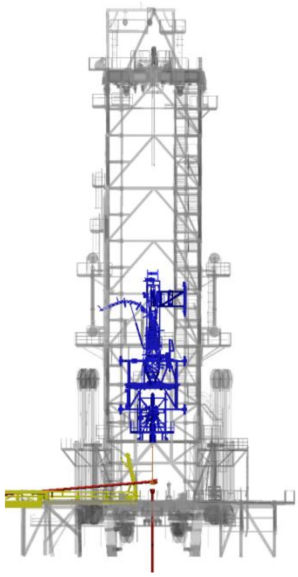
Disconnect riser element from riser stack

Step-4:



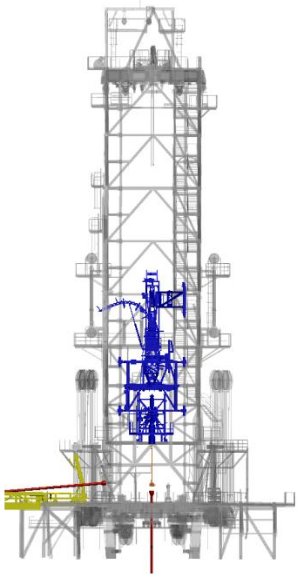
Lower the riser element down to the CWM trolley slide

Step-5:



Lay down the riser element on the CWM trolley

Step-6:



Disconnect the yoke. Skid CWM to storage area to store the riser element

Figure 6: Handling sequence illustration, 6 step procedure

3.2.1 Critical operation situations

Equipment in tower

The Coil Tubing (CT) equipment, located in the Coil Tubing Tension Frame (CTTF), reach higher than the Upper Cursor Frame when mounted in the tower. This is of importance since the CT might collide with the tower top if it is elevated to high. The alarm indicating the maximum position of equipment in the tower is calibrated for the Upper Cursor Frame (hook system), and has to be recalibrated for safe operation. The alarm system is illustrated in Appendix 1.

Another alteration is that the Coil Tubing arm has to be lowered so that it does not collide with the sides of the tower structure.

3.3 Space evaluation

The aim of this section is to identify the operation space inside the derrick. Equipment located in the tower can be divided into three main groups;

1) In-tower equipment (Dynaplex hook, CTTF & SFT),
2) New handling tool
3) Riser elements

The total tower handling zone is reduced due to clearances, safety margins, intervention equipment located inside the tower, new handling tool, and safe handling zone for the riser element. Necessary in-tower clearances consist of handling safety clearances at the crown and work floor. The margin is a minimum safety distance adding to the riser handling zone. An appropriate length for the new yoke system can be selected when the available space is identified.

3.3.1 In-Tower space and handling tool length

Clearances around moving equipment are important to ensure safe handling. There has to be room for operating tools and manoeuvring disconnected riser element in a safe manner. At this point in the design phase it is of interest to find out how much space the new handling tools can take before conflicting with safe handling and existing equipment.

Following are an illustration of the tower with equipment. Measurements of the SolidWorks 3D model give an indication of the different in-tower equipment lengths. The different equipment overlaps, so the in-tower height is not equal to the actual equipment length.

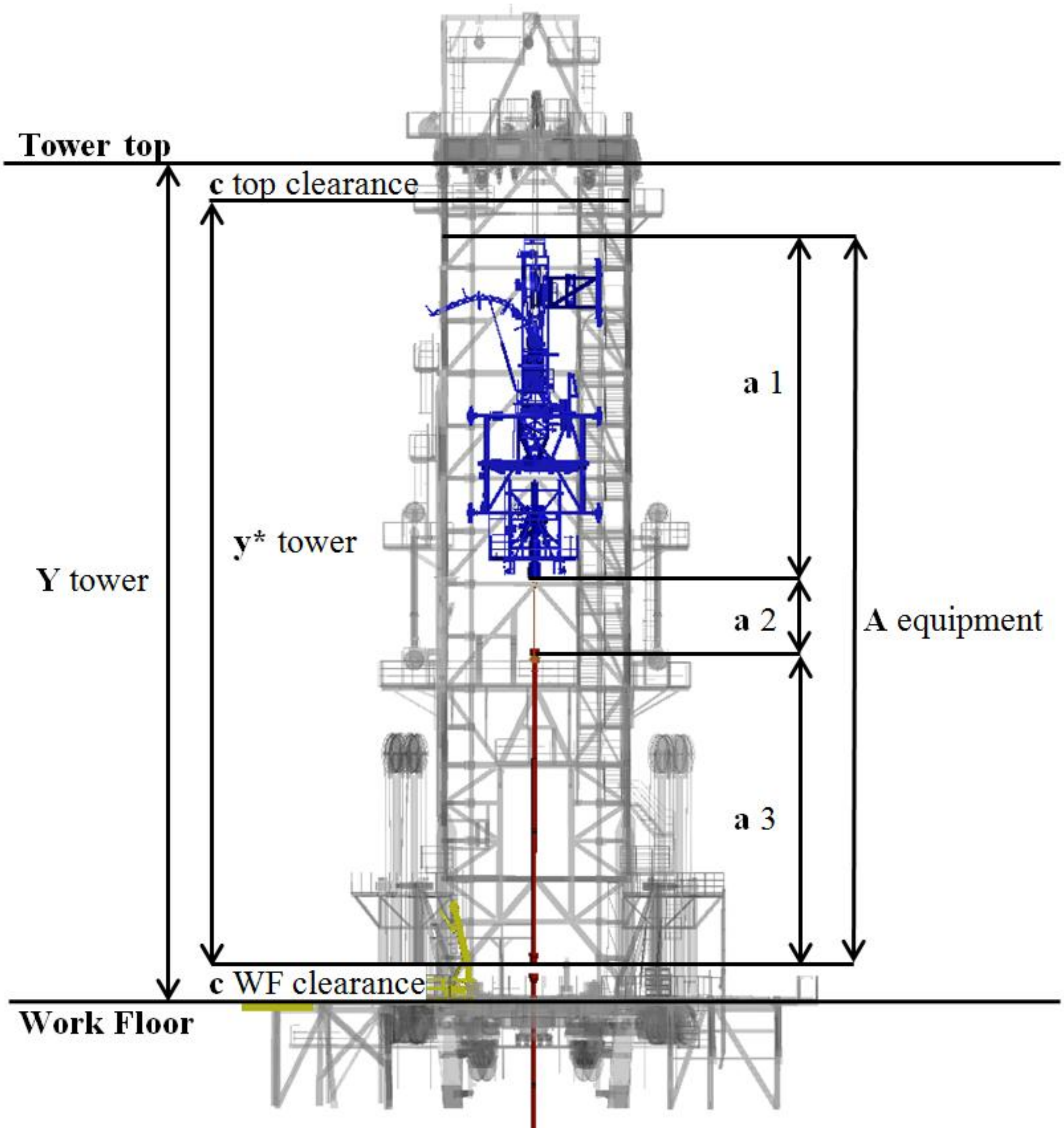


Figure 7: Inside tower lengths of equipment

A equipment:	*Overall equipment length.*	*36.9 m*
a-1:	*Total height of Dynaplex hook, CTTF and SFT combined*	*17.0 m*
a-2:	*Length of new handling tools*	*3.2 m*
a-3:	*Riser element length.*	*15.7 m*
C clearance:	*Total tower clearances.*	*~ 4.6 m*
c-top clearance:	*Clearance at the top beam (Including a safety margin).*	*~ 1.9 m*
c-WF clearance:	*Clearance at the work floor (Including a safety margin).*	*~ 2.7 m*
M-margin:	*Extra in-tower margin ($y^*_{tower} > A_{equipment}$).*	*~ 1.0 m*
Y tower:	*Total tower length.*	*41.5 m*
*y*tower:*	*In-tower handling space.*	*35.9 m*

Following relations are assumed:

$$Y_{tower} = \ y^*{}_{tower} + C_{clearances}$$

$$A_{equipment} = a_1 + a_2 + a_3$$

$$M_{margin} = \ y^*{}_{tower} - \ A_{equipment}$$

From these relations the maximum additional equipment length can be calculated. The maximum length is when the in the tower margin has its lower limit, i.e. the in-tower height is equal to the total length of the combined equipment minus the margin.

$$A_{equipment,\ max} = \ y^*{}_{tower} - \ M_{min}$$

$$\left(a_1 + a_{2,max} + a_3\right) = \left(Y_{tower} - c_{top\ clearance} - c_{WF\ clearance}\right) - M_{min}$$

$$a_{2,max} = Y_{tower} - c_{top\ clearance} - c_{WF\ clearance} - M_{min} - a_1 - a_3$$

The a2 max in tower length is measured from the SFT connection point to the riser element top. The actual equipment length would include the SFT-handling equipment overlap and the riser-handling equipment overlap. This indicates that the elevator height does not limit the system.

$$a_{2,\ actual} = a_{2,max} + overlap_{SFT-hanling\ equipment} + overlap_{riser-hanling\ equipment}$$

Introducing values:

The free inside height of the derrick is of approximately 41.5 metres between work floor and crown base. The top clearance includes the equipment and a half meter safety zone measuring in total 1.9 meters. The work floor clearance includes the spider jaw, upper riser neck and a safe working zone (1 meter). The total work floor clearance is 2.7 meters. The in-tower equipment measures approximately 17.0 meters and the standard 50 feet riser joint is 15.67 meters. This result in a maximum in-tower length for the handling equipment of:

$$a_{2,max} = 41.5 - 1.9 - 2.7 - 1.0 - 17.0 - 15.7$$

$$a_{2,max} \approx 3.2\ meters$$

This emphasizes the strong restriction in equipment length.

The net in-tower handling height without surface equipment is approximately 30 meters from work floor to the elevator. The surface equipment builds 6.4 meters in the tower. With additional handling equipment measuring 3.9 meters the net in-tower handling zone is reduced to a total of 19.7 meters.

3.4 Potential Time Savings

The aim is to establish a riser handling system that reduces rigging time. A lot of work is associated with the rigging operation of Upper Riser Pack (URP). The idea is to store the Coil Tubing Tension Frame and Surface Flow Tree in the tower while disconnecting riser elements. This eliminates the need of rigging down the Upper Riser Package. The total time saved, is the time used to rig up and down the URP, minus the time used to install the new handling system. The rest of the procedure is the same.

$$Time_{saved} = Time_{rigging\ up/down, URP} - Time_{rigging\ up/down,\ \ handing\ tool}$$

The rigging time is very weather dependent and would vary from locations. An estimate for minimum rigging time is provided by Henrik Vedeld.

Rigging the surface stack in and out of the tower is estimated to take approximately 12 x12 hours for the URP, i.e. a total of 24 hour. Connecting and disconnecting the new handling tool to the surface stack might take 2-4 hour. This indicates that the total time savings would range from 20-22 hours. Note that this is best case scenarios so the savings may be much more.

Day rate for operation, including crew costs, are estimated to be 600 000 dollars. Whit the conservative estimate for potential time saving of 20-22 hours, with an hour rate of 25 000 dollar, the total savings is between 500 -550 thousand dollars per trip. In Norwegian kroners (NOK)[12] this equals 2.8-3.1 million NOK per trip. In a yearly basis, with an average of 19 trips, this increases to 50-60 million Norwegian kroners saved per year.

In addition not rigging down the Upper Riser Pack will reduce risk for damaging the equipment. This is also a big advantage with the suggested solution.

[12] 1 dollar = 5.6 NOK (cf. 20.12.12)

Chapter 4: Commercial solutions

This chapter presents some different handling solutions available on the market. There are many good solutions, but not all has the functionality that is desired. Information on functionality, limitations and interface is needed when electing a suitable concept for the riser handling equipment.

4.1 Design factors

There are many different hoisting solutions available on the market, but only a few meet all desired requirements for the handling system. Requirements and restrictions taken into consideration are strength, capacity, space, interface, functionality, and offshore certificates.

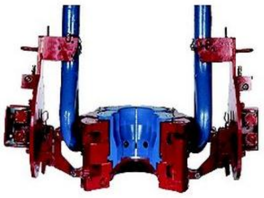
The new handling system should not limit the existing hoisting system either on capacity or space. This means that the new equipment is rated with approximately the same load as the main handling system, i.e. a capacity of 450 mT. The riser elements are deployed and retrieved through the moonpool. This part of the ship has high space limitations. A lifting device using the existing main winch would have an advantage since it does not reduce the available deck area. A disadvantage would be to meet all the interface challenges. Interface is the limiting conditions between new and existing equipment. This implies a special focus on mounting, collisions and manoeuvrability in the tower area. The functionality of the handling tool should also be in focus. The outer diameter (OD) of the riser elements used on Skandi Aker range from 9.5” to 10.5”. The tool should be able to grab and hold the entire riser stack in a safe manner. It should also be easy to manoeuvre a single riser joint and place it in the Catwalk Machine trolley. A rotating function is therefore needed in addition to adequate clearance between work floor and existing equipment. The last restriction is that it should be designed for the rough offshore environment. All this should be considered when searching for commercial concepts.

4.2 Commercial solutions

It is differed between active and passive equipment when searching for commercial solutions. There are many *active* lifting systems as cranes available on the market, but few that meet the space limitations in the moonpool area. This reduces the search to different *passive* “grip and hold systems” that relay on the existing hoisting system. A passive system has to be connected to the existing hoisting able to use this system. Modification of the existing equipment has to be evaluated.

A “grip and hold system” could be different types of elevators and clamps. Elevators are a clamp like tool which may be used as heavy lifting equipment. There are many suppliers of elevators, but National Oilwell Varco (NOV) is the supplier of all the existing equipment in the tower, so they are a natural choice when searching for a suitable solution. Table 1 gives tree different systems offered by NOV. The centre latch system consists of a gripper divided in two equal parts, split at the centre line. There exist split systems where parts can be completely separated, or the more common system where they are connected at one side. Other elevators have a main part where the pad eyes are mounted using one or two doors when closing the elevator. The operating systems are either; manually, air, or hydraulic driven.

Table 1: Elevators supplied by NOV (10)

Air or manually operated		**Hydraulic operated**
Centre latch system	**Side door system**	**Three part system**
		
100-500 sT capacity	65-500 sT capacity	500-750 sT capacity
pipe sizes: 2-3/8" to 6-5/8" OD	pipe sizes: 1.660" to 30" OD	pipe sizes: 3-1/2" to 11" OD

To connect the elevator to the existing system different connecting arms can be used. Connecting arms could be links, chains, or wires. The offshore market is dominated by links followed by chain slings. Wire solutions are not that common in this load range. The most common solution when using an elevator system is two links with two pad eyes.

Table 2: Illustration of different connecting arms alternatives available on the market

Single	**Double**	**Multiple**	**Multiple**
Link	Links	Chain sling	Wire sling
			

The combination of elevators and links are called a yoke. This is a commonly used system in the offshore industry. Skandi Aker has a NOV yoke mounted in the tower, and the new system could resemble this lifting solution. An illustration of the existing equipment can be found in Appendix 1.

4.3 Selecting a commercial alternative

A yoke is a passive lifting system using the main winch for elevating the riser stack. Any other system, as cranes or additional winch systems would demand storage space around the moonpool area and is thereby not good alternatives for the new handling system. A yoke consist of an elevator and usually two arms. One arm systems are flexible but lack of redundancy and are not commonly used. Two arm solutions have good flexibility and manoeuvrability. Multiple arm solutions have the highest redundancy, but multiple connection points on the elevator would limit handling. The suggested system is therefore a yoke system with two connecting arms.

The elevator could be a centre latch, side door, or three-part system. The split system is not available in the desired load range. The side door system is assumed to be stronger than the centre latch elevator since the pad-eyes are connected on one single piece. The side door systems are manually or air operated. The three-part system has eyes mounted on a single piece, providing high lifting capacities. The two door system is easy to operate and is available with a hydraulic operating system providing safe operation.

NOV has delivered all the existing equipment in the tower and a NOV hydraulic lifting yoke is mounted as a part of the existing lifting equipment. This means that there already exist routines and procedure for such lifting appliances which reduced work and potentially spare parts. This makes NOV a natural choice when selecting commensal equipment, since the equipment is well proven both by crew and engineering team. Another advantage is NOVs knowledge of the existing tower equipment and AKOFS requirements and routines. One vendor systems are favourable since there less people to contact if errors occur.

NOV has an elevator-series, the BX-series, which is a good alternative for the new riser handling system. The BX3 has too low load capacity and the BX4 cannot handle riser elements with greater diameter than 9-3/4 inches. The BX5 has a capacity of 500 sT[13] (453.6 mT) and an upper limit of 11 inches OD. The BX7 has a capacity of more than 1000 sT (907.2 mT) which is not needed.

The links provided by NOV have two pad-eye configuration. The size of the upper and lower pad-eye is custom made for the different purposes, and the length can be adjusted to fit the system. They are weld-less which gives high strength capability.

The recommended commercial alternative for the handling equipment is a passive lifting appliance connected to the main hoisting winch. The system should be mounted straight underneath the equipment in the tower and centred over the moonpool. The BX5 elevator system provided by NOV is a good alternative. This system is a hydraulic operated elevator with a capacity of 500 sT. The hydraulic operating system gives the elevator additional tilt functionality and safe handling which is desirable during operation.

[13] Short ton: 1 sT =0.90718474 mT

Chapter 5: Design specification

The purpose of this chapter is to outline the design requirements and to state the load case. The first section is the design specification, while the second part gives a brief description of relevant rules and regulations included in the design specification. The third and final part uses the design specification together with the relevant rules and regulations to define the load case.

A design specification is a document that *establishes the scope of work for an engineering study* (11). Following may be included in the design specification:

- *Scope*
- *Schedule*
- *Standards*
- *Operation Condition*
- *Design Requirements*
- *General specification*
- *Painting specification*
- *Welding specification*
- *Safety measures*
- *Inspection & Test specification*
- *Documentation (including approved drawings)*
- *Responsibilities*

The scope is a short description of the overall design goal. A schedule is important to ensure progress in the design process and to set a deadline / delivery date. Too make sure that the equipment is designed for safe operation, all components must follow the applicable requirements stated in the latest edition of the standards. Relevant standards should be listed, but details should not be included. Operation condition should be listed to make sure that all equipment is designed for the same environmental state. The design requirements include the desired functionality of the equipment, while the general specification state all additional requirements such as desired drawings, material selection, calculations, safety measures, protection, control systems, classification descriptions, approvals etc. Paint, welding, inspections & testing, safety measures, and desired documentation are often included in own sections. If tooling is ordered, the design specification is sent to a vendor and some additional information should be included. Vendor and manufacture documents, together with description of how responsibilities are divided between company and vendor, are examples of additional information.

5.1 Design Specification

Following information is included in the design specification:

- *Scope*
- *Standards*
- *Operation Condition*
- *Design Requirements*
- *General Specification*
- *Painting & Welding specification*
- *Documentation*

Scope

Provide a new handling system for Aker Oilfield Services well intervention unit (WIU). The system should not limit the derrick capacity of 450 metric tonnes. It should maintain safe handling of riser elements in and out of the tower, with the same operating speed as the original handling equipment.

Standards

Aker Oilfield Services follows the rules and regulations provided by *Det Norske Veritas* (DNV). Relevant for the equipment at hand is rules and regulations for lifting and hoisting equipment for drilling plants. The DNV includes the specifications given by the *American Petroleum Institute* (API). Following standards is therefore to be used:

- *DNV-OS-E101*
- *API specification 8C*

Operation Condition

- The equipment will be installed in derrick.
- The equipment will be operating in ambient temperatures ranging from -20 to 40^{o}C.
- Main operation condition is a significant wave height of 2 meters.

Design Requirements

The equipment should consist of one adapter and one hydraulic lifting yoke. The adapter should be connected between the SFT and the lifting yoke. The hydraulic lifting yoke consist of two links and one elevator.

Overall Handling System:

- Capacity of 450 Metric Tonnes.
- Ability to grip and hold the riser system while elevated.
- Ability to handle a disconnected riser element from vertical to horizontal position
- Ability to place riser element on Catwalk Machine Trolley.

Elevator:

- Gripper capacity 450 Metric Tonnes.
- Ability to grip and hold risers with outer diameter from 241.3 mm to 266.7 mm.
- Ability to rotate riser elements from vertical to horizontal position.

Links:

- Link capacity of 225 Metric Tonnes (per element).
- Connect elevator to adapter.
- Give handling clearance between SFT and riser element.
- Shall be produced in pairs.

Adapter:

- Adapter capacity of 450 Metric Tonnes.
- Connect yoke to SFT without damaging existing equipment.

General Specifications

Material:

- Adapter, Links and Elevator: 551 MPa steel.
- Bolts: 640 MPa steel

Drawings:

- Drawings that state the overall dimensions of the equipment.

Calculations:

- Strength analysis (Computer Aided Design programs may be used).
- Hand Calculations verifying the CAD analysis.

Any requirements not stated but necessary for safe and reliable operation of the equipment, shall be included.

Painting & Welding

In addition to the stated standards all painting should follow the NORSOK standards while all welds should be welded in accordance to the codes and standards of the American Welding Society.

Documentation

The documentation needed should be according to *TWWW-DD-0004-Final Documentation Procedure* and *TWWW-DI-0005-Engineering Numbering System.*

5.2 Relevant Rules and Regulations

The design specification states that the equipment should follow the DNV standards for drilling plant equipment on well intervention facilities, which can be found in *DNV-OS-E101*. The DNV standard refers to two other documents; *DNV Standard for Certification No. 2.22 Lifting Appliances* and *API*[14] *Specification 8C.* API standards are often more conservative than DNV, and are often used by AKOFS when dimensioning new equipment. Following are a short description of relevant rules applicable for the new handling system, divided after DNV and API specifications.

5.2.1 DNV-specification

The governing rules for the equipment at hand are the Offshore Standard (OS) from DNV[15]. This section is a short description of relevant requirements from the *DNV-OS-E101* regulation including *No. 2.22 Lifting Appliances*.

5.2.1.1 Design Load Conditions

DNV requires that load combinations for different operation and non-operation conditions shall be evaluated. Appropriate loads due to operation and environment shall be evaluated for each case.

Load cases:
Following operation and non-operation conditions shall be evaluated:

- *Operational*
- *Waiting on weather*
- *Survival*
- *Transit*
- *Accidental heel*

Operational loads:
There are four load contributions that have to be accounted for; *principal loads, vertical loads, horizontal loads,* and *accidental loads*. Principal loads are loads due to weight which, in normal sense, always acts vertically ((12) B202). This includes loads due to deadweight, working load and pre-stressing. If the component experiences heel or trim the principal loads should be included with vertical and horizontal components. Pre-stressing is not an issue on the components at hand. *Vertical loads* are associated with unit motion and should be accounted for by multiplying the work load with a dynamic coefficient ψ. The minimum value is stated or can be calculated according to *No2.22-B304* (12). The *Horizontal loads* are also due to operational motions. It may be differed between equipment motion and unit/ship motion. Equipment motions are often associated with maximum acceleration when elevation is started or stopped. *Accidental loads* are all additional loads due to accidents as collisions, accidental heel etc.

[14] API: American Petroleum Institute
[15] The DNV Rules and Standards are freely available online.

Environmental loads:

DNV states that the *unit motion, wind loads, air temperature and humidity, ice and snow loads* are aspects that are to be evaluated when establishing the environmental loads. The unit motion is the environmental loads case that restrains the equipment. Detailed accelerations for the ship may be found in Appendix 3. The wind load is mainly for exposed equipment and will in this case not give any relevant contribution. Wind induced loading is negligible comparable with the hydrodynamic loads. The design temperature is as stated in the design specification and varies between -20 to 40 ^{o}C. No high temperature fluids will be in contact with the tool. Temperature deteriorating is therefore not considered in this analysis, as well as humidity, which does not alter the load case. The well intervention unit operates around the vest cost of Africa and Brazil. This makes the snow and ice loading negligible.

Unit motions are accounted for by combinations of accelerations. The maximum value of the following three combinations shall be used for calculating the worst possible load scenario for the unit motion, as described in DNV-OS-E101 ((13) H305.4).

- $Heave_{max} + \sqrt{(Roll_{max})^2 + (Pitch_{max})^2}$

Load testing:

Test load for lifting appliances with Safe Working Load (SWL) exciding 50 tonnes, should be 1.1 times the SWL (table D1 at (13) D300). DNV defines SWL to be the maximum available mass to be lifted. This load test will not be applied due to API 8C's more conservative recommendation.

5.2.1.2 Equipment class

The rules and guidelines differ between what type of crane and operation the equipment are intended for. According to No 2.22 the crane is a *heavy lift offshore crane*, due to its capacity and is intended for loading and discharging from the seabed. Any crane with a Safe Working Load above 200 mT is classified as heavy lifting appliance.

The equipment is also classified as *loose lifting gear*. Loose lifting gear is any load carrying accessory with a lifting appliance which is used to attach the useful load to the main hook. The equipment is not a part of the permanent arrangement, and may be stored separately from the crane.

5.2.1.3 Design Calculations

Following modes of failure shall be evaluated:

- *Extensive yielding*
- *Structural stability*
- *Fatigue Fracture*

The *extensive yielding* analysis shall be based on elastic theory. When evaluating with respect to excessive yielding, equivalent stresses shall be calculated according to von Mises yield criterion. Plastic theory may be used where appropriate (ultimate strength). The *structural stability* analysis (buckling) shall be according to generally accepted theories. The equipment at hand is not exposed to buckling loads, and this analysis is omitted. The *fatigue analysis* shall include areas of the mechanical component that are suspected to be damaged by fatigue. The analysis shall be based on a time period of no less than 20 years with a representative load spectrum for the occurring loads. The fatigue strength is expressed as the critical amplitude of alternating stresses. The load basis is normal operation and the critical amplitude is defined as the value corresponding to 90% probability of survival. Calculated maximum stress amplitude shall not be greater than the critical stress amplitude divided by a safety factor of 1.33.

- $\sigma_{allow} = \sigma_{cyc}/1.33$

The permissible stress for elastic analysis is 1) for normal condition and 2) for cranes subject to exceptional loadings, where σ_y is the material yield strength.

1. $\sigma_{allow} = \sigma_y/1.50$
2. $\sigma_{allow} = \sigma_y/1.10$

In addition, the following is stated in the OS-E101 I 203 (13):

The yield strength used in calculations shall not exceed 0.85 of the specified minimum tensile strength.

5.2.1.3 Load and Safety Factors

Dynamic factor:

The Dynamic Amplification Factor (DAF) for an offshore lifting appliance normally refers to still water condition, but it may also refer to a specific significant wave height. The coefficient is a variable factor representing the dynamic effects that the working load is subject to. The dynamic factor is associated to the unit motion and the stiffness in the system. It can be calculated from the following formula:

- $\psi = 1 + V_R\sqrt{\frac{C}{W \cdot g}}$

C = Geometric stiffness i.e. spring constant.
g = Acceleration of gravity (9.81 m/s^2)
W = Working load (static weight of load lifted plus weight of accessories)
V_R = Relative velocity

The spring constant of the system is difficult to assess with accuracy. Conservative dynamic coefficient is selected form No2.22 B304 vertical loads. The assumed work load is the SWL 450 mT plus accessories ~1 mT (conservatively); $W = (450 + 1\,)mT * 9{,}81\ m/s^2 \approx 4400\ kN.$

For offshore cranes with work load between 2500 kN and 5000 kN the dynamic factor is an linear interpolated value between 1.3 and 1.1. In this case the DAF is set to 1.15 (12) B304.

Safety factors:
When calculating the strength of the equipment, safety factors shall be applied. The Safety factor (S_F) is the load factor (γ_f) multiplied with the material safety factor (γ_m).

- $S_F = \gamma_f \cdot \gamma_m$

The verification of safety may be based on two methods; *the permissible stress method* or *the limit state method.* The methods are equivalent in all cases when the selected safety factors in table D1, specification No. 2.22 is applied to the load.

Service Limit State:
According to Table D1 (No2.22 Ch2 Sec: D (12)) some factors for load and materials shall be applied in the service limit state.

- *Load factor of 1.3*
- *Material factor of 1.15 for elastic analysis.*
- *Material factor of 1.3 for plastic analysis.*

Accidental Limit State:
According to Table D1 (No2.22 Ch2 Sec: D (12)) following coefficients shall be applied in the accidental limit state:

- *Load factor 0.96*
- *Material factor of 1.15 for elastic analysis.*
- *Material factor of 1.3 for plastic analysis.*

5.2.2 API specification 8C

API Specification for *drilling and production hoisting equipment* covers among others connectors and link adapters (d), elevator links (g), and casing, tubing, drill pipe, and drill collar elevators (h) ((14) chapter 1.2). This indicates that all the equipment at hand is included in this specification. The API 8C has a more detailed design specifications than the DNV provides. Following functional and operational requirement is stated in the API 8C:

'Hosting equipment shall be designed, manufactured, and tested such that it is in every respect fit for its intended purpose. The equipment shall safely transfer the load it is intended for. The equipment shall be designed for simple and safe operation.' *[(14); 1.3]*

The *Design Load* (L_D) is the static load (L_s) and the dynamic load (L_d) that would cause maximum allowable stress in the component. The dynamic contribution is due to acceleration effects on the equipment. The *Rated Load (R)* is numerically equivalent to the design load, while the *Safe Working Load* (SWL) is equivalent to the design load minus the dynamic load i.e. equivalent to the static load component under maximum allowable stress condition. ((14); 3)

5.2.2.1 Strength analysis

API 8C states that the equipment design analysis shall address *excessive yielding, fatigue, and buckling* as possible modes of failure. These are the same as in the DNV standard. For each cross section considered in the analysis, the most unfavourable *combination*, *position*, and *direction* of forces are used ((14); 4.2.2)

The fatigue and stability analysis shall be according to generally accepted theories which are in compliance with the DNV requirements.

The <u>strength analysis</u> shall be based on elastic theory. The nominal equivalent stress caused by the design load shall not exceed the maximum allowable stress criteria defined by:

$$Maximum\ Allowable\ Stress = \frac{Specified\ Yeld\ Strength}{Design\ Safety\ Factor}$$

The nominal equivalent stress is defined according to Von Mises-Hencky Theory:

$$\sigma_e = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_1\sigma_3}$$

$$\sigma_e = \frac{\sqrt{2}}{2}\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2}$$

Where:

σ_e = *flow stress in tension*

$\sigma_1, \sigma_2, \sigma_3$ = *true principal stresses in combined loading*

In contact areas and in areas plastic analysis is permitted. The ultimate strength analysis states that the equivalent stress shall not exceed the maximum allowable stress criteria defined by:

$$Maximum\ Allowable\ Stress = \frac{Specified\ Ultimate\ Strength}{Design\ Safety\ Factor}$$

5.2.2.2 Load Rating

Rated load or the design load is based on the design safety factor, yield strength of material used, and the stress distribution. The Design Safety Factor is calculated according to the load rating:

Table 3: Design Safety Factor form API 8C

Load Rating [mT]	**Value:**
136 and less	3.00
136 to 454	$SF_D = 300 - \frac{0.75(R - 150)}{350}$
454 and more	2.25

5.2.2.3 Load Testing

The equipment should be tested with the design load. Strain gauge shall be applied on critical areas (three element strain gauge are recommended). The test load shall be calculated as follows, but never set less than 2R.

$$test\ load = 0.8 \cdot R \cdot SF_D, \qquad if \quad 0.8 \cdot SF_D > 2$$
$$test\ load = 2 \cdot R, \qquad if \quad 0.8 \cdot SF_D \leq 2$$

The load should be applied carefully until it reaches the test load, to prevent dynamic amplification. In addition to this test, a proof load test shall be performed. A test load of 1.5 R shall be applied and held for a period of no less than 5 minutes. Control of tested equipment shall be done.

5.2.2.4 Contact surface radii

Table 4 gives API recommendations for the radii of a load rating between 454.9-591 mT.

Table 4: Contact surface radii from API regulations table 9B

Component	Radii type (Section/ Surface)	API mark (figure)	Radii [millimetres]	Max/Min Limitation
Adapter eye	Cross section radii	C1	101.6	Max
	Outer surface radii	D1	57.15	Min
Link upper eye	Outer surface radii	C2	120.65	Min
	Cross section radii	D2	47.63	Max
Link lower eye	Cross section radii	G1	57.15	Max
	Outer surface radii	H1	127.00	Min
Elevator eye	Outer surface radii	G2	60.32	Min
	Cross section radii	H2	127.00	Max

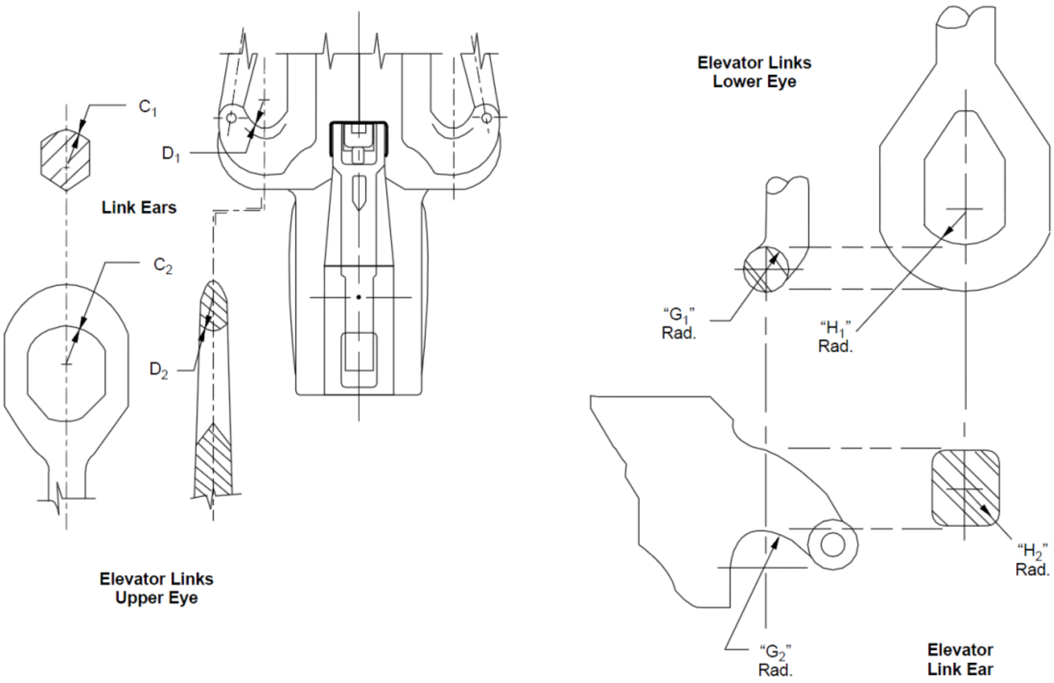


Figure 8: Contact surface radii for adapter, link and elevator eyes from API 8C figure 8.

5.2.3 Acceptance Criteria

DNV acceptance criteria:

- *The yield strength used in calculations shall not exceed 0.85 of the specified minimum tensile strength. [E101; I 203]*
- *Load test of minimum 1.1 times the design load (greatest of SWL and* $0.75 \cdot \psi \cdot SWL$*) [E101; D 300]*
- *Calculated maximum stress amplitude shall not exceed the permissible stress amplitude. [No2.22; D 403]*
- *Fatigue analysis shall be based on a time period of no less than 20 years with 90% probability of survival. Calculated maximum stress amplitude shall not be greater than the critical stress amplitude divided by a safety factor of 1.33. [No2.22; D 400]*

API acceptance criteria:

- *The maximum working stress associated with the design load is ≤ maximum allowable stress [API; 4.7.3]*
- *For the purpose of calculations involving shear, the ratio of yield strength in shear to yield strength in tension shall be 0.58.* $\left(1/\sqrt{3}\right)$ *[API; 4.8]*
- *Localized yielding shall be permitted at areas of contact.*
- *Load test of minimum 2 times rated load or* $0.8 \cdot R \cdot SF_D$*, whichever is greatest.*
- *In a unit that has been load tested, the critical permanent deformation determined by strain gauges or other suitable means shall not exceed 0.002 inch per inch.*

5.3 Load Cases

Each part of the drilling plant is designed to operate safely under the different load conditions expected during operation. The design conditions normally evaluated are operation, waiting on weather (WOW), transit, survival and accidental.

During normal operation the yoke is a part of the cursor frame. This operation is in calm weather with a significant wave height limit of 2 m. This limit is due to wave forces in the moonpool. If the wave height exceeds this limit, the vessel is set to WOW condition.

In WOW, transit, and survival the loads are expected to exceed the tower capacity limit of 450 mT. Additional lifting appliances are used to carry parts of the load in these cases, such as the spider and the tensioner system. For extreme weather conditions the storm hang-off are used to secure the riser stack. WOW, transit, and extreme weather are not furthered considered in this case study since the tower load condition is controlled by other safety systems.

Accidental condition could be collision, wrong operation or accidental heel. Colliding with the tower top or work floor could lead to deformations of the Coil Tubing Tension Frame or the Surface Flow Tree frame. The accidental loads are assumed to be relatively small compared to a subsea collision with weak link break. The last accidental case is accidental heel. Heel would create huge bending moments at the connection point between the new equipment and the SFT. Accidental heel and weak link break are further evaluated.

In addition to the DNV conditions an API case are evaluated. This case is uses the conservative API Design Safety Factor of 2.25. This load case is included so that the equipment is in accordance with the American Petroleum Institute Standards.

Following are a summary table presenting the load cases for the new handling equipment. Details around each case are presented in the next section, and calculations can be found in Appendix 3.

Table 5: Load Cases summary table

Load Case	***Safety factor/ Design factor***	***Total Design Force***
Operation Limit	$1.3 \cdot 1.15 = 1.5$	*7 540 kN*
Accidental heel	$0.96 \cdot 1.15 = 1.1$	*5 529 kN*
Drift off with subsea collision	$0.96 \cdot 1.15 = 1.1$	*7 982 kN*
API-load	$API\ SF_D = 2.25$	*11 310 kN*

NOTE: The loads presented here include both the load safety factor and the material factor. In calculation and simulation, load factor is added to the load, while the allowable limit is set by dividing the yield strength by the material factor.

5.3.1 Load Case Calculations

DNV requires that load combinations for operation and non-operation conditions shall be evaluated. The design load is the static and the dynamic load that would cause maximum allowable stress in the component (14). The static contribution is from the working load i.e. SWL and a DAF of 1.15 for still water is already selected. The dynamic contribution is due to acceleration effects on the equipment and riser stack associated with the operation condition.

Forces from the riser stack can be divided into mass forces and drag forces. The riser stack is approximately 3000 meters long. The water would reduce the movements in transversal and lognitudal direction, while movement in the vertical direction would be somewhat equal over the entire length. The worst acceleration combination would only be multiplied with the upper part of the riser stack, while the lower part is assumed to only have acceleration in the vertical direction. The total dynamic load can be calculated as follows:

$$F_{dynamic,mass\ 2Hs} = m_{top\ stack} \cdot a_{\max,\,2Hs} + m_{riser\ stack} \cdot a_{vertical,2Hs}$$

The mass of the top stack is the mass of the new handling tool and some of the upper riser elements conservatively set to a total mass of 5 mT. The mass of the riser stack is the SWL of the new handling tool set equal to $SWL_{tower} - W_{tower\ existing\ equipment}$. SWL for the existing tower equipment is 450 mT and the weight is assumed to be approximately 30 mT leaving 420 mT available for the riser stack and the lower riser package. The maximum DNV top accelerations are calculated to be 2.06 m/s^2 where 0.61 m/s^2 is in vertical direction.

$$F_{dynamic,mass\ 2Hs} = (5\ mT \cdot 2.06\ m/s^2) + (420\ mT \cdot 0.61\ m/s^2)$$

$$F_{dynamic,mass} = 266.5\ kN$$

Rough calculations show that the resistance force in axial direction due to skin friction between water and riser surface are small compared to the other forces and are negligible.

Increasing the DAF associated with still water condition to include operation condition for significant wave height of 2 meters indicates an increase from 1.15 to 1.215. SWL for existing elevator and hook systems is 450 tonnes, and the rated load is 550 mT (15). This indicates a DAF of 1.2223 which is close to the DAF calculated here. The DAF and the safety factors (SF) given from DNV and the DAF calculated here are summarized in the following table.

Table 6: Summary of the load and amplification factors presented in section 5.2.1.3

Dynamic Amplification Factors	***Value***
DAF – 2Hs	*1.22*
Safety Factors	
Load Factor – Service Limit	*1.3*
Load Factor – Accidental Limit	*0.96*
Material Factor – Elastic Analysis	*1.15*
Material Factor – Plastic Analysis	*1.3*

Normal operation limit

The SWL for the new handling equipment is the SWL for existing elevator minus the weight of the in-tower equipment (CTTF and SFT stack = 30mT). The design load is the SWL multiplied with a DAF to include both static and dynamic contributions. The design load is multiplied with a safety factor that includes a 30% increase of the working load and a material safety factor of 1.15. The total design force is calculated as follows:

$$F_{design} = SWL \cdot DAF_{2HS} \cdot SF \cdot g$$

$$F_{d,normal} = (450 - 30)mT \cdot 1.22 \cdot 1.5 \cdot 9.81\, m/s^2 = 7540\, kN$$

Accidental heel

The accidental heel load is the operational load applied with an unfortunate angel expected during accidental heel. A critical situation during accidental heel is if the elevator is tilted so that one of the links loses contact with the elevator and the entire load are transferred through only one link. This is very conservative since the actual angle of the riser would be limited by the moonpool walls and the spider. Details can be found in Appendix 4.

Accidental load is based on normal operation but with a reduced safety factor due to the additional extreme condition. The total force through one link is:

$$F_{d,heel} = (450 - 30)mT \cdot 1.22 \cdot 1.1 \cdot 9.81\, m/s^2 = 5529\, kN$$

Drift-off with subsea collision

The vessel operates in a safe handling zone. Colliding with other subsea equipment is considered probable only in combination with loss off propulsion. The worst case scenario would be that the riser stack collides with subsea equipment so that the Coil Tubing frame hits the work floor and the weak link brakes. The weak link breaks for loads greater than 250 mT. The maximum force felt by the handling tool is at the point where the weak link breaks. The force is calculated as the normal operation load contribution with accidental load safety factors, plus the brake load of the weak link at 250 mT:

$$F_{accidental\ operation} = (450 - 30)mT \cdot 1.22 \cdot 1.1 \cdot 9.81\, m/s^2 = 5529\, kN$$

$$F_{break} = 250\, mT \cdot 9.81\, m/s^2 = 2453\, kN$$

$$F_{d,subsea\ collision} = 5529\, kN + 2453\, kN = 7982\, kN$$

API-load

The API-load is the conservative estimate taken from the API standard of the rated load, i.e. the design load, multiplied with a safety factor of 2.25. This is more conservative than DNV and will therefore be used. The stress acceptance level is up to yield.

$$R = (450 - 30)mT \cdot 1.22 = 512.4\, mT$$

$$F_{d,survival} = 512.4\, mT \cdot 2.25 \cdot 9.81\, m/s^2 = 11310\, kN$$

5.3.2 Material selection

There are two main types of material used for the new handling equipment. The same steel material is selected for the adapter, links and elevator assembly. The links need high strength and no weak points, so wielding is not considered for this part. The material selected is 80ksi equal 551 MPa. The second steel material is for the bolts, where load concentrations are expected. The links are cold drawn, i.e. stamped out of one piece of metal, and the adapter and elevator are forged.

Table 7: Material specifications

Component	Yield Strength [MPa]	Ultimate Tensile Strength [MPa]	E-modulus [Pa]	Poisson
Main steel	*551*	*690*	*2.07E11*	*0.3*
Bolts	*640*	*800*	*2.07E11*	*0.3*

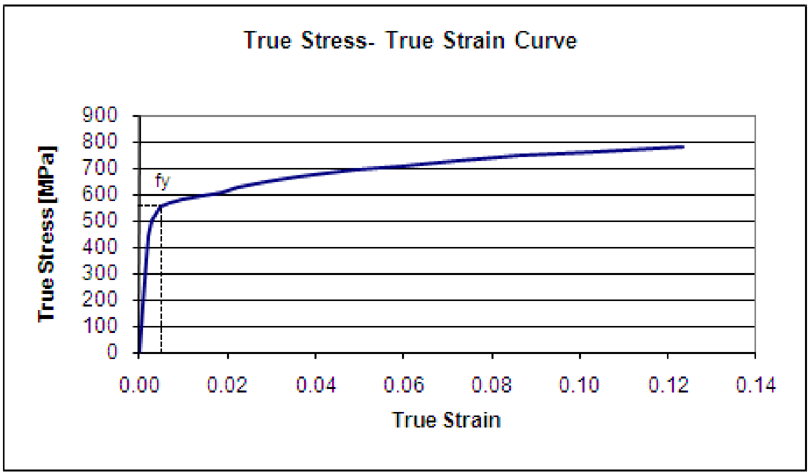


Figure 9: True Stress - True Strain curve links; 551

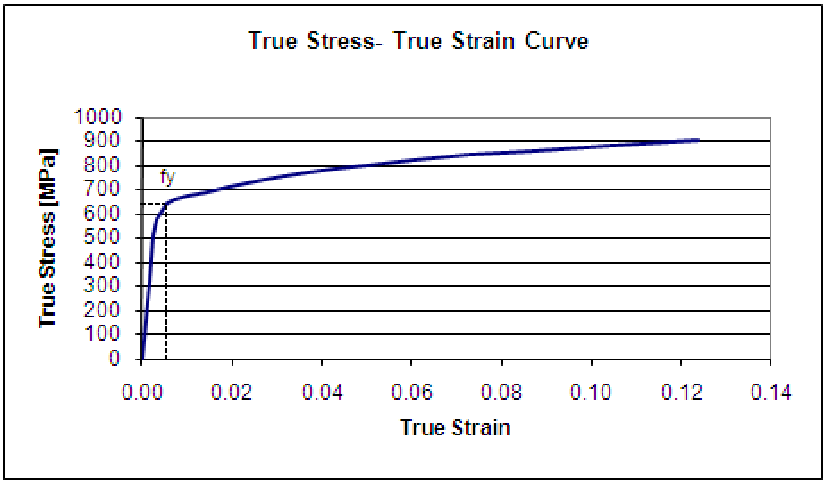


Figure 10: True Stress - True Strain curve bolts; 640

Chapter 6: Equipment design

It was concluded that a hydraulic lifting yoke provided by NOV is the best solution currently available on the market. Components drawn in this chapter would resemble NOV components by some key design factors. The key design resemblances are areas of contact and distance between contact points. Elevator eye dimensions would also be somewhat the same. If these design points are similar, then the overall load pattern would be similar and AKOFS stands freely to select the solution they want. The main difference between NOV's yoke and components designed here is the hydraulic parts. Components in this chapter are manually operated, and no hydraulics is needed. The design is in accordance with the design specifications stated in chapter 5.

After the design features are described some simplified hand calculations are computed to prove the strength and to verify the following SolidWorks analysis.

6.1 Component design

This chapter describes the design features. The desired functionality of the equipment was given in *chapter 5.1 Design Specification*, and governing rules and regulations are summarized in *chapter 5.2.3 Acceptance Criteria*. The combination of this is the background for the following design. The design features will be described thoroughly in the following subchapters.

- **Elevator assembly:**
    - Main part
    - Clamp assembly
    - Padding
- **Link assembly:**
    - Right and left link
- **Adapter assembly:**
    - Main part (including pad-eyes)
    - Adapter pin head

On the next page an illustration of the total assembly can be found. The elevator is a component designed to hold riser elements. The links are connection arms that give the necessary handling space between the Surface Flow Tree and the risers. The adapter is a component designed to connect the yoke and the Surface Flow Tree. This component is tailor made for this operation sequence and the existing equipment. It is designed so that no additional modifications have to be made on the existing equipment.

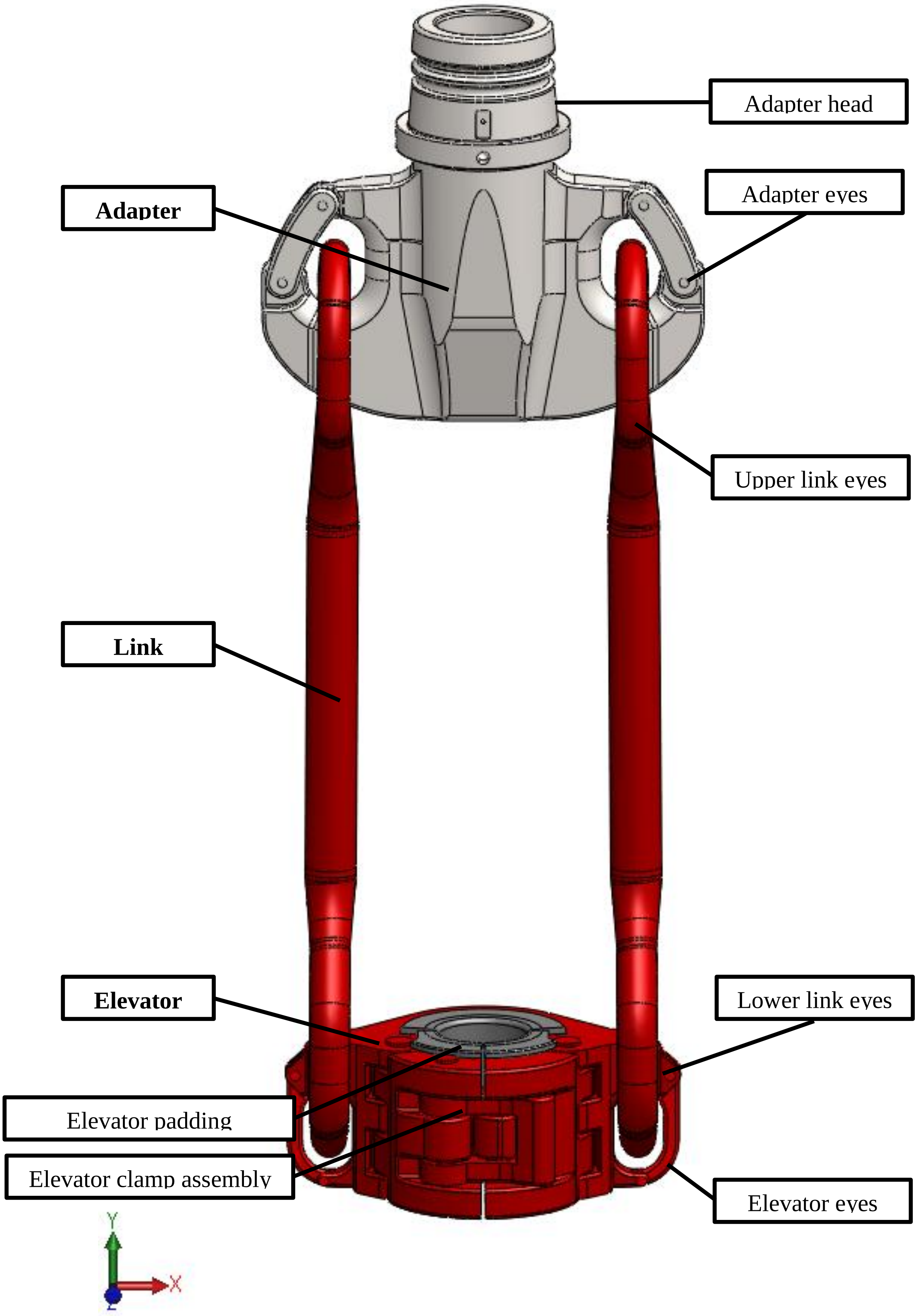


Figure 11: Total assembly with name tag on the different parts

6.1.1 Elevator

The elevator consists of several parts.

- **Elevator assembly:**
  - Main part
    - *Main elevator part (including elevator pad-eyes)*
    - *Elevator pad-eye lock pieces(x2) (incl. bolts)*
    - *Bolts(x2)*
  - Clamp assembly
    - *Right clamp*
    - *Left clamp*
    - *Lock piece (incl. bolt)*
  - Padding
    - *Main padding*
    - *Right clamp padding*
    - *Left clamp padding*

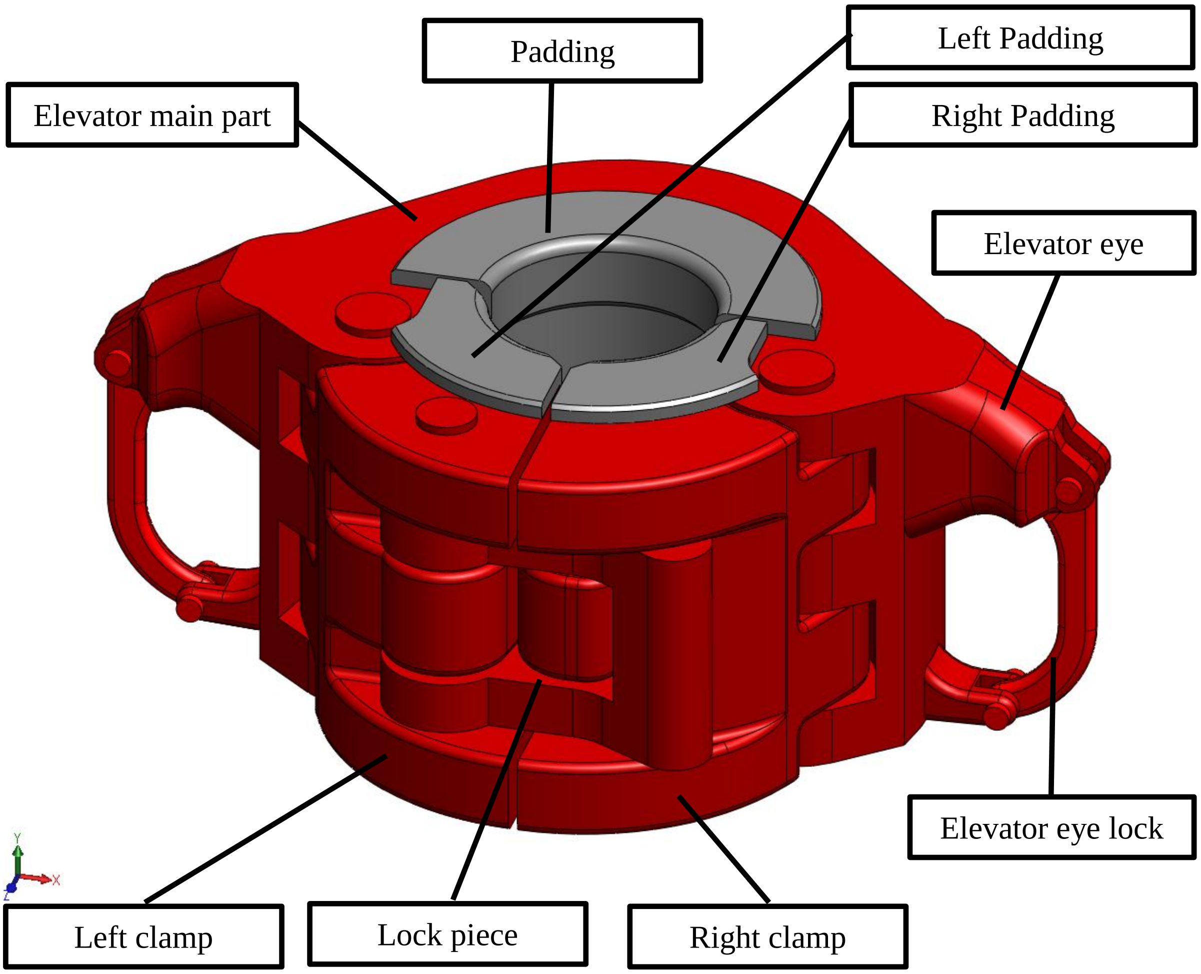


Figure 12: Elevator assembly with name tag on the different parts

The design specification states that the elevator should be able to grip and hold riser elements with diameters ranging from *241.3 mm to 266.7 mm.* The design specification also states the elevator should be able to rotate when placing a riser element onto the Catwalk trolley. The load case states that the load capacity should be 420 metric tonnes. This is the total information at hand prior the design.

NOV resemblance features

In chapter 3 it was concluded that a design resembling the existing tools would be favourable. The reason for having some design resemblance is to allow AKOFS to order the NOV components instead of manufacturing their own new equipment in addition to controlling the existing equipment. The design features of interest are the areas of contact and the distance between the contact points. The distance between the link eye connection points are *930 mm.* The contact surface radii are *70 mm* on the cross section and *70 mm* curve on the upper part of the pad-eye. This gives the additional design relations from the NOV elevator:

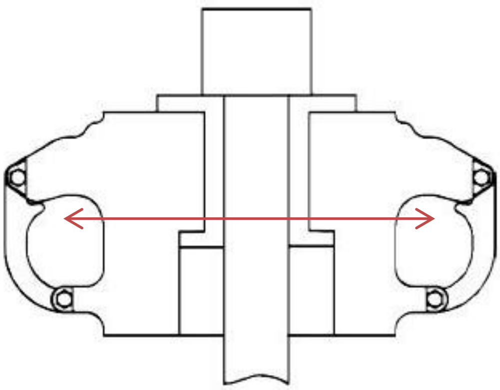


Figure 13: Cross section NOV elevator parts

Main elevator

The elevator is basically a circular pipe segment with pad-eyes. The inner diameter of the elevator should be changeable due to different riser diameters. The diameter range is set to 241.3 mm to 266.7 mm. Calculations show that it is difficult to make the rigid elevator changeable without making a large gap between the doors and potentially weakening the component. The problem is solved by introducing padding with different thicknesses.

Padding

The minimum padding thickness is assumed to be approximately 15 mm. The upper and lower part is a bit thicker than the middle section. The padding also covers parts of the elevator top. It has a guide flange that locks it into place on the elevator without introducing any bolts. The padding material is assumed to be a POM material (16).

The edges on the right and left protection cap are cut with a 40 degree angle so that it would not limit the total opening when the elevator doors are opened. When both elevator doors are open at 45 degrees a riser with 280 mm OD can be placed into the elevator.

Elevator door assembly

The suggested elevator has a two door system with a lock piece. The geometry can be seen in figure 14. The lock piece prevents the right and left clamp arms to open while lifting a riser element. It can only be opened when there is no outwards force on the inner clamp padding.

Elevator eyes

The elevator eyes transfer forces to the links. API has detailed recommendations for the different radiuses that are to be used on the eyes. Due to the importance of the contact areas, all radiuses are thoroughly described in *6.1.4 Contact surface radii*.

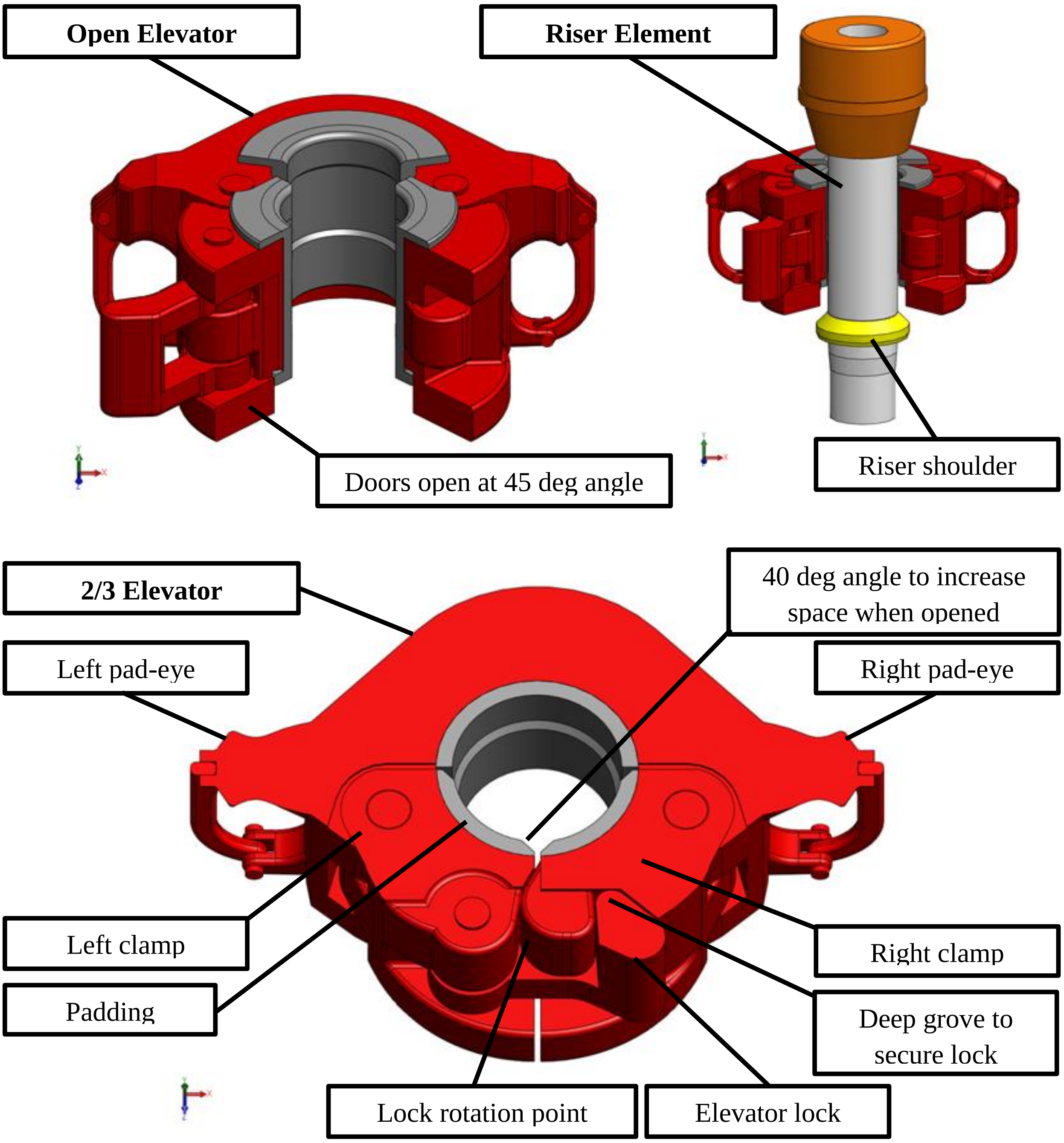


Figure 14: Elevator assembly with name tag on the different parts

6.1.2 Links

Links are basically extended connection arms. They are introduced to allow additional handling clearance. Links have the same functionality as wires except that they are not bendable. The design consists of a long circular metal piece with a hoop on each end called pad-eyes.

For heavy loads weld-less links are recommended. It is common that links in this load range are forged from one piece of alloy steel to provide sufficient tensile strength characteristics and light design. API state that links should always be made and used as pairs, i.e. it is not allowed to mix two different sets of "equal" links.

NOV provide high quality links with sufficient capacity. The recommendation is to order Varco BJ links with following design. This information, together with a discussion with Henrik Vedeld, results in following design for the links (see table 8). The Varco BJ links are dimensioned in inches, but the SolidWorks design is in millimetres.

NOV provides two alternatives for links. The existing links is illustrated in figure 15; straight configurations with equal upper and lower link eye. The lengths of the existing links installed in the tower are 10 feet. The new design should also have a straight configuration with equal outer dimensions for the upper end lower elevator link eyes. The radii and cross section radii are selected according to the API 8C specification. The total length is set to 2.44 meters (~8 feet) to ensure that there are enough handling space between the SFT frame and the work floor for the crew to maintain safe operation.

Evaluation of existing equipment gives following measurements and suggested design:

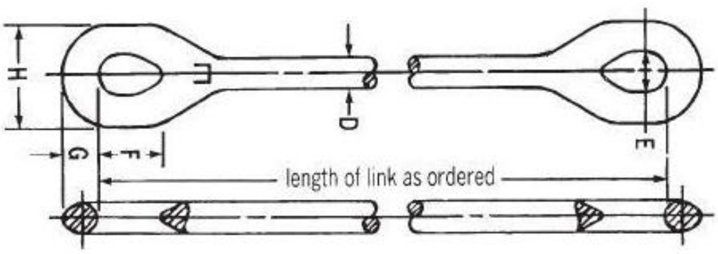


Figure 15: Installed NOV Varco BJ Links

Table 8: Dimensions installed NOV Varco BJ Links and new link design

		Varco BJ Links from NOV				*Design*	
Length	-	*10.0*	*feet*	*3048*	*mm*	*2440*	*mm*
Cross section diameter arm	*D*	*6.0*	*Inches*	*152.4*	*mm*	*150*	*Mm*
Inside width clearance link head	*E*	*10.0*	*Inches*	*254.0*	*mm*	*250*	*Mm*
Inside height clearance link head	*F*	*14.5*	*Inches*	*368.3*	*mm*	*370*	*Mm*
Cross section diameter link head	*G*	*7.5*	*Inches*	*190.5*	*mm*	*190*	*Mm*
Outer width link head	*H*	*23.0*	*Inches*	*584.2*	*mm*	*585*	*Mm*

Following is an illustration of the component drawn in SolidWorks including all dimensions in millimetres [mm].

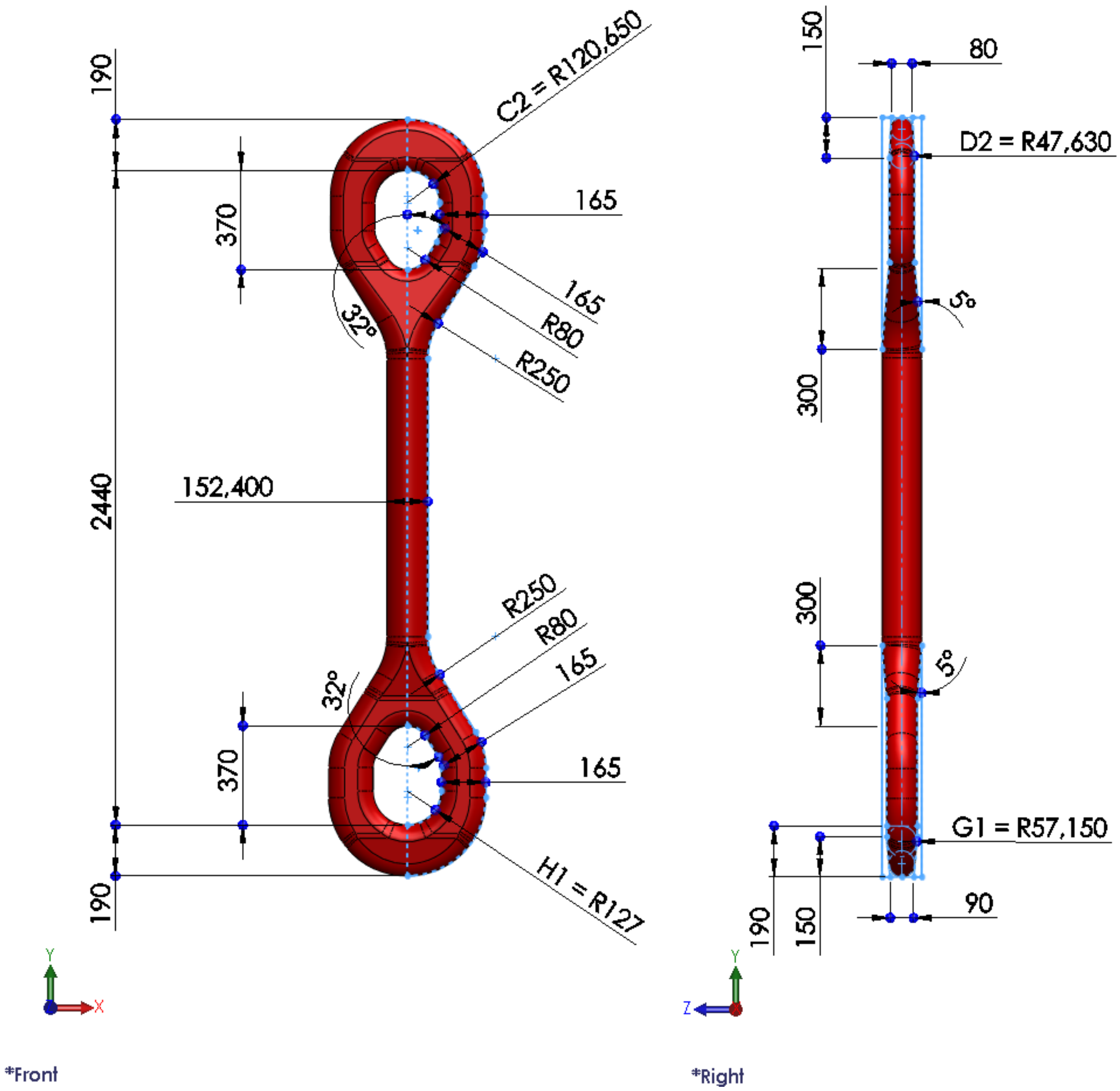


Figure 16: New links with dimensions

6.1.3 Adapter

The purpose of the adapter is to connect the yoke (elevator + links) to the Surface Flow Tree structure. It is important that the frame structure and the Surface Flow Tree (SFT) are not at risk of being damaged by the new equipment or by the new load cases. The frame is not strong enough to have the equipment mounted directly on it so the additional equipment is therefore connected to the SFT.

It is desired that changes on the existing equipment is kept on a low level. One reason is that changes may alter the SFT and could affect the strength or introduce a weakness. It would also lead to down-time when the SFT is modified. Welding eyes directly on the SFT is therefore not a desired solution. Using a clamp to fit the eyes would need a large surface area and the clamp may damage the SFT surface.

A simple solution is therefore to use the same connection as for the landing joint. The load case for connecting the landing joint and the adapter is assumed to be the same at small angels of heel. This means that no additional strength analysis has to be conducted for the SFT and the SFT frame. This assumption would also validate the locking mechanism and connection points. In addition the tooling is located beneath the SFT which reduces the risk of collision or surface damages on the SFT.

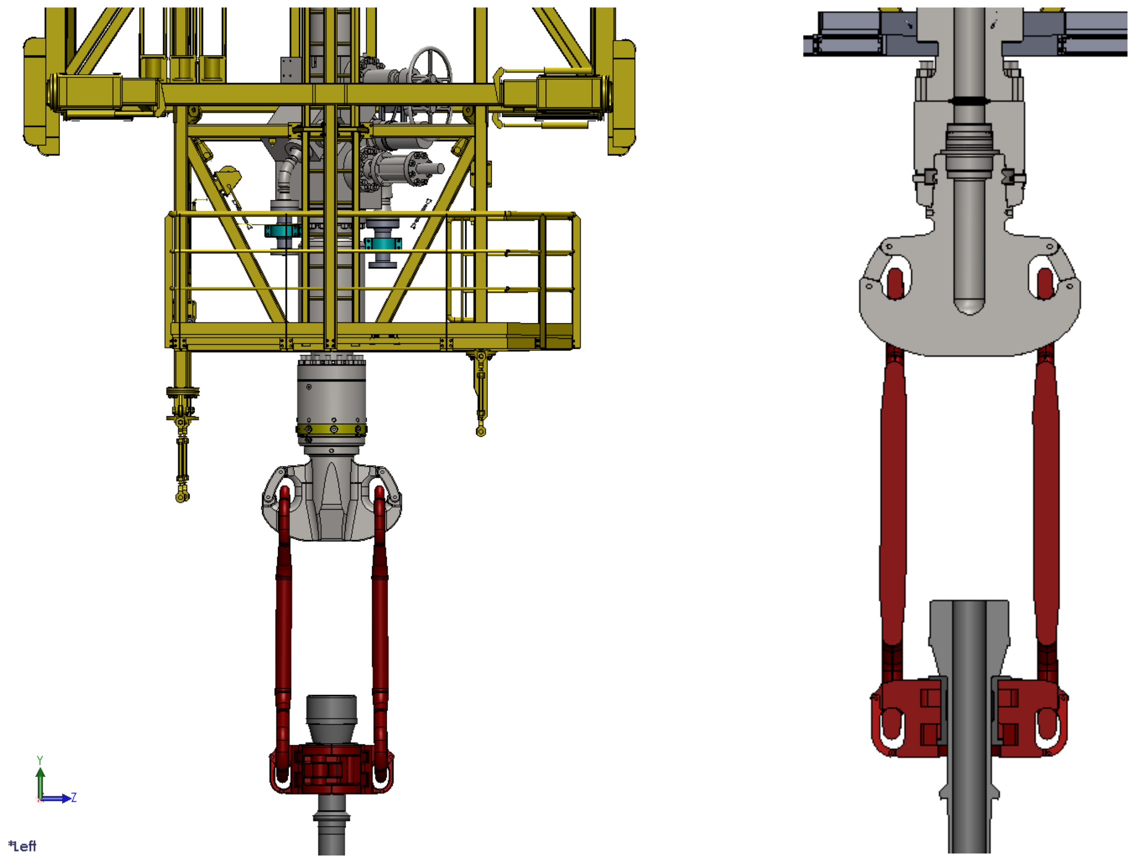


Figure 17: Total assembly connected to SFT

Pin from landing joint.

The upper part of the adapter is the same as the landing joint (LJ). The Landing Joint pin head is an existing design and *no alteration* is done to this part of the adapter. This is very important so that the adapter would fit into the SFT. Analysis of the pin head already exists, and this part of the adapter is therefore not included in further analysis in this thesis.

Figure 18: Landing Joint pin head

Additional Design Comments:

The adapter is tailor made for this operation, and following are some design features that are implemented (see figure 19).

1. Adapter pin head is copied from the landing joint
2. Short and compact neck that withstands stresses in accidental heel condition.
3. Vertical distance between link connection points is close to the elevator pad-eye distance so that the links is in a vertical position.
4. Pad-eye is divided into two parts connected with a lock piece.
5. Pad-eye inside diameter should give room for the link.
6. Additional height on the outer part of the lower eye to prevent the links to slide off.
7. The lower part of the eye need a minimum section area to ensure the desired capacity (height > width). The cross section is increasing to be able to withstand stresses due to bending forces.
8. Section area for upper eye is less than the lower eye due to less force traveling through this part. (see appendix 6)
9. Rounded upper corners of the lower eye to increase contact surface between link and adapter and reduce stress.
10. Small angle on the upper part of the pad-eye arm reducing stress.
11. The widest solution for the lock piece is when it has the outer ends around a pin/bolt. (Single ended lock piece has approximately half the width).
12. Use an installation guide connected around the pin flange (4 bolt holes are included) when installing to secured for falling equipment.

Fillets are added to the entire component to reduce stress concentrations. A lot of thought has been made to this design.

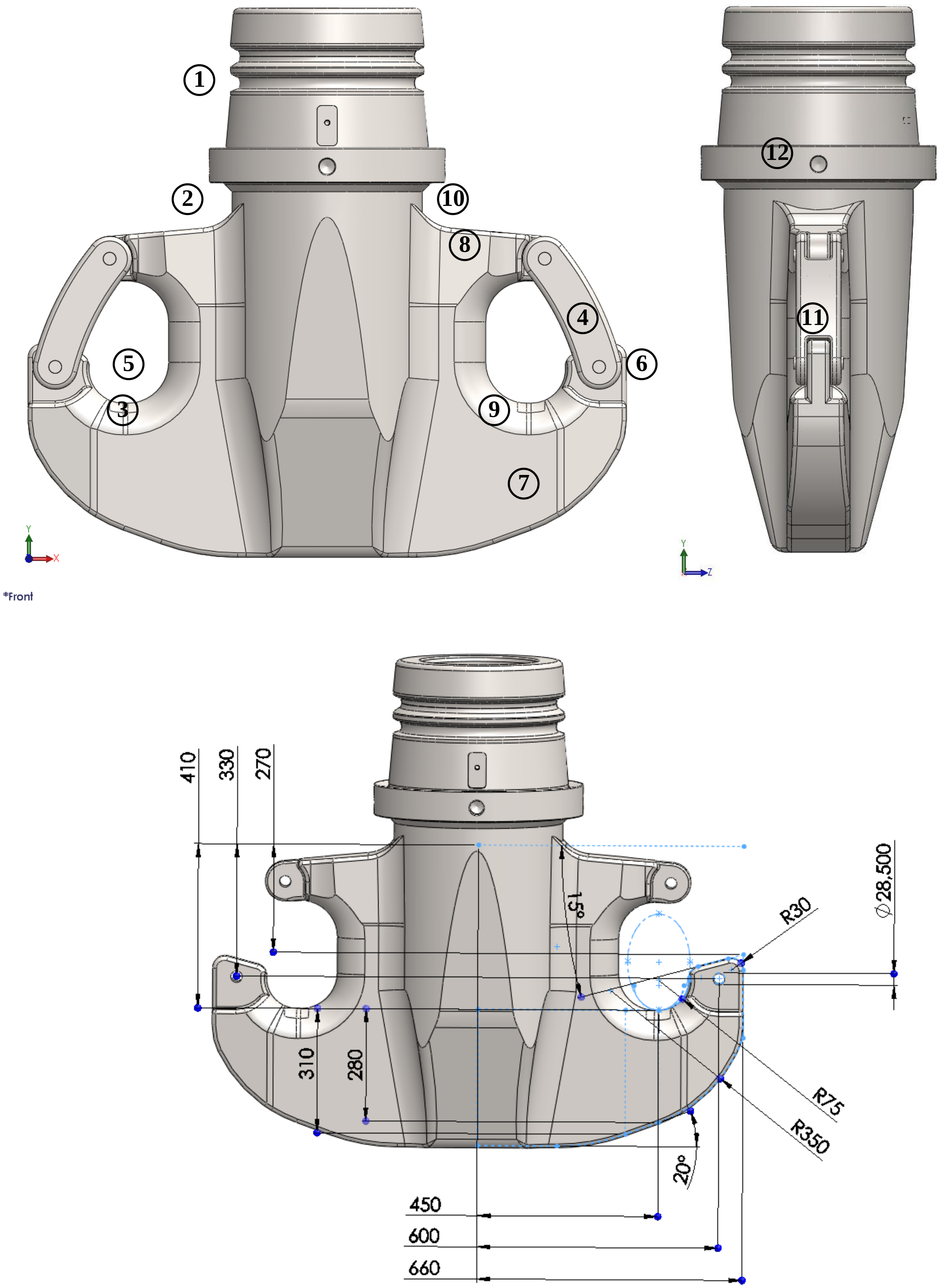


Figure 19: Adapter with some selected features

6.1.4 Contact surface radii

This subchapter uses the information in chapter 3 and 5 to draw a design basis. The rest of the design is set by strength and functionality. Features due to functionality have been described in chapter 6.1.1-6.1.3 and calculations proving the strength can be found in chapter 6.2.

API contact surface radii

It was stated, in chapter 5.2.2, that in addition to strength requirements, API also provided some recommendation on the surface radiuses for the contact regions. The table gives API radiuses together with the selected design for the new handling system.

Table 9: Contact surface radii from API regulations table 9B

			API mark		*Design*
Component	***Radii type (Section/ Surface)***	***mark***	***Max/ Min***	***Radii [millimetres]***	***Radii [millimetres]***
Adapter eye	*Cross section radii*	*C1*	*Max*	*101.6*	*70*
	Outer surface radii	*D1*	*Min*	*57.15*	*90*
Link upper eye	*Outer surface radii*	*C2*	*Min*	*120.65*	*127*
	Cross section radii	*D2*	*Max*	*47.63*	*47.63*
Link lower eye	*Cross section radii*	*G1*	*Max*	*57.15*	*57.15*
	Outer surface radii	*H1*	*Min*	*127.00*	*127*
Elevator eye	*Outer surface radii*	*G2*	*Min*	*60.32*	*70*
	Cross section radii	*H2*	*Max*	*127.00*	*120*

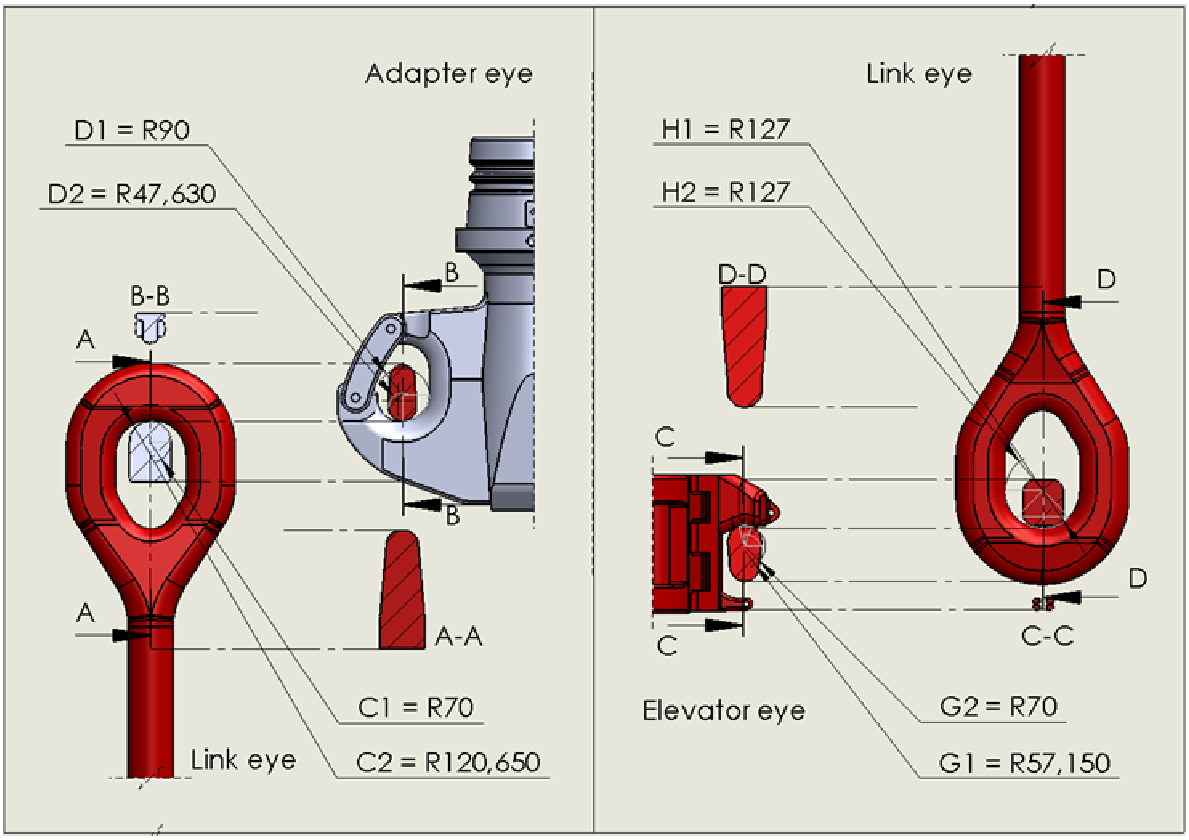


Figure 20: Design radii according to API specifications.

6.2 Control Calculations

The analysis is based on elastic theory. For every cross section that are considered the most unfavourable combination of force and direction are used.

Global force analysis – force path

The load is transferred from the riser joint through the padding to the main part of the elevator. Due to small connection surface between the elevator doors, the main load path goes through the main elevator body to the elevator pad-eyes. The force is taken mainly in the upper part of the elevator pad-eye. From the pad-eyes the load travels to the adapter eyes via the links. The load is distributed equally through both sides of the link pad-eyes. On the adapter, the load goes from the adapter eyes through the pin head and to the Surface Flow Tree. The adapter and elevator eye locks are not considered as load carrying components, and will therefore be excluded from analysis (Calculations proving this can be found in appendix 6).

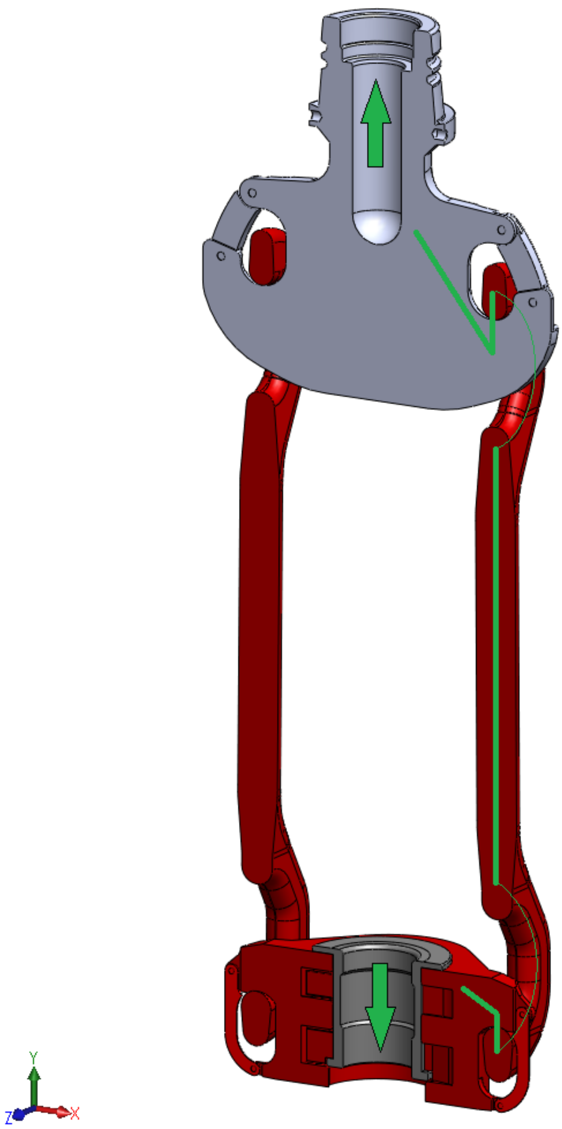


Figure 21: Main load path illustrated through one link

Geometric relations:

Rectangular: $I = \frac{bh^3}{12}$, $y = h/2$, $A = bh$

Circular: $I = \frac{\pi}{4}(r_{out}^4 - r_{in}^4)$, $y = r_{out}$, $A = \pi\,(r_{out}^2 - r_{in}^2)$

Stresses:
Stresses used in elastic analysis due to forces in axial direction and momentums.

Axial Force: $\sigma_n = \frac{N}{A}$

Momentum: $\sigma_m = \frac{M}{I} y$

Shear:
Shear used in elastic analysis.

Shear Force: $\tau = \frac{V}{I\,t} S'$

Shear in rectangular cross section: $\tau_{max} = \frac{3}{2} \cdot \frac{V}{A}$

Assume that the shear stresses are distributed equally among the entire cross section the middle shear stress is used in the calculation:

Middle value: $\tau_{middle} = \frac{V}{A}$

Von Mises stress:
According to DNV and API, Von Mises shall be used when calculating the total stress level.

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2}$$

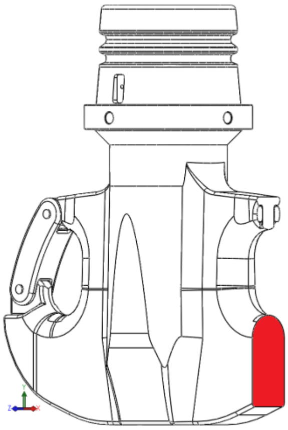
Acceptance levels:
The acceptance levels are set according to API and DNV. The design safety factors vary between the cases. When material and load factors are specified separately the load factor is multiplied to the load, and the design safety factor is equal to the material safety factor.

$$Maximum\ Allowable\ Stress = \frac{Specified\ Minimum\ Yield\ Strenght}{Design\ Safety\ Factor}$$

$$\sigma_{allow} = \frac{\sigma_y}{SF_D}$$

6.2.1 Adapter

Case 1: API-load - Adapter pad-eye – Critical shear; cross section at link connection
Shear stresses are assumed to be dominant.



Geometry:

$h = 280\ mm$

$b = 140\ mm$

$$V = F_D/2$$

$$\tau_v = \frac{V}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{3} \cdot \tau_v = \sqrt{3} \cdot \left(\frac{F_D}{2hb}\right)$$

API-load:

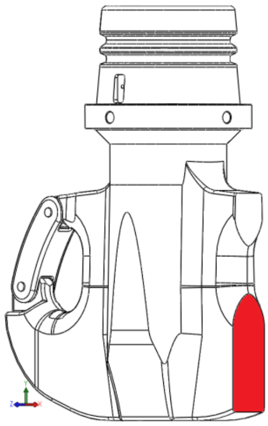
$F_D = 11\ 310\ kN$

$\sigma_{tot} = 249.9\ MPa$

$\sigma_{allow} = 551\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Longrightarrow OK$

Case 2: API-load - Adapter pad-eye – Combination; cross section 80 mm from link
Stress due to axial force is assumed to be small in comparison to shear and stress due to moments.



Geometry:

$h = 310\ mm$

$b = 140\ mm$

$x = 80\ mm$

$$M = F_D/2 \cdot x$$

$$V = F_D/2$$

$$\sigma_m = \frac{M}{I} y = \frac{\left(\frac{F_D x}{2}\right)}{\left(\frac{bh^3}{12}\right)} \cdot \left(\frac{h}{2}\right) = \frac{(F_D x)}{bh^2} \cdot 3$$

$$\tau_v = \frac{V}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{\left(\frac{(F_D x)}{bh^2} \cdot 3\right)^2 + 3\left(\frac{F_D}{2hb}\right)^2}$$

API-load:

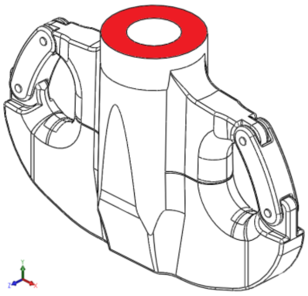
$F_D = 11\ 310\ kN$

$\sigma_{tot} = 302.7\ MPa$

$\sigma_{allow} = 551\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Longrightarrow OK$

Case 3: API-load - Adapter neck – Critical stress; cross section at SFT connection
Stresses due to axial forces are assumed to be dominant, and $N = F_D$.



Geometry:

$r_{out} = 210\ mm$

$r_{in} = 93\ mm$

$$\sigma_n = \frac{N}{A} = \frac{F_D}{\pi\ (r_{out}^2 - r_{in}^2)}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{\left(\frac{F_D}{\pi\ (r_{out}^2 - r_{in}^2)}\right)^2}$$

API-load:

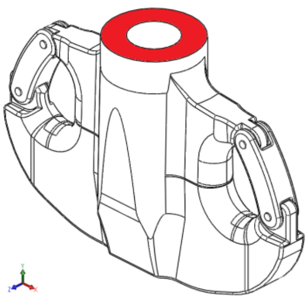
$F_D = 11\ 310\ kN$

$\sigma_{tot} = 101.6\ MPa$

$\sigma_{allow} = 551\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Rightarrow OK$

Case 4: Accidental - Adapter neck – Combination; cross section at SFT connection.
Shear stresses are assumed to be negligible.



Geometry:

$r_{out} = 210\ mm$

$r_{in} = 93\ mm$

$x = 450\ mm$

$$M = F_D \cdot x$$

$$N = F_D$$

$$\sigma_m = \frac{M}{I} y = \frac{(F_D x)}{\left(\frac{\pi}{4}(r_{out}^4 - r_{in}^4)\right)} \cdot (r_{out})$$

$$\sigma_n = \frac{N}{A} = \frac{F_D}{\pi\ (r_{out}^2 - r_{in}^2)}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{(\sigma_n + \sigma_m)^2} = \sigma_n + \sigma_m$$

$$\sigma_{tot} = \left(\frac{F_D x \cdot r_{out}}{\frac{\pi}{4}(r_{out}^4 - r_{in}^4)}\right) + \left(\frac{F_D}{\pi\ (r_{out}^2 - r_{in}^2)}\right)$$

Accidental heel:

$F_D = 5\ 027 \cdot 0.96 = 4\ 826\ kN$

$\sigma_{tot} = 353.8\ MPa$

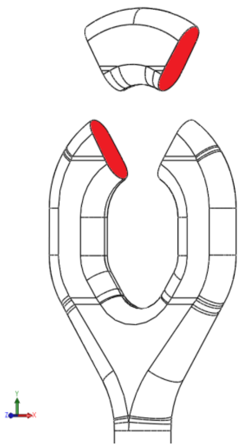
$\sigma_{allow} = \frac{551}{1.15} = 479.1\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Rightarrow OK$

6.2.2 Links

Case 5: Accidental heel - Link pad-eye –Tear out at adapter connection

The force is tearing out a part of the upper pad-eye. The part removed are simplified as a beam with a length equal the adapter width. The cross section is conservatively taken as 180 mm high and 90 mm wide. The momentum is assumed to be equal a beam fixed in both ends with equally distributed load. Normal forces are assumed to be zero.



Geometry:

$h = 180\ mm$

$b = 90\ mm$

$l = 140\ mm$

$$q = \frac{F_D}{l}$$

$$M = \frac{ql^2}{12} = \frac{F_D l}{12}$$

$$V = \frac{ql}{2} = \frac{F_D}{2}$$

$$\sigma_m = \frac{\mathrm{M}}{\mathrm{I}}\mathrm{y} = \frac{(F_D l/12)}{\left(\frac{bh^3}{12}\right)} \cdot \left(\frac{h}{2}\right) = \frac{F_D l}{2bh^2}$$

$$\tau_v = \frac{V}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{\left(\frac{F_D l}{2bh^2}\right)^2 + 3\left(\frac{F_D}{2hb}\right)^2}$$

Accidental heel:

$F_D = 5\ 027 \cdot 0.96 = 4\ 826\ kN$

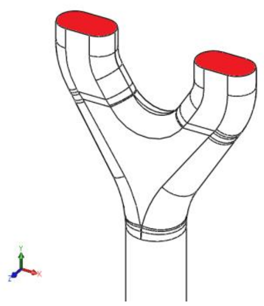
$\sigma_{tot} = 282.8\ MPa$

$$\sigma_{allow} = \frac{551}{1.15} = 479.1\ MPa$$

$\sigma_{tot} \leq \sigma_{allow} \quad \Longrightarrow OK$

Case 6: Accidental heel - Link pad-eye – Critical stress; cross section 80 mm from link connection

Forces are assumed to be taken mainly in axial/normal direction. Shear and moments are zero.



Geometry:

$h = 165\ mm$

$b = 96\ mm$

$l = 420\ mm$

$$N = F_D/2$$

$$\sigma_n = \frac{N}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sigma_n = \frac{F_D}{2hb}$$

Accidental heel:

$$F_D = 5\,027 \cdot 0.96 = 4\,826\ kN$$

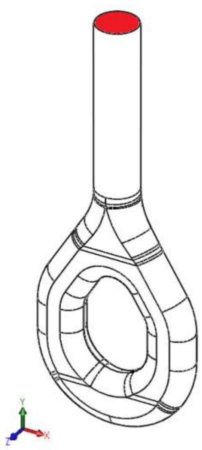
$$\sigma_{tot} = 158.7\ MPa$$

$$\sigma_{allow} = \frac{551}{1.15} = 479.1\ MPa$$

$$\sigma_{tot} \leq \sigma_{allow} \implies OK$$

Case 7: Accidental heel - Link pad-eye – Critical axial shear; middle cross section

Forces are assumed to be taken in axial direction. Shear and moments are assumed negligible.



Geometry:

$r = 76.2\ mm$

$$N = F_D$$

$$\sigma_n = \frac{N}{A} = \frac{F_D}{\pi r^2}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sigma_n = \frac{F_D}{\pi r^2}$$

Accidental heel:

$$F_D = 5\,027 \cdot 0.96 = 4\,826\ kN$$

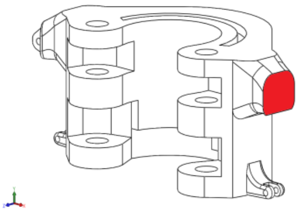
$$\sigma_{tot} = 264.5\ MPa$$

$$\sigma_{allow} = \frac{551}{1.15} = 479.1\ MPa$$

$$\sigma_{tot} \leq \sigma_{allow} \implies OK$$

6.2.3 Elevator

Case 8: API-load - Elevator pad-eye – Critical shear; cross section at link connection
Shear stresses are assumed to be dominant.



<u>Geometry:</u>

$h = 160\ mm$

$b = 138\ mm$

$$V = F_D/2$$

$$\tau_v = \frac{V}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{3} \cdot \tau_v = \sqrt{3} \cdot \left(\frac{F_D}{2hb}\right)$$

<u>API-load:</u>

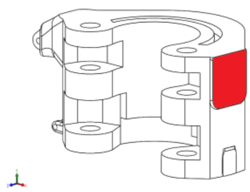
$F_D = 11\ 310\ kN$

$\sigma_{tot} = 443.6\ MPa$

$\sigma_{allow} = 551\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Longrightarrow OK$

Case 9: API-load - Elevator pad-eye – Combination; cross section 80 mm from link
Shear stress and stress due to moments are assumed to be dominating.



<u>Geometry:</u>

$h = 210\ mm$

$b = 180\ mm$

$x = 80\ mm$

$$M = F_D/2 \cdot x$$

$$V = F_D/2$$

$$\sigma_m = \frac{M}{I} y = \frac{\left(\frac{F_D x}{2}\right)}{\left(\frac{bh^3}{12}\right)} \cdot \left(\frac{h}{2}\right) = \frac{(F_D x)}{bh^2} \cdot 3$$

$$\tau_v = \frac{V}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{\left(\frac{(F_D x)}{bh^2} \cdot 3\right)^2 + 3\left(\frac{F_D}{2hb}\right)^2}$$

<u>API-load:</u>

$F_D = 11\ 310\ kN$

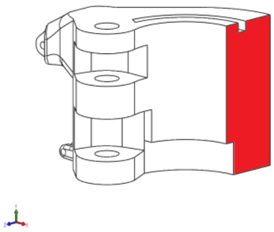
$\sigma_{tot} = 429.0\ MPa$

$\sigma_{allow} = 551\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Longrightarrow OK$

Case 10: API-load - Elevator – Critical combination; cross section 460 mm from link

Shear stress is assumed to be dominating. Moments are assumed to be zero, and axial stress is not taken into account in this calculation.



Geometry:

$h = 210\ mm$

$b = 180\ mm$

$x = 460\ mm$

$z = 352\ mm$

$V = F_D/2$

$$\tau_v = \frac{V}{A} = \frac{F_D}{2hb}$$

$$\sigma_{tot} = \sqrt{(\sigma_n + \sigma_m)^2 + 3\tau_V^2} = \sqrt{3}\,\tau_V = \sqrt{3}\frac{F_D}{2hb}$$

API-load:

$F_D = 11\ 310\ kN$

$\sigma_{tot} = 118.3\ MPa$

$\sigma_{allow} = 551\ MPa$

$\sigma_{tot} \leq \sigma_{allow} \quad \Longrightarrow OK$

6.3 Fatigue analysis

The number of cycles for the lifting aperture can easily be estimates. The life span is according to DNV, and should be at least 20 years. The estimated number of well connections per year is 18-20. The number of well connections is conservatively set to 20 per year, indicating a total of 400 well connections during a normal life span.

The number of maximum stress cycles per trip is equal to the number of riser elements retrieved and deployed. The average riser stack adjustment is conservatively set to 150 meters, due to the suggested location of the Lubrication Valve. The lubricator valve is placed 150 meters down, so that it is not necessary to disconnect this component when changing operation site. With a standard riser element length of 15.6 meters, this indicates approximately 10 elements, i.e. 10 max stress cycles retrieving riser elements, and 10 cycles when deploying the riser stack.

$$N = 20\frac{year}{life} \cdot 20\ \frac{wells}{year} \cdot 20\frac{cycles}{trip}$$

$$N = 8000\ \frac{cycles}{life}$$

If the life span is increased to 30 years the number of cycles are 12 000 if the average depth difference between wells are 150 meters. If the average depth is increased to 500 meters, the number of cycles during 20 years is approximately 26 000 cycles. The fatigue analysis is in all cases a low-cycle analysis.

The highest stress amplitude in the handling system during normal operation is case 8 at elevator eye, measuring 256.3 MPa. Multiplied with the fatigue safety factor of 1.33, this becomes 341 MPa. DNV provide following S-N curve proving that the equipment at hand are not in risk of fatigue until 110 000 cycles.

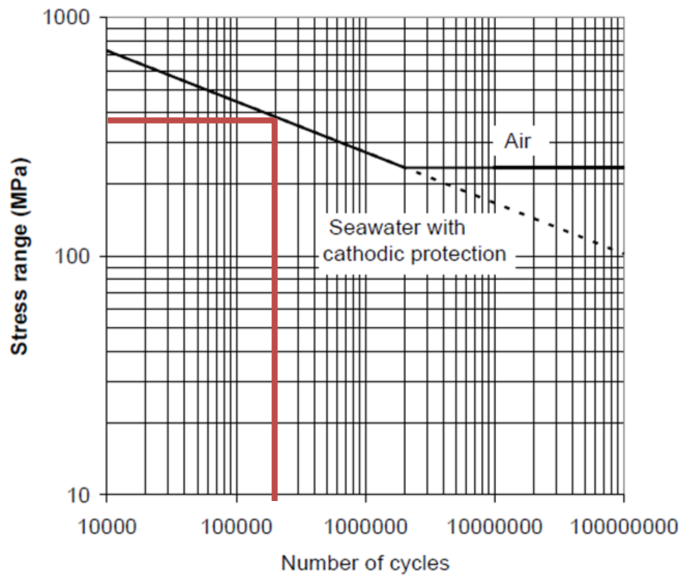


Figure 22: S-N curve for high strength steel (DNV RP-C203 figure 2-10)

Chapter 7: Strength Analysis

In this chapter the results of the strength analysis computed in SolidWorks are presented. First is a short summary of how the solver works and aspects which are important when setting up and evaluating the results. Then the analysis results are presented. Three load cases are used as basis for evaluating the strength of the different components; normal condition, accidental heel condition, and the API-load condition.

As stated in the load case the loads used in the analysis are summarized in following table. The collision case is less critical than API-load and is therefore excluded from the simulation. Normal case is included as a reference case. The design force is $(450 - 30)mT \cdot 1.22 \cdot 9.81\ m/s^2 = 5027\ kN$.

Table 10: Simulation Load Cases

Load Case	***Design Force***	***Load factor***	***Case Force***
Operation Limit	*5027 kN*	*1.3*	*7 540 kN*
Accidental heel	*5027 kN*	*0.96*	*4 825 kN*
API-load	*5027 kN*	*2.25**	*11 310 kN*

* includes the material factor

The DNV material factor in normal and accidental case is 1.15. When testing with API-load the material factor is included in the design safety factor calculated from the rated load. The main material used is 551MPa (80 ksi). This gives an acceptance level of 479.1 MPa in normal and accidental condition, while API-load is tested up to yield 551 MPa.

$$\sigma_{allow,normal/accidental} = \frac{\sigma_y}{SF_D} = \frac{551}{1.15} = 479.1\ MPa$$

$$\sigma_{allow,API} = \sigma_y = 551.0\ MPa$$

7.1 SolidWorks analysis tool

7.1.1 Study and Plots

In this part a simple *static study* of the equipment are conducted. The focus is set on deformation and stresses in the components. SolidWorks provide a *displacement plot* to help the designer evaluating the model deformation. To evaluate the stresses over the components SolidWorks creates a *Von Mises plot*. This plot indicates the Von Mises stress distribution over the complete assembly. The component strength is related to the material and the safety factor is set to the limit strength in the material data. Some materials are likely to yield and other tends to fracture when stress levels reaches a certain point. The materials used in this study are ductile, and yielding are therefore the most likely failure. The static studies assume a linear stress-strain relation, and the strain can be displayed in a strain plot.

7.1.2 Bodies and Material

SolidWorks supports single body parts, multi body parts and assemblies. Each body in the simulation can have different physical properties and they may interact with each other in different ways. To simplify the analysis following steps may be taken:

- Exclude bodies
- Define connectors
- Treat bodies as *rigid and movable* or as *rigid and fixed in space*
- Use symmetry as an advantage

Bodies that are often excluded are bolts and pins. To make sure that the body reacts correctly when bodies are excluded, connectors are defined instead. If the model has multiple bodies the type of contact between them has to be stated. The contact types may be; no penetration, bonded (no clearance), or allow penetration. The rigid bodies do not have stress or strain distribution. The body evaluated deform relative to fixed bodies or surfaces. Using symmetry to simplify the model is a great aid to limit the simulation time. If symmetry applies the number of elements can be reduced and the simulation will be faster. If one-plane symmetry applies the number of elements is reduced by a factor of ~2, in two-plane symmetry the factor is ~4. Two-plane symmetry may be applied on the links and the adapter. Symmetry does not apply on the elevator. Figure 23 illustrates how two-plane symmetry may be set on the adapter.

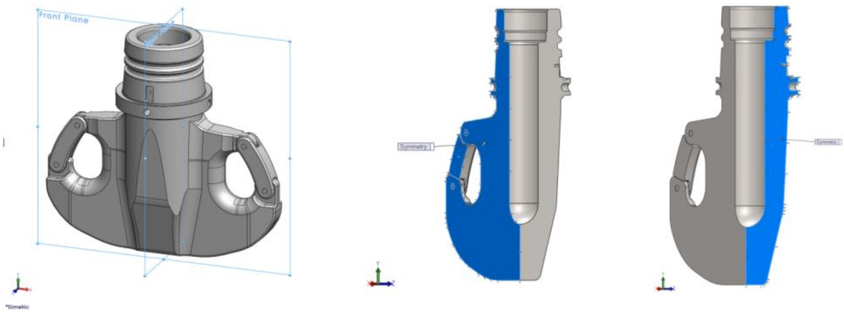
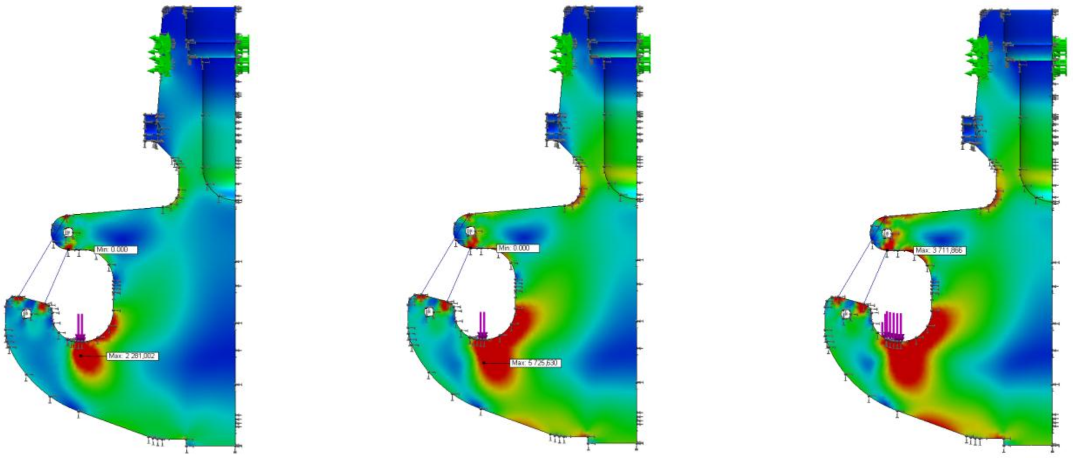


Figure 23: Two-plane symmetry on the adapter.

7.1.3 Interaction, Fixture and Force

SolidWorks uses *interactions* to describe how bodies relate to each other and the environment. Interaction can be loads, fixtures, connectors, and contact. It is very important to add interactions in a correct manner. The results are heavily dependent on how the loads and restrictions are stated. Fixtures describe how the model is supported. Connectors can be pins, bolts, springs, or links. Figure 24 show the adapter whit two-plane symmetry, fixed at the pin, with bolts excluded, and links added instead of the lock pieces. The figure illustrates how different ways of adding the same load may affect the result. The magnitude is the same, but the area of contact and direction varies.



Small force area *Force normal to surface*	*Small force area* *Force normal to top plane*	*Large force area* *Force normal to top plane*

Figure 24: Force area and direction

A small force area with the load acting normal to the surface gives much lower stress levels through the body. The most conservative and correct result are found if the force are assumed to affect a relatively large force area with force only acting in the downward direction, i.e. normal to the top plane. The load area can be restrained by drawing a split line on the surface.

The static study assumed that load direction and area of contact does not change during simulation. If this is not true, then a contact interaction has to be applied instead. In the cases evaluated in this report direction and contact area are constant during maximum load condition. Loads can therefore be applied instead of interactions in all cases.

7.1.4 Mesh properties

The mesh is very important for the accuracy of the simulation. Usually the SolidWorks default mesh would give reasonable accuracy evaluating the deformation. If stress and strain are to be evaluated, an improved and more detailed mesh is needed. Figure 25 illustrates how the mesh element size affects the accuracy when evaluating stress. The standard mesh uses the Voronoi-Delaunay meshing scheme. This mesh is used during pre-limanary studies. During the final study a curvature meshing scheme is used. The curvature mesh automatically creates more elements in areas with steeper curvature. It is also possible to add mesh control. This function would also create more elements at specified locations.

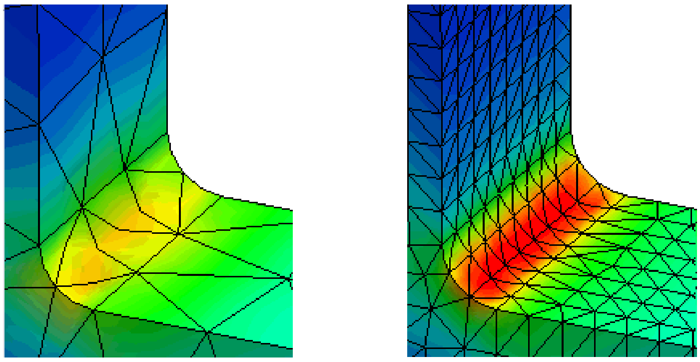


Figure 25: Illustration from SolidWorks advisor; mesh properties

7.1.5 Solver

The SolidWorks solver uses iteration process to calculate the results. The analysis is static and linear. For the elastic material where deformations are large a non-linear study should be used. The solver start solving the environmental constrains as fixtures and load restrains. Then all interactions and surface constrains are set. When the solver is finished setting the boundary conditions, the force can be added. The load is added fraction wise, solving the complete model before increasing the load. The solver uses numerical iterations with convergence parameters that are accepted if they reach a pre-set stopping threshold.

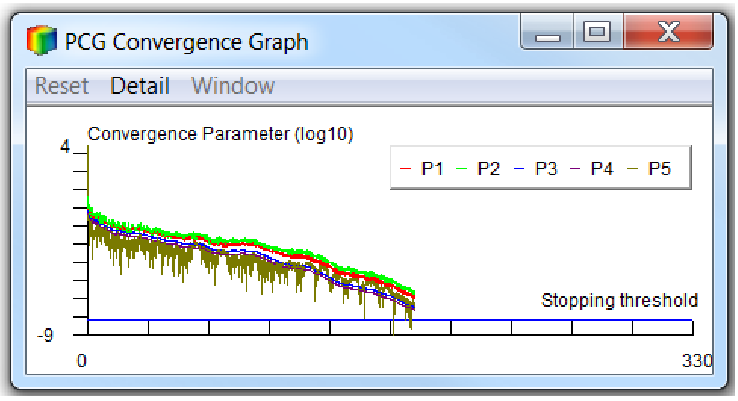


Figure 26: Illustration from SolidWorks solver; convergence graph

7.2 Adapter

The adapter is the component connecting the yoke and the Surface Flow Tree. It is rigidly mounted in the same manner as the landing joint. Teeth are pushed into the guide groves at the adapter pin head, locking the adapter completely. The loads are transferred to the adapter via the link connections.

7.2.1 Simulation settings Adapter

All bodies are included in the analysis of the adapter. To simplify the simulation, pins are assumed to be rigid but movable. There are three cases that are to be analysed; normal, accidental heel, and API-load. The adapter has valid two-plane symmetry in normal and API-load condition. In heel condition, the load is not distributed equally on both pad-eyes, so only one-plane symmetry is valid in this case. This result in two different simulation setups illustrated in *figure 27: Adapter settings; mesh, fixture, and load*. Due to symmetry the total load applied on the quart adapter are divided by 4, and the total force applied on the half model for accidental heel are reduced by a factor of 2. The fixture is applied to the guide groves at the adapter pin head, and is assumed to be fixed (also fixed in space, not roller fixture). In figure 27 the fixture are illustrated by green arrows, force by pink arrows, and symmetry by blue arrows.

Normal and API-load *Accidental heel*

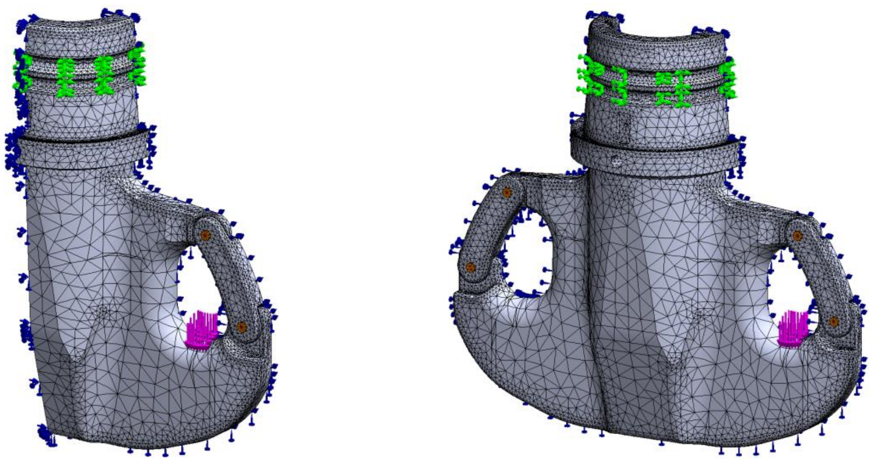


Figure 27: Adapter settings; mesh, fixture, and load

The material used specified in chapter 5 is 551 steel and component contacts is set to no penetration. The components in the study are treated as solid bodies and meshed with solid elements. A curvature based mesh is used with the following mesh parameters:

- *Max element width:* *50 mm*
- *Min element width:* *10 mm*
- *Min number of element in a circle:* *8*
- *Size growth ratio:* *1.6*

The mesh is a mixed mesh with high quality and has 87 150 elements and 136 692 nodes for the 1/4 model.

7.2.2 Simulation Results Adapter

SolidWorks simulation is run and the following results were posted.

7.2.2.1 Adapter Displacement

DNV and API state the displacement and deformation is a concern to evaluate. SolidWorks has a displacement plot that is used as an aid to evaluate the displacement while the component is loaded. The displacement is plotted as combined displacement in; x-, y-, and z-directions, hence resultant displacement (u_{res}). The illustration shows the model in its deformed state.

In all cases the maximum displacement is found at the lower end of the adapter pad-eye. Displacement-pattern for normal and API-load would seem identical if displayed with different scale. Figure 28 illustrates displacement in all three cases with the same scale. In heel, the pattern is different and the deformation is much larger than in the two other cases.

In normal and API condition most of the movements are in y-direction along the centre axis of the model. In normal condition the greatest displacement (u_{res}) measures 0.744 mm. With API-load this increases to 1.277 mm, where 1.268 mm is in y-direction. This means that the component basically flexes downwards. Relative deflection between point of fixture and maximum displacement is 0.15% in y-direction[16].

The overall maximum occurs during accidental heel, and measures 3.6 mm at the loaded pad-eye. This indicates that the whole model is rotated around the centre axis, as the right side is pushed downwards while the left is shifted upwards. The right eye is forced 3.15 mm down, while the left eye is lifted 2.05 mm. Relative deflection between point of fixture and max displacement is 0.37% in y-direction, while the maximum beam deflection is 0.57 %[17].

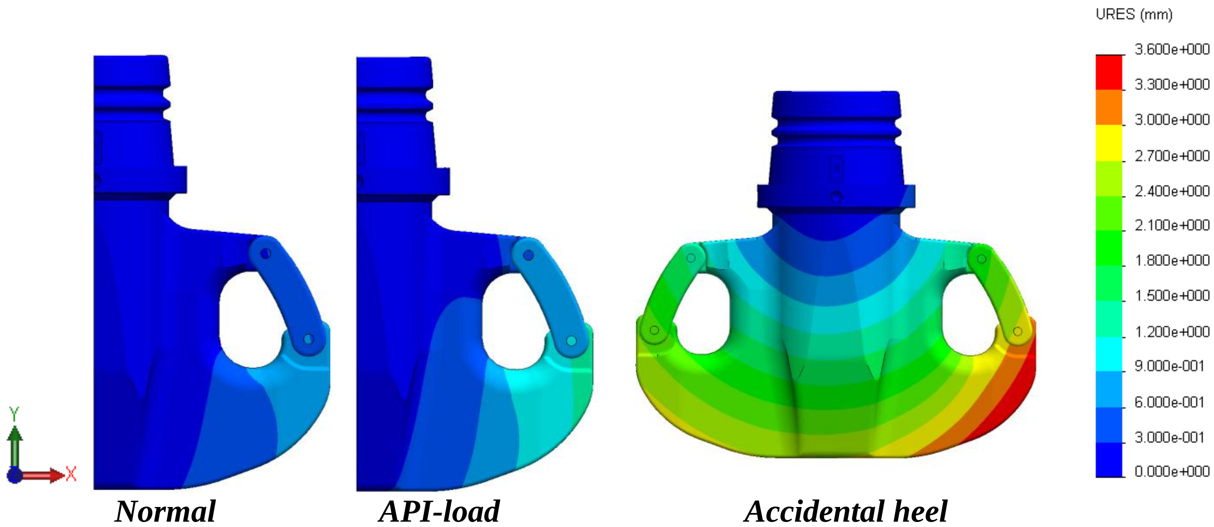


Figure 28: Displacement levels in Adapter at normal, API-load, and accidental condition

[16] Distance between fixture and maximum displacement is measured to 845 mm in y-direction.
[17] Lower pad-eye beam measures 300 mm with u_y =1.45 mm at base and u_y= 3.15 at tip.

7.2.2.2 Von Mises Stresses in Adapter

The adapter is checked for extensive yielding. During API-load the acceptance level is yield strength; 551 MPa. This means that regions with red colour are at high risk of yielding. Under normal operation and accidental heel, the acceptance is decreased to 479.1 MPa. This means that in these cases, orange and parts of yellow regions are considered in risk of yielding.

Stress levels in the adapter subjected to loads in normal, API-load and accidental heel are illustrated in Figure 29. It confirms the assumption that the upper pad-eye does not carry any load, and that the critical case for the pad-eye is when subjected to the API-load load. It also shows that the adapter has several hot-spots. One is where the adapter is fixed to the SFT, one is at the adapter neck, and two are located at the top of the lower pad-eye. A fifth stress concentration point can be seen in the contact surface between the main body and pad-eyes.

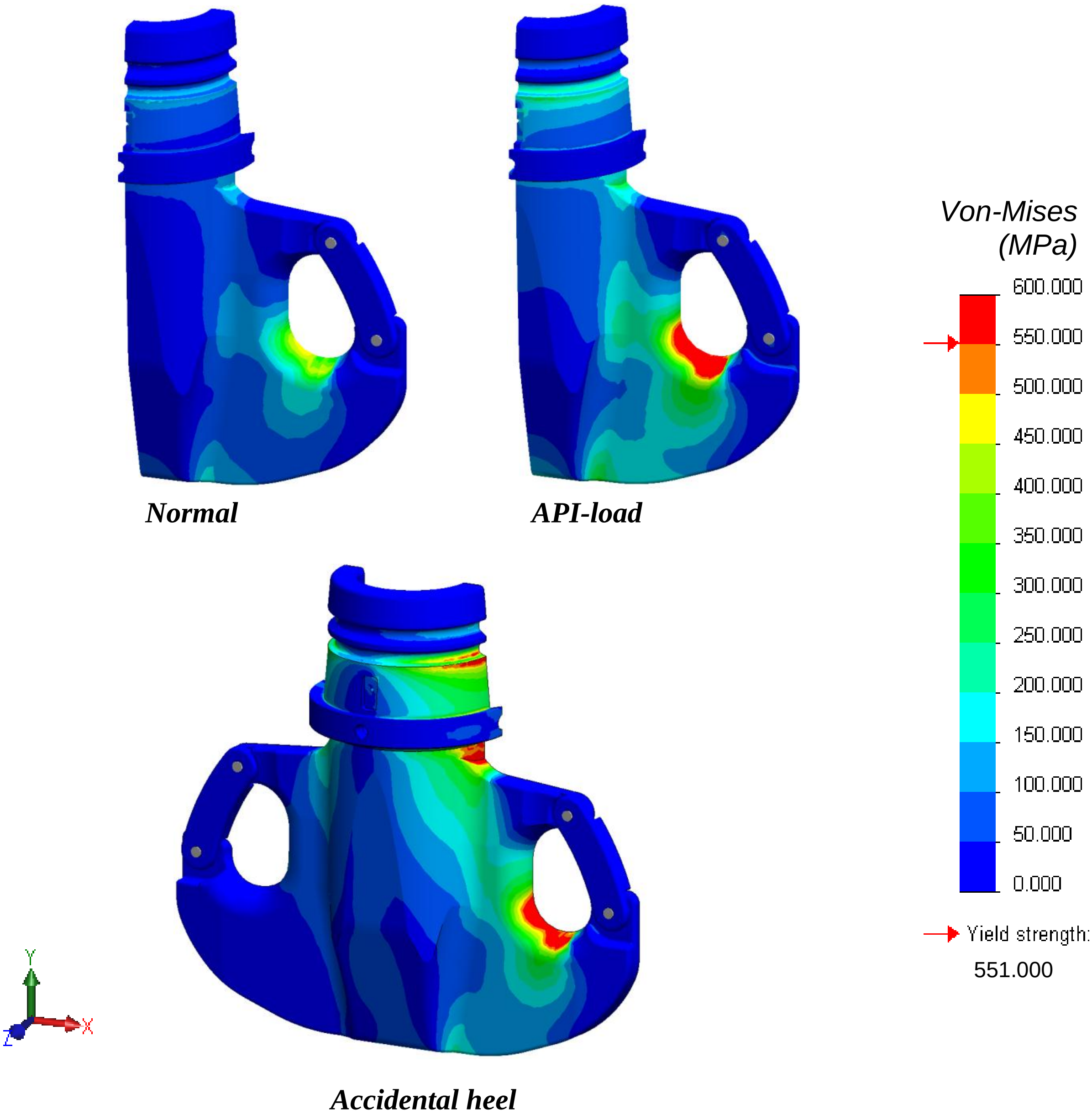


Figure 29: Stress levels in Adapter at normal, API-load, and accidental condition

Critical stress concentrations are illustrated using ISO-clipping tool in SolidWorks (see Figure 30, clipped at 479.1 MPa black arrow). This picture illustrates all regions that are in risk of yielding under accidental. Hot spot 1 and 4 are direct contact zones between the SFT-adapter and link-adapter respectively. 1 and 4 are accepted since API 8C accepts local yielding in contact regions and they only appear in accidental loading condition. After some lifts the adapter eye and link will deform so that the contact region increases which might alter the stress pattern.

Region 2 and 3 are stress concentrations due to sudden change in geometry. Suggested methods for improving the component design are to increase the fillet radiuses or change the material. The component is not at risk of rupture, but deformations might occur at these points if subjected to heel. The stress levels observed are acceptable since these deformations occur during accidental state, which is expected to occur seldom, if ever. This would not be acceptable during normal operation due to risk of fatigue.

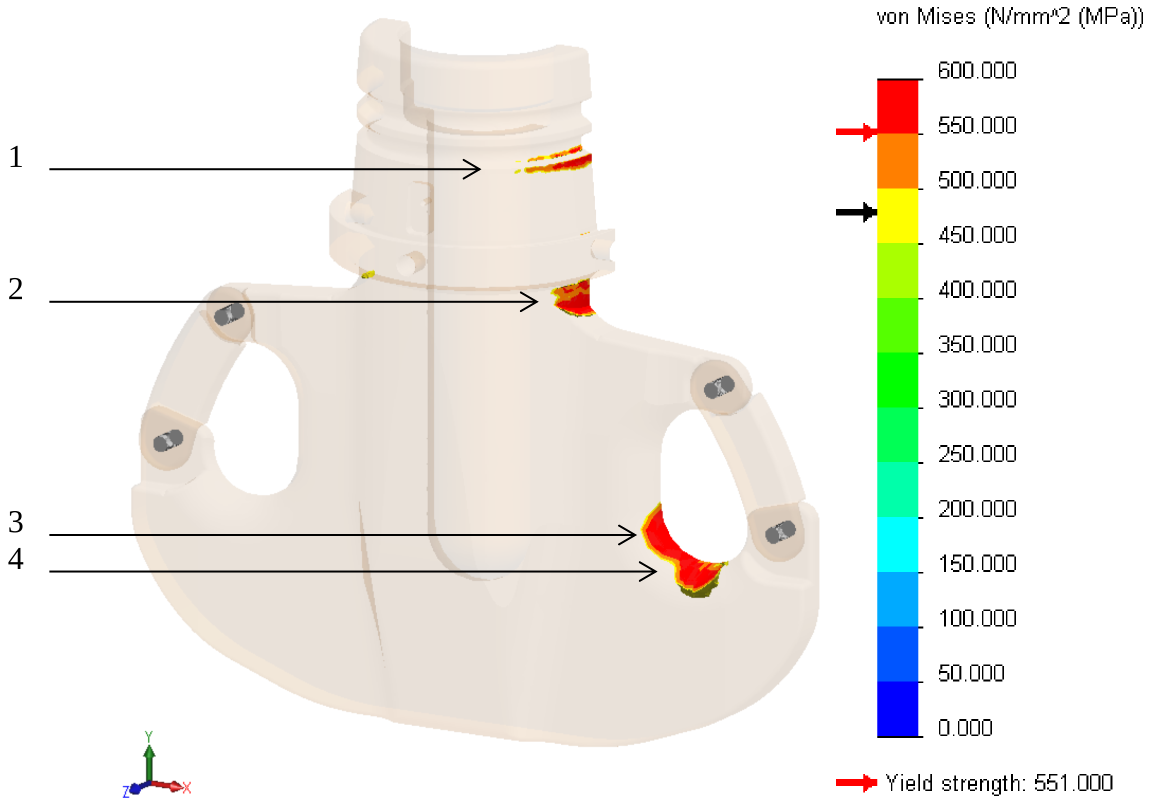


Figure 30: ISO-clipping adapter accidental heel condition

In general the stress level during normal operation is well below yield. The most critical region is located close to the links connection points. The maximum stress level in these areas is measured to 494.5 MPa, which is less the yielding, but higher than the acceptance level of 479.1 MPa.

The pad-eye experiences the highest stress levels during API-load condition. The average stress level over the cross sections in the pad-eye is illustrated in Figure 31. Calculations state that the stress level in the cross sections in the pad-eye should be close to 250 MPa. This seems to be close to the average value simulated by the SolidWorks solver.

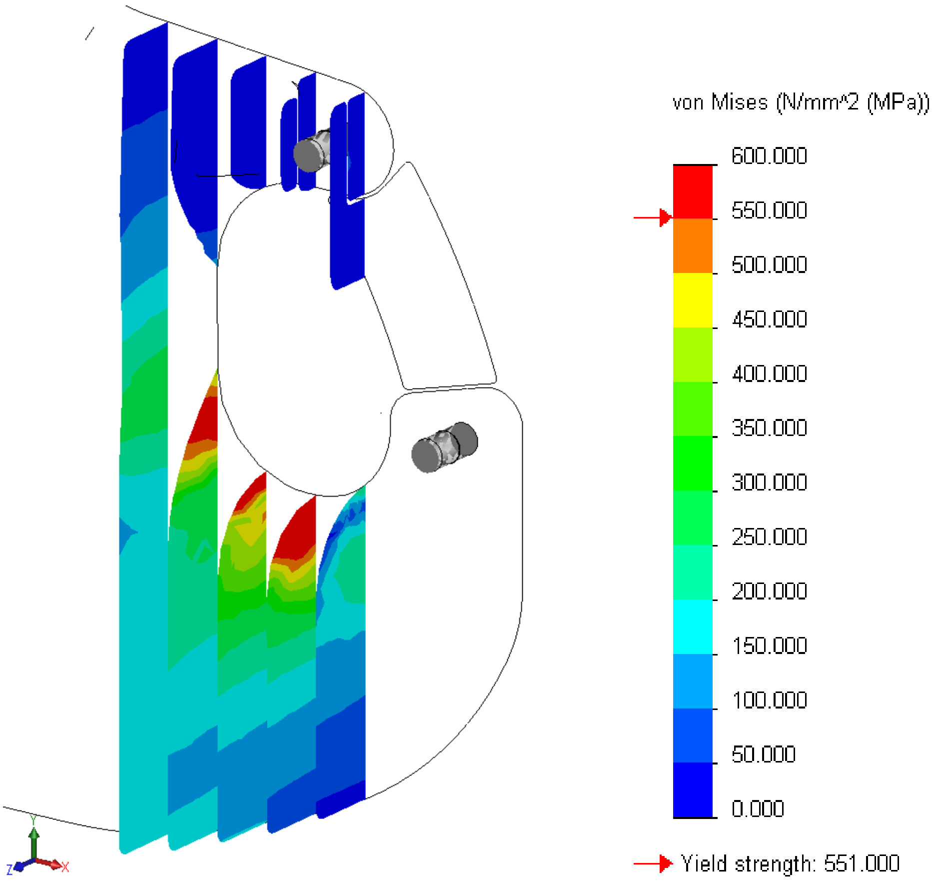


Figure 31: Adapter eye during API-load condition

The adapter has yielding at three places during accidental heel. Figure 32 and 33 shows section clippings along the load path and on the adapter pin head. The average stress level in the pin cross section is approximately 200 MPa which is less than the calculated level of 354 MPa. The adapter has yielding where the pad-eye meets the adapter neck. This is due to the sudden change in the geometry at this point. Increased fillet radii might solve the problem.

This is accepted since the area is small and would only be a problem if repeated many times. A suggestion is to control the adapter if it is subjected to accidental heel loads.

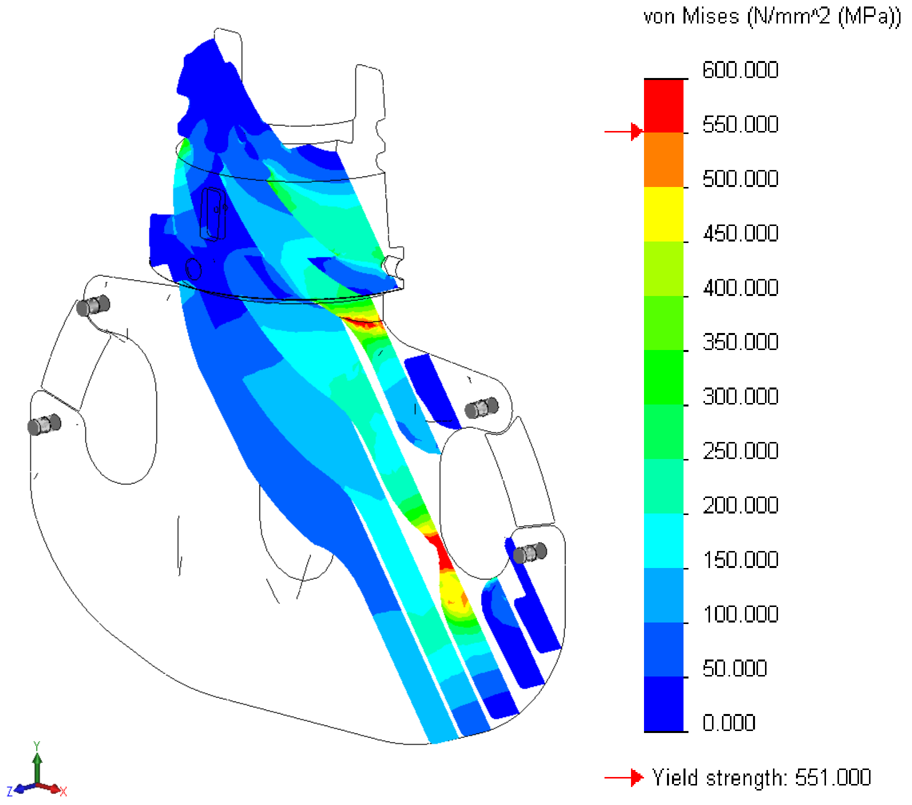


Figure 32: Adapter stresses at load path under accidental heel condition

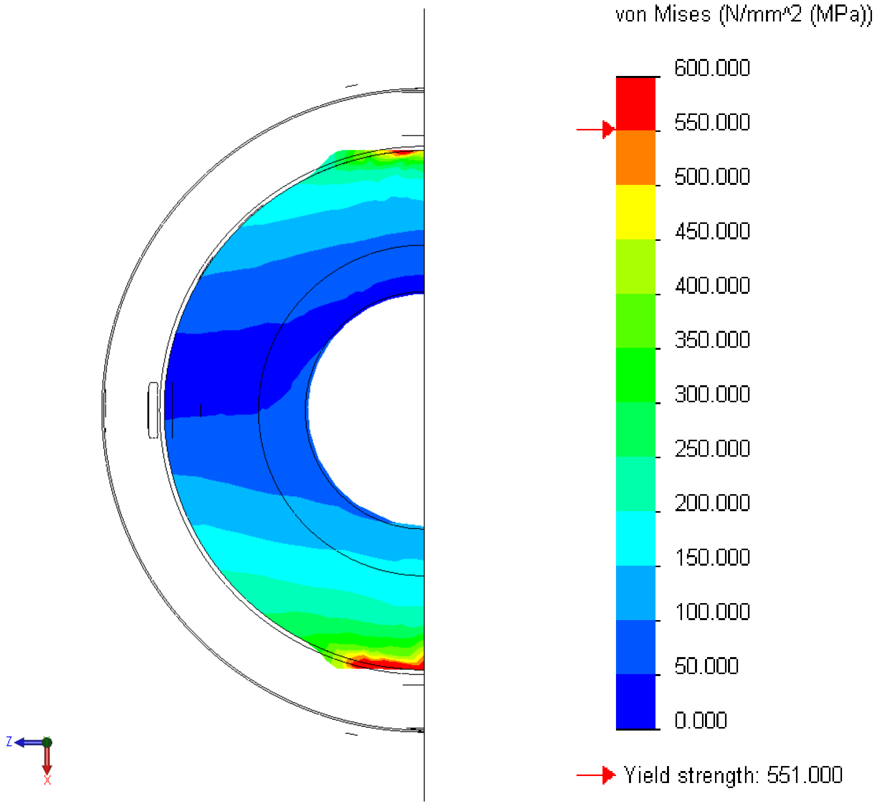


Figure 33: Stress in adapter pin head at accidental heel

7.3 Links

The purpose of the links is to connect the elevator to the adapter. They are enclosed in the pad-eyes and are simply supported in both ends. In normal and API-load condition two links are used to transfer the load, but in accidental heel it is assumed that the complete load is transmitted through one link. The load through a link during the tree cases are: *normal=3770kN*, *API-load=5655kN*, and *accidental heel=4826kN*. The stress levels are greatest in the API-load case, but the stress acceptance level is smallest in accidental heel condition, which makes it difficult to state the dimensioning case.

7.3.1 Simulation settings Links

The links are a single part, and is easy to simulate. One-plane symmetry does apply, but is not needed due the simplicity of the model. An adapter contact surface was modelled to help evaluating the interaction between adapter and links. The simulation gave unrealistic deformations of the link, so this simulation was simplified by adding a fixture at the contact surface on one eye and apply load on the second. This gives more reasonable deformations, but the stress levels are very similar in the two simulation settings. Figure 34 illustrates the mesh where fixtures are green arrows and forces are pink arrows. The orange part is the modelled adapter piece.

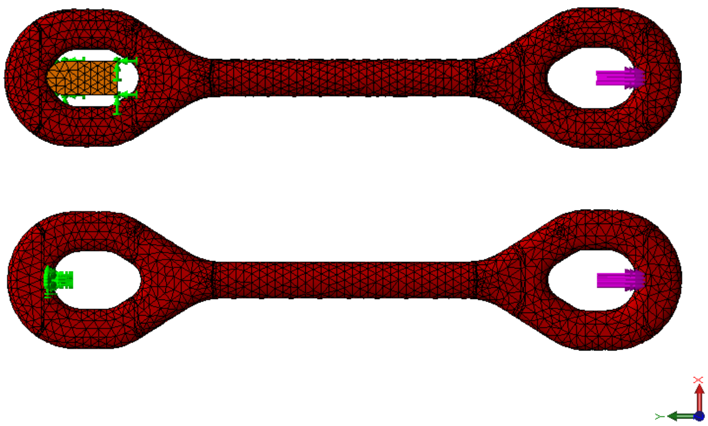


Figure 34: Link settings; mesh, fixture, and load

A curvature based mesh is used with the following mesh parameters:

- *Max element width:*	*40 mm*
- *Min element width:*	*13 mm*
- *Min number of element in a circle:*	*8*
- *Size growth ratio:*	*1.5*

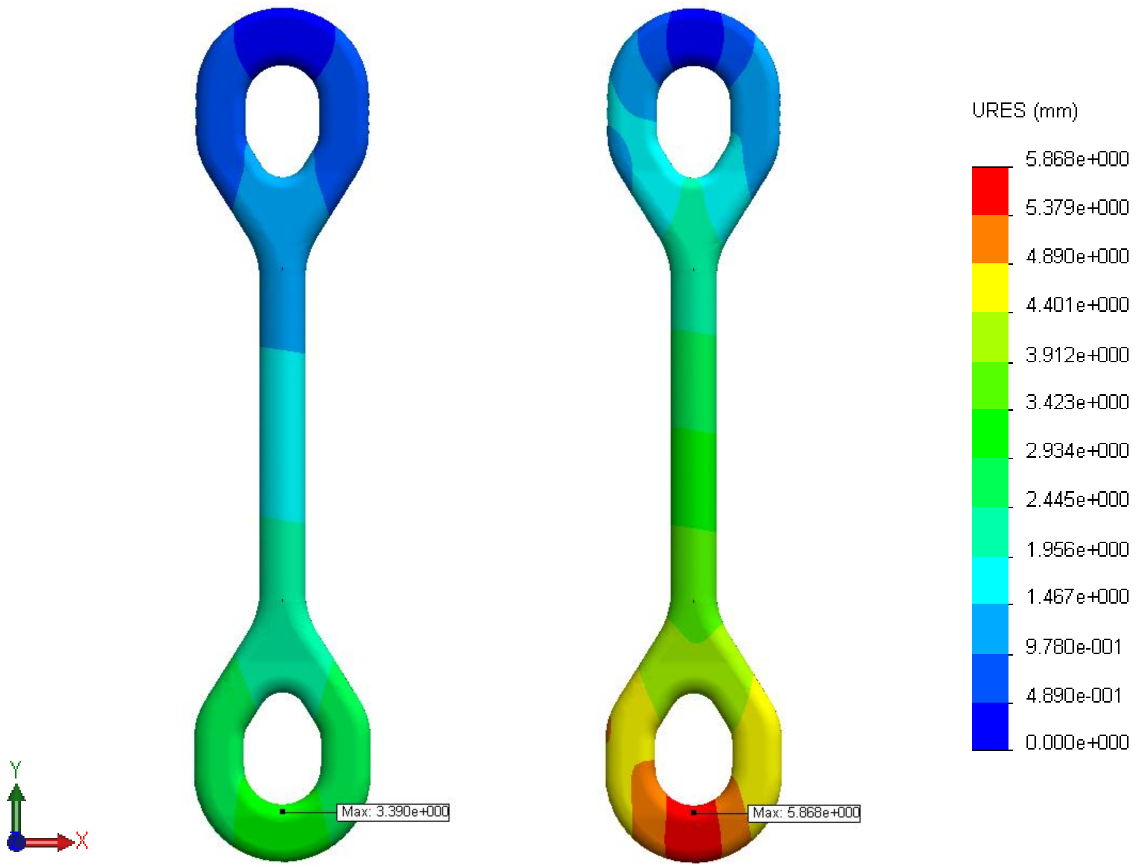
The mesh is a mixed mesh with high quality and has 14 621 elements and 23 603 nodes. The components in the study are treated as solid bodies and meshed with solid elements. Material selected for these components is 551 MPa steel.

7.3.2 Simulation Results Links

Stresses and displacement are greatest in API-load, but the heel acceptance level is so low that the cases are practically just as dimensioning. The most critical case is displayed when evaluating the results.

7.3.2.1 Link Displacement

When evaluating the link displacement, it can be seen that the links are rotated slightly around the centre axis. The main deformation is thus in y-direction. Measurements of the model show that the downwards movement in API-load condition is 5.845 mm, which is an increase of 2.468 mm from the normal condition measuring a maximum y-displacement of 3.377 mm. The relative increase of the overall length during normal operation is 1.1 ‰, while this is increased to 2.2 ‰[18] when subjected to the API-load. Even though the links length is increased by 6 mm, the relative length increase is very low. The displacements in the links can be assumed within acceptance level.



Normal:3770 kN ***API-load:5655 kN***

Figure 35: Displacement levels in Adapter at normal, API-load, and accidental condition

[18] The $u_{y,res}$ at normal is 3.2 mm, $u_{y,res}$ at survival is 5.5 mm and the overall link length is 2820 mm.

7.3.2.2 Stress in contact surfaces

An additional simulation case, including an actual contact surface, was set up to review the differences in stress level between body interference and fixture. It is differed between points that are fixed, and points that are subjected to a force. Force interaction is assumed to be more accurate than fixtures. Applied force is a good estimate for the equipment subjected to some use. This is due to the change in the contact regions and the level of contact increases. The materials in the contact zone will deform so that the surface contact between components are more complete.

The results show that the stress levels increase slightly when an additional body is included. It can also be seen that the stress level through the body is the same with and without the additional adapter piece. The changes are so small and local, so simulated stress levels for fixtures can be assumed accurate enough.

API-load
Fixed at split line surface

API-load
Resting on adapter ear

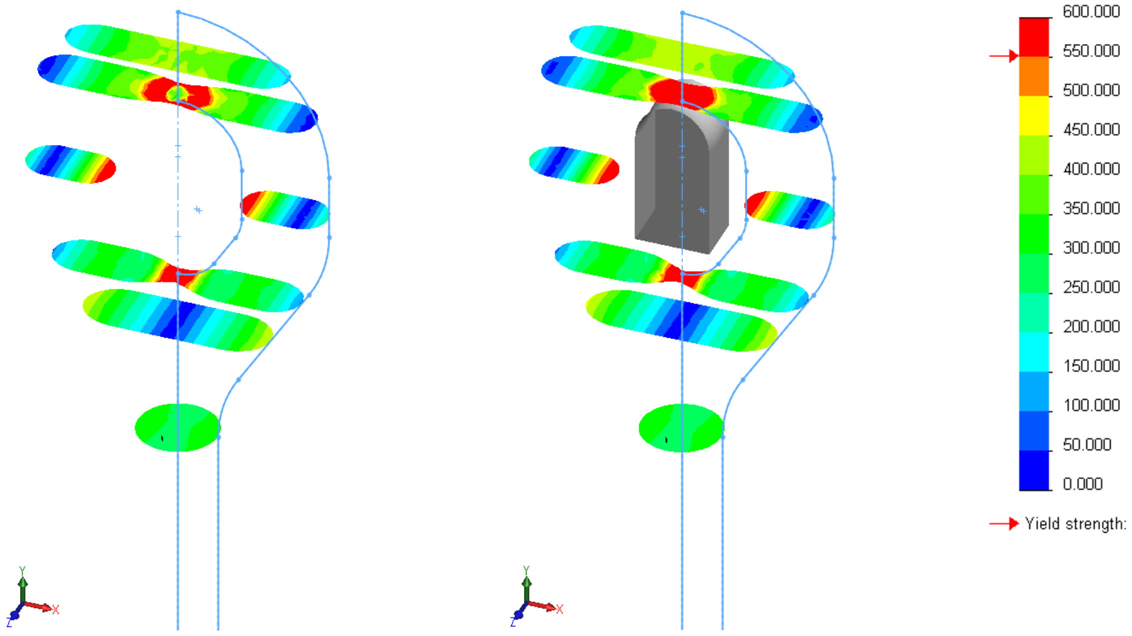


Figure 36: ISO-clippings of d link eye during API-load with different contact constrains

7.3.2.3 Von Mises Stresses in Link

Stress level in the links during normal operation is generally well below the yield limit. There is some level of yielding at the contact regions, but this is acceptable according to the API 8C standard. The DNV acceptance level during normal operation is set to 479.1 MPa. ISO clipping of the links show that the main component, except for the contact points, is within the DNV acceptance level. Case 7 calculated for normal operation gives the same value as the simulated stress level in the main beam, measured to 179 MPa. The same accuracy is found for API-load and heel conditions.

The highest stress levels occur during API-loading. Simulation show that the components experiences local yielding at most parts of the inner pad-eye. This level of yielding would be seen both in API-load and accidental heel. The most critical region is located at the upper link eye. This is consistent with the calculations. This point will be subject to further evaluation later in this chapter. Minor deformations may occur in the inner eye, but the component is not at risk of failing.

Stress concentrations occur on the inner eye at contact point, at both sides, and in the junction. In addition, there are two stress concentration points on the outer pad-eyes; one at the top (above the connection), and the other are where the eye meets the link beam.

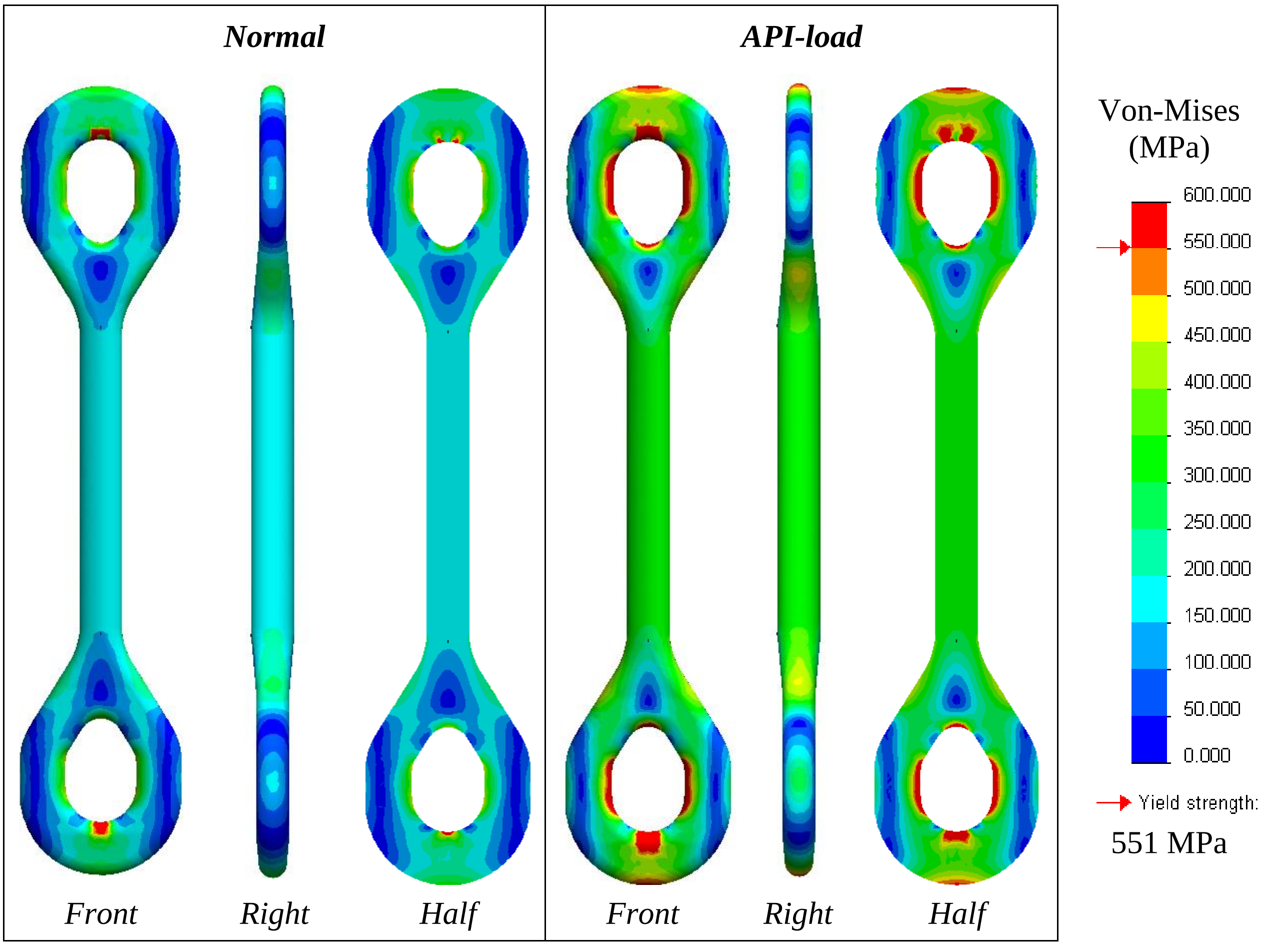


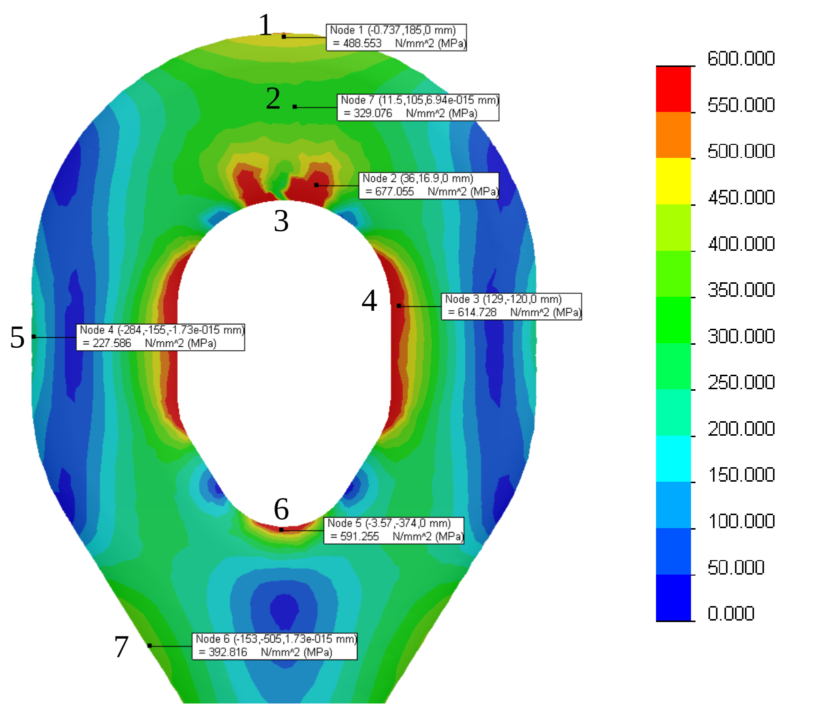
Figure 37: Stress levels in Link at normal and API-load condition

Table 11 gives some stress values over the pad-eye subjected to the three load cases. The DNV acceptance levels are stated at the top of each column. Values exceeding the acceptance level are marked red. The Some local yielding are always expected at connection point (3). The top of the eye (point 1) are within the DNV acceptance level during normal condition, but are at risk of yielding in heel and API-load. Yielding may also occur in the inner eye at point 4, and 6. Point 2, 5, and 7 are non-critical areas. Non-yielding regions are located at every critical cross section. This indicates that even though it is yielding at both sides of the cross sections at the upper part of the link eye, the middle part has more capacity, and the link is not at risk of tear-out.

These high stress regions are due to compressive loading and it is likely that some elastic deformation of the contacting surfaces would occur before yielding starts. This indicates plastic behavior and could be considered as a form of stress relief.

Table 11: Stress levels in upper eye at normal, accidental heel, and API-load condition

Accidental heel



	Normal 479	*Heel* 479	*API* 551
1	335	**490**	**575**
2	225	330	385
3	440	**680**	**685**
4	430	**615**	**635**
5	155	230	270
6	410	**590**	**690**
7	270	390	465

For the accidental case, approximately 40% of the cross section area is at risk of yielding while 60% still has capacity to take more load. A correct modelled contact region would increase the stress level, but it is not likely to increase it so much that failure would occur. The stress levels in the links are therefore acceptable in API-load and accidental heel.

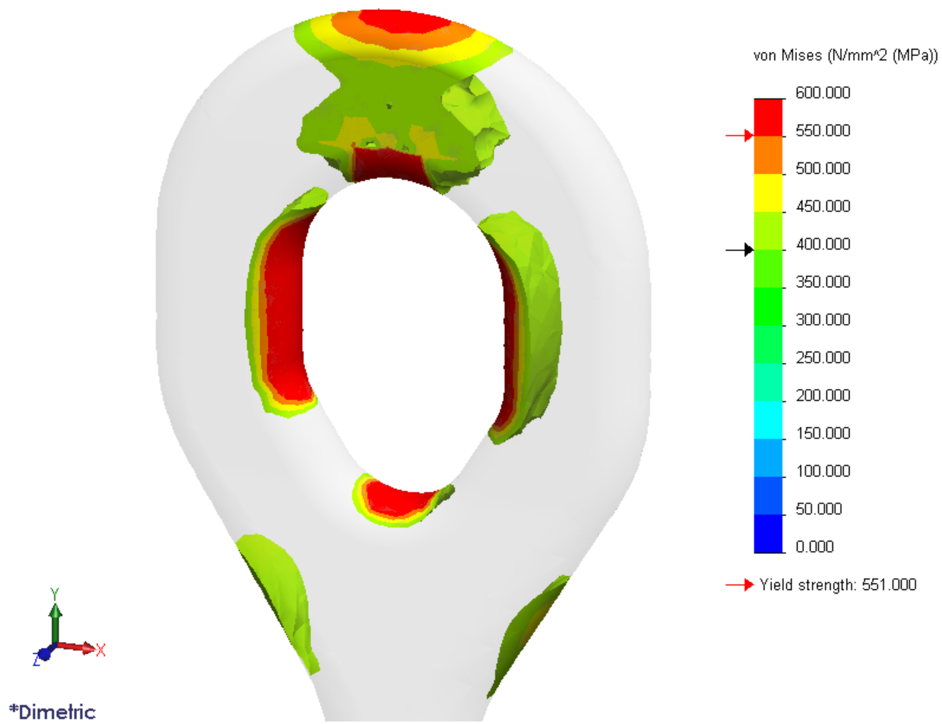


Figure 38: ISO-clipping at 400 MPa of upper link eye under API-load condition

Figure 38 show an ISO clipping of the upper pad-eye at 400 MPa during API-load. This figure proves that the most likely failure mode is tear-out at adapter connection. Figure 39 compare accidental heel at 85% of acceptance (407MPa) and API-load at 85% of yield limit (at 468 MPa). The API-load and heel condition seem to be equally dimensioning.

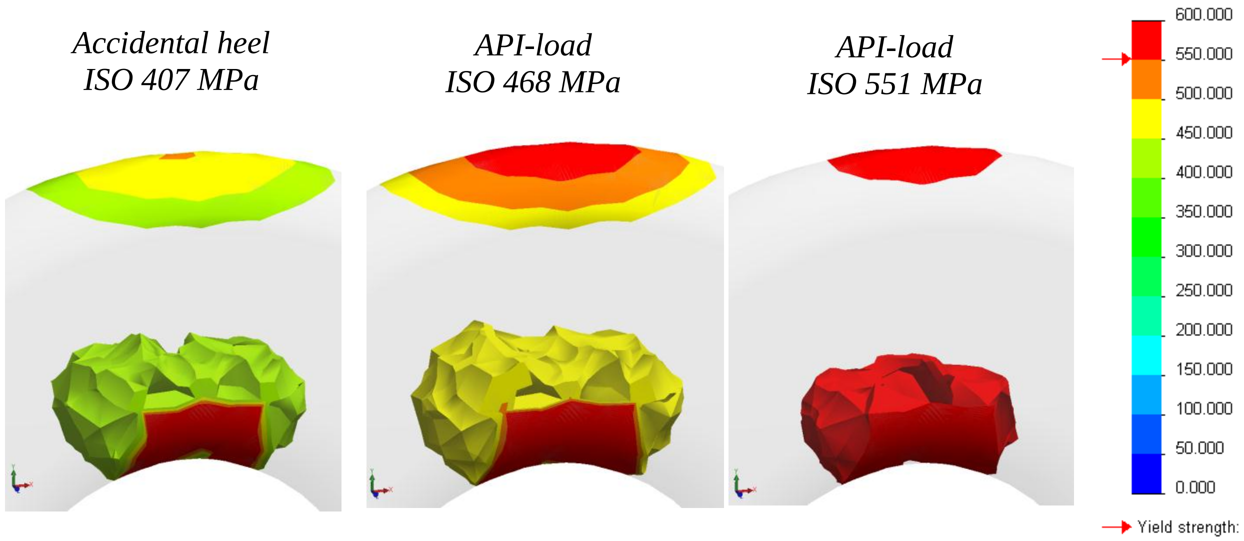


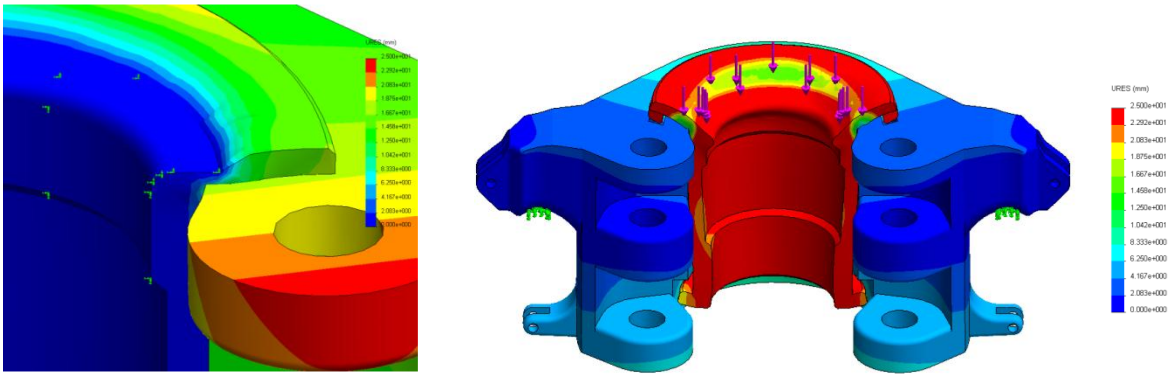
Figure 39: ISO-clippings of upper link eye under API-load and heel condition

7.4 Elevator

The elevator is a complex component to analyse. This is due to the number of elements and the different material types. The padding are especially challenging due to the level of displacement.

7.4.1 Simulation settings Elevator

The main problem simulating the elevator is the large deformations in the POM material in the padding. For the POM material where the deformations are large, a non-linear study should be used. This is not done since it is very time consuming and has little to gain considering that the padding is not a focus of the study. If fixture is added to the padding, large deformations in the complete model force SolidWorks to enter large deformation mode (non-linear). This mode makes the solver very slow and the results are difficult to read since the deformation scale are relative to the inner part of the protection cap (Illustration 1 figure 40). The simulation gives an incorrect picture of both deformations and stresses if the force is added directly to the protection cap while the ears are fixed (Illustration 2 figure 40).



1) Large displacement main body **2) Incorrect deformations of the protection cap**

Figure 40: Simulation errors, Elevator.

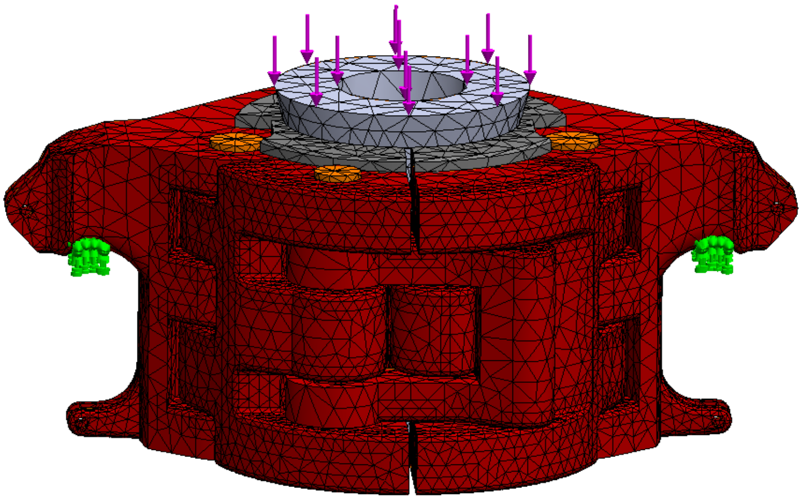
A solution is to include the riser element. To prevent the riser element form “falling out” of the elevator during simulation, the complete door assembly has to be included. The link connections are fixed so that the overall deformations are kept small, while the POM is allowed to deform fully. The riser flange and wall restrains the padding deformations. In normal and API-load, force is added to the riser head and kept normal to the top plane. In heel the force are added to the right pad-eye and set normal to the top plane. To simplify the model the bolts are assumed to be rigid and the pad-eye lock pieces are excluded from the simulation. Additional to this, the riser is assumed rigid in accidental heel.

The material used for the main elevator is 551 steel and the protection cap is a POM material. The POM material used is the one available in SolidWorks material library. Global component contacts is set to no penetration, components are treated as solid bodies, and meshed with solid elements.

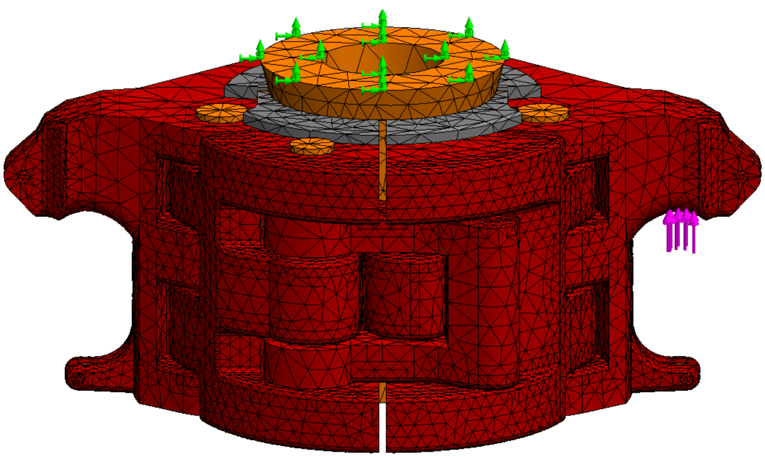
The mesh is a mixed mesh with high quality in both cases. It is curvature based with following parameters:

- *Max element width:* *80 mm*
- *Min element width:* *16 mm*
- *Min number of element in a circle:* *8*
- *Size growth ratio:* *1.6*

The component has 30 000 elements and 40 000 nodes. Green arrows are fixtures and pink are load. Orange elements are treated as rigid bodies. Bodies with red colour have yield strength of 551 MPa and E-modulus of 2.07e+11. Grey materials are SolidWorks materials: Padding is POM with E-modulus of 2.6e+09 [19], and riser is alloy steel with yield 620MPa and E-modulus of 2.1e+11.



Normal and API-load



Accidental heel

Figure 41: Elevator settings; mesh, fixture, and load

[19] Yield strength not specified.

7.4.2 Simulation Results Elevator

Using Normal and API-load simulation set-up and apply total load of the operation load 7845 kN and the API-load load of 11310kN. The accidental heel set-up has a total force of 4825 kN.

7.4.2.1 Elevator Displacement

The differences in displacement in normal and API-load are relative small. The relative overall difference is less than 1 mm over the door assembly. The biggest difference is found at the middle part of the main elevator beam. Measurements of the elevator indicate elevator beam has a max y-displacement of 3.6 mm during API-load condition, and in normal operation this is measured to approximately 2.0 mm.

The door assembly are moved approximately 6 mm by the riser. The base of the doors, follows the main beam deflection, and is shifted approximately 1.5 mm down. The front of the elevator doors are affected by the riser stack load and are moved approximately 5 mm down in y-direction. The relative movement of the doors are thus 3.5 mm in y-direction.

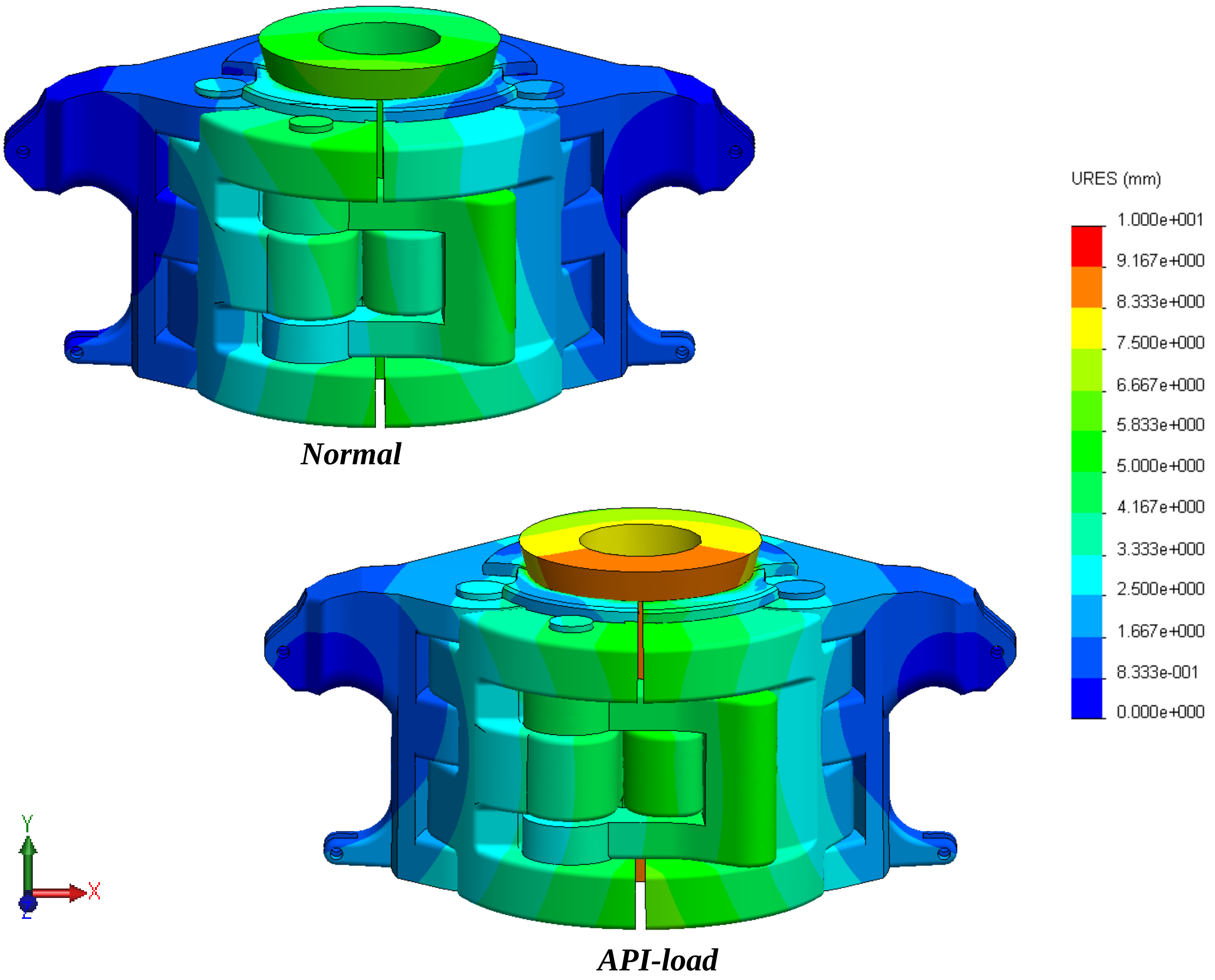


Figure 42: Displecements of Elevator at normal and API-load

The accidental heel case is very different from the two other simulations. It has a visual point of rotation located a bit higher than the centre of the elevator. This “point” goes straight through the model and is actually a rotation axis. It is clear that the whole model is shifted counter-clock wise. Elements near the axis are not moved, while the pad-eyes have displacement in both y- and x-direction.

The maximum res-displacement of the elevator subjected to heel is 25 mm at the tip of the loaded pad-eye. The other pad-eye is moved slightly less. The displacement over the door assembly is similar to what seen in the normal and API load case. In the total displacement view this is visible as a slight disturbance of the rotation point. This disturbance is not visible at the back of the elevator.

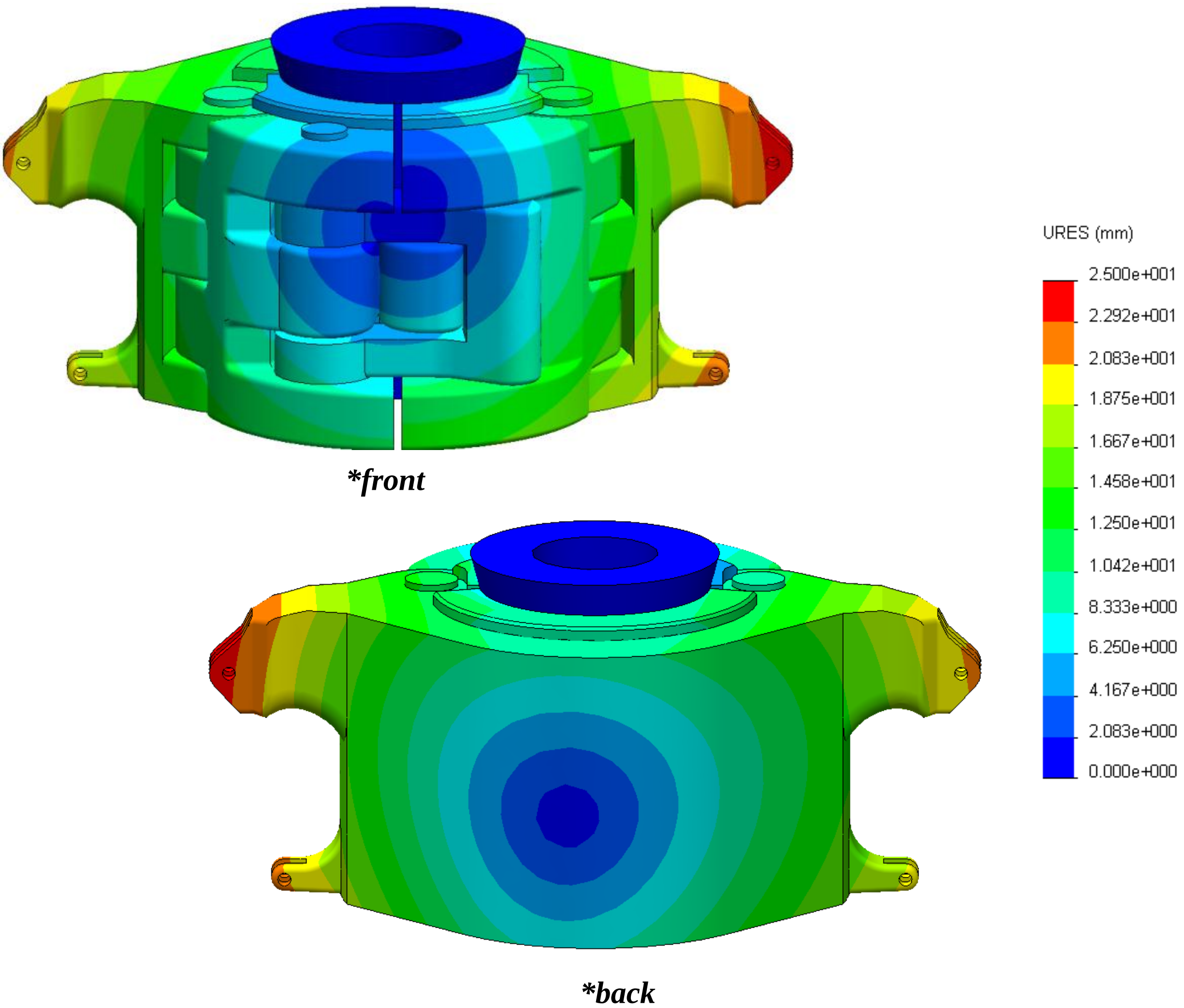


Figure 43: Displecements of Elevator at accidental heel

7.4.2.2 Von Mises Stresses in Elevator

Figure 44 illustrates the stress levels on the elevator in a non-deformed state. The simulation shows that the elevator doors do not carry any load. This is consistent with the assumptions taken when calculating the model. It is also possible to see how the lock piece interacts with the doors and prevent the riser element form falling out of the elevator. In addition to this, figure 44 also shows that the cross section at the middle of the main body is not a critical point. A more correct cross section would have been at the start of the elevator beam.

The load path goes through the main elevator body and via the elevator eyes to the links. Some local yielding can be observed in the contact regions. The elevator experiences higher stress levels that the adapter, but less than the links.

It can be seen that stresses in the overall body is reduced in the accidental heel case. This is the opposite of what happens with the adapter. The riser element is tilted so that it loses contact at one end and the load travels through only a small part of the elevator.

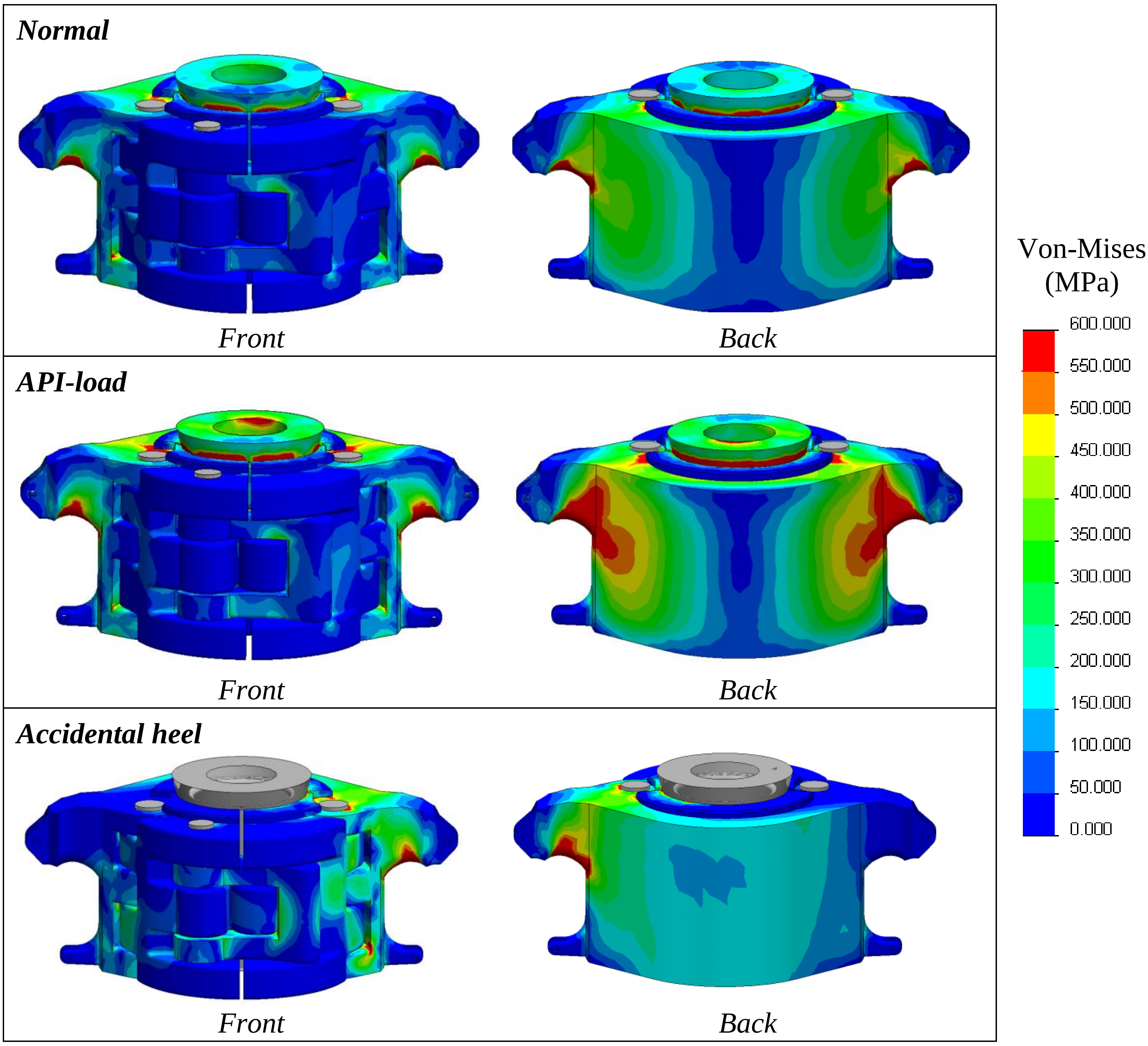


Figure 44: Stress levels in Elevator at normal and API-load condition

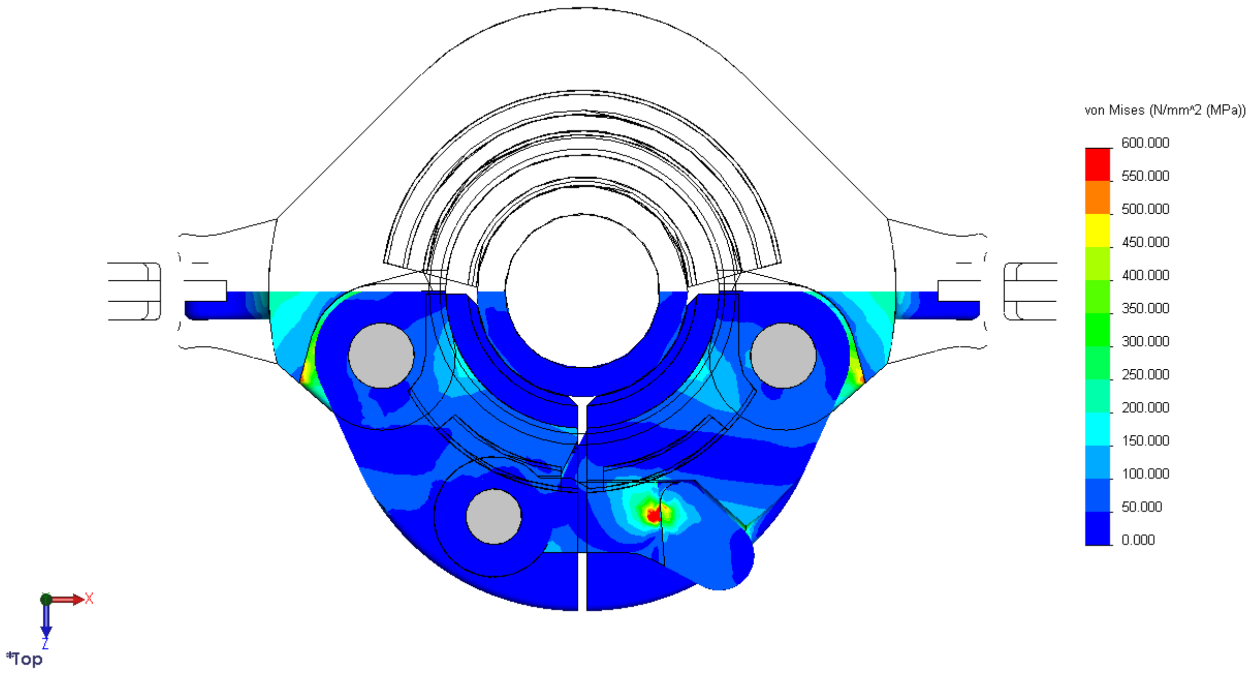


Figure 45: Stress levels in Elevator lock at API-load condition

Figure 44 and 45 illustrate how the lock piece will interact with the two doors. It is not possible to open the lock before the pressure is released. The maximum contact pressure is above yield during API-load, but no critical deformations are expected.

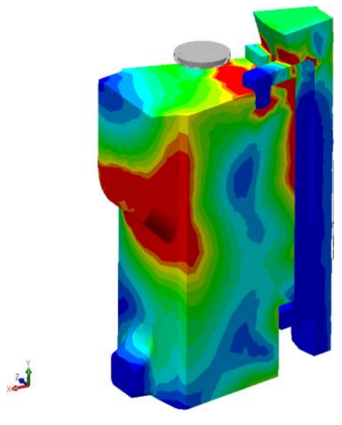
Figure 47 shows the stress distribution on the elevator cut in half. The illustration shows the component in a deformed state. The scale is 1:1. There is always some local yielding in the contact regions at the link connection and at the top flange under the riser joint.

A small difference makes accidental heel the most critical load case for the elevator eye. The API-load makes larger regions yield, but in combination whit lower acceptance level and higher overall cross section stresses make the accidental heel most critical. The API case has a minimum cross section stress of less than 200 MPa with an acceptance level of 551 MPA, while the stress levels measured in accidental heel is at least 200 and the acceptance is reduced to 479 MPa.

A cake section of the elevator shows that it is not as critical as figure 44 might suggest. Yielding occur mainly at the surface of the elevator.

Improvements for the elevator design could be to add a fillet between the main elevator body and the upper part of the pad-eye. This will reduce stress due to sudden geometrical changes.

Figure 46: Cake section elevator at Accidental heel →



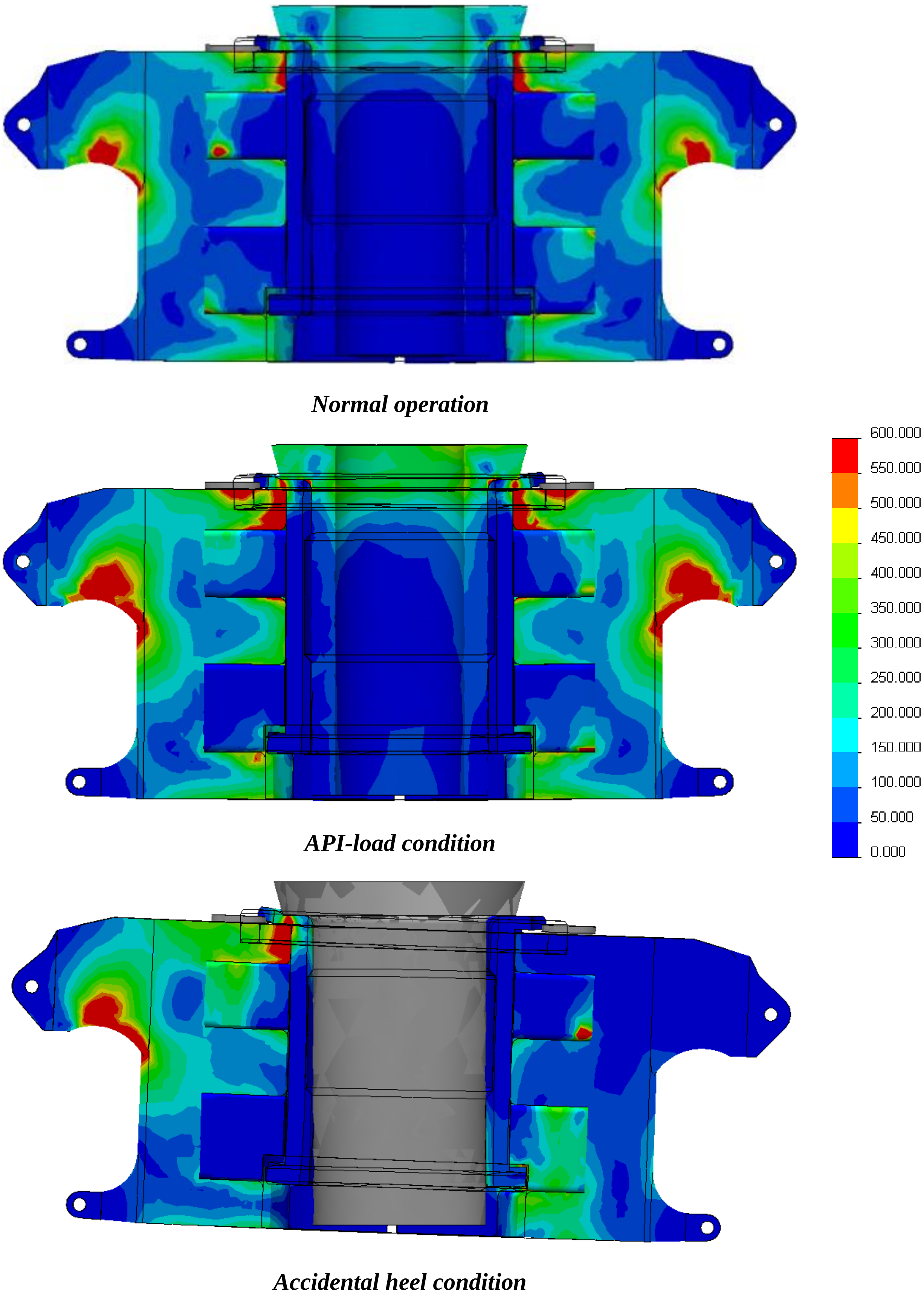


Figure 47: Half elevator assembly at normal and API-load condition

Chapter 8: Conclusion

The new riser handling system implies potentially huge savings, both in time and in money per year. The estimated time savings are in the order of 20-22 hours per trip, indicating a potential yearly saving of 50-60 million NOK. This should be a good motivation for taking the idea further.

The suggested handling sequence of storing the Upper Riser Pack (URP) in the tower has proven itself to be a plausible solution. Evaluation of the tower proves that there is sufficient space available for installing a new riser handling system. The available in-tower length is 3.2 meters. The new equipment measure 2.23 meters in total in-tower length (Adapter: 0.53m, Links: 1.7m, Elevator: 0m. See chapter 3).

The yoke is probably easier and cheaper to order than to manufacture. A suggested commercial solution is a hydraulic lifting yoke provided by National Oilwell Varco. This yoke consist of two weld-less BJ links and a BX 5 elevator.

The adapter has a tailor made design and has to be special made for this purpose. The adapter is fashioned so that modifications on the existing tower equipment are avoided.

The hand calculations and the SolidWorks simulation show that the equipment is not subjected to extensive yielding. Fatigue has also been ruled out as a problem.

Following has been done:

- ✓ Potential saving is sufficient.
- ✓ New procedures can be initiated with minor changes (adjust the alarm system)
- ✓ Space in the tower is sufficient for installing the new riser handling tool.
- ✓ Recommended solution is a hydraulic lifting yoke provided by NOV.
- ✓ Elevator and links are consistent with the NOV design.
- ✓ Suggested design for adapter has been drawn.
- ✓ Modification of the existing tower equipment is avoided.
- ✓ Strength and deformation are found on the new equipment using SolidWorks.
- ✓ The SolidWorks calculations are checked against hand calculations.
- ✓ Rules, regulations, and codes safety factors are met.

This implies that initiating the new riser handling sequence is both possible and profitable, and the new riser handling equipment has a feasible design.

Chapter 9: Further Work

New knowledge equipment on the CTTF frame makes it much heavier than stated in the gimbal calculation. The allowable weight of the riser stack is decreased by approximately 50 mT. Since the new yoke is mounted below the CTTF this weight reduction would reduce the needed capacity of the yoke from 420 to 370 mT. Modifications of the handling equipment is not needed, but should be considered

Other changes in the design might be to alter the padding material. The material used does not need to be so soft. This material selection of the padding would only lead to unnecessary ware of the padding and require too many spare parts.

It should also be investigated if increased fillet radiuses will reduce the stress levels on the hot-spots on the adapter and at the elevator eye.

It could also be of interest to look further into the investment, including production costs. If investment is proven to be profitable, then an order can be placed and manufacture drawings of the adapter can be created.

The last point, which is of importance, is to set up an action on how to adjust the alarm system in the tower to fit the new handling sequence.

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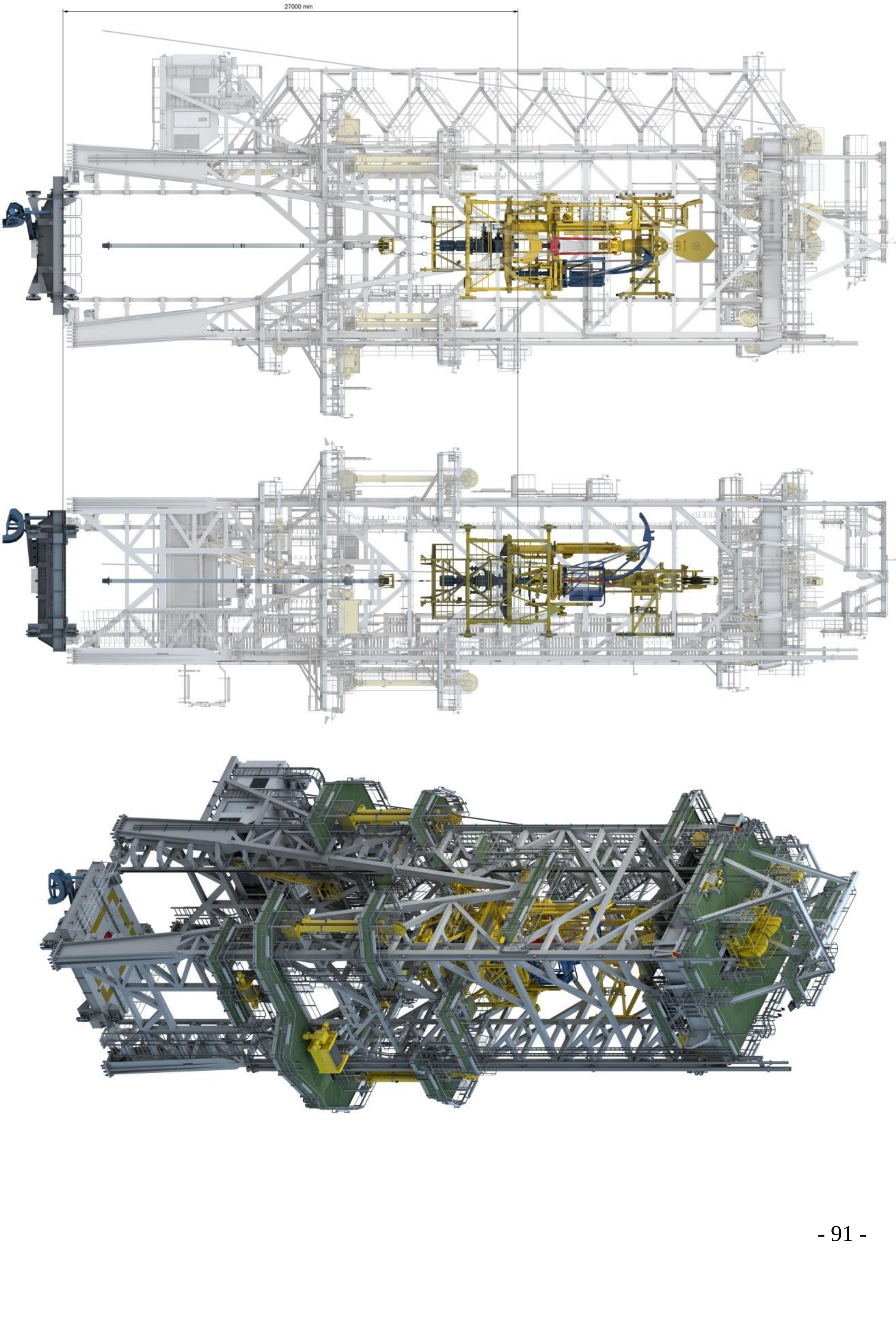
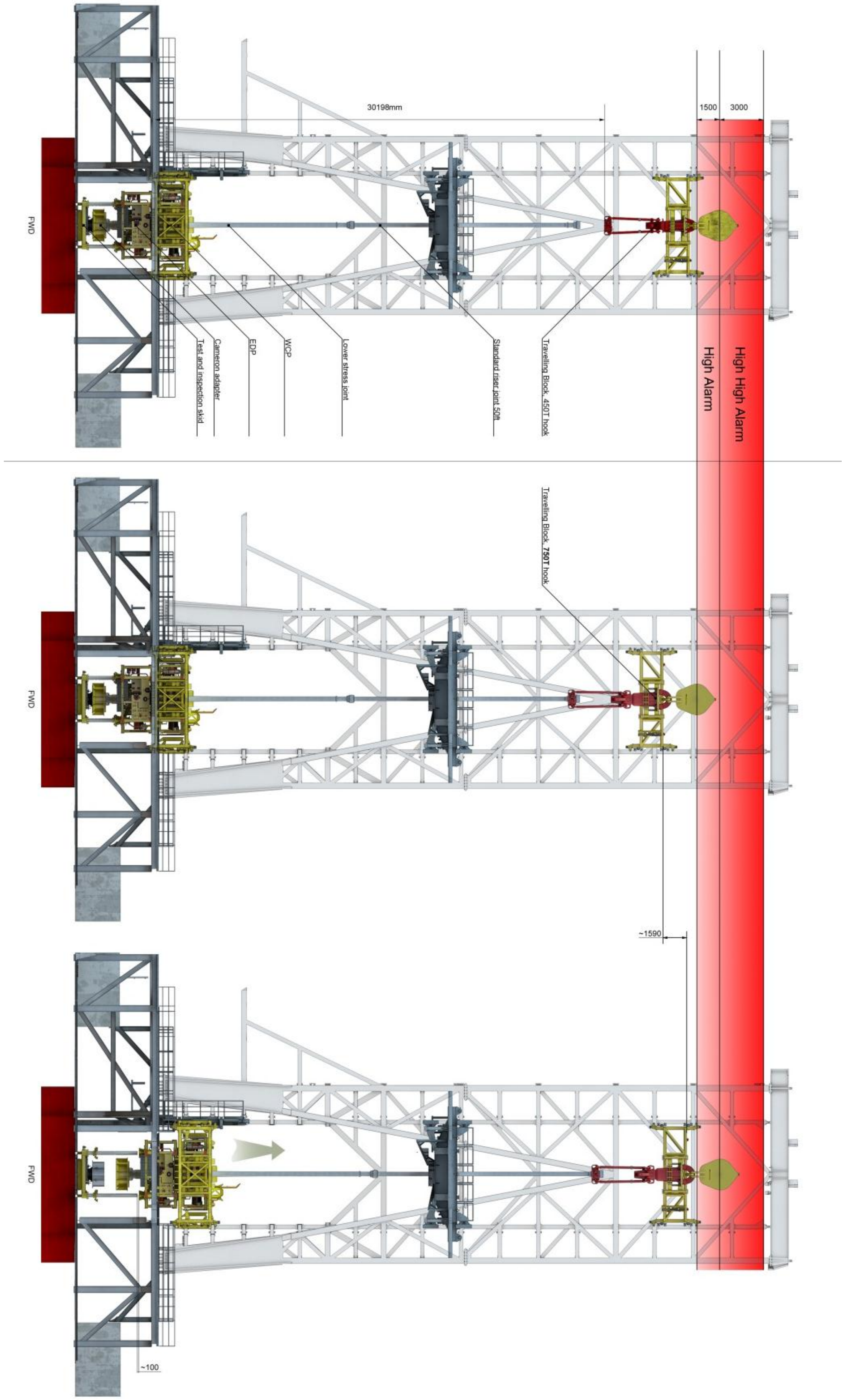
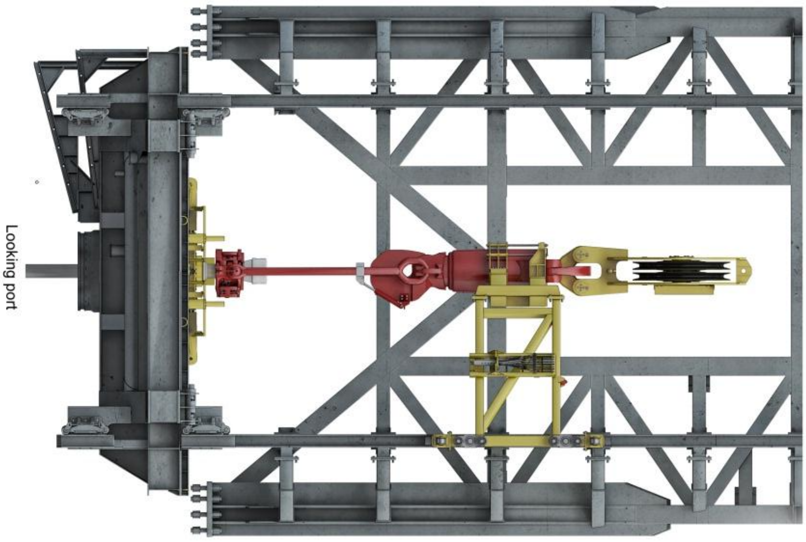
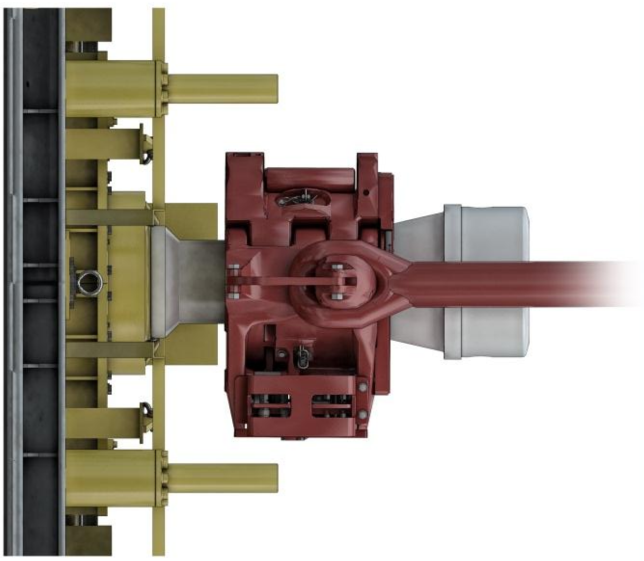
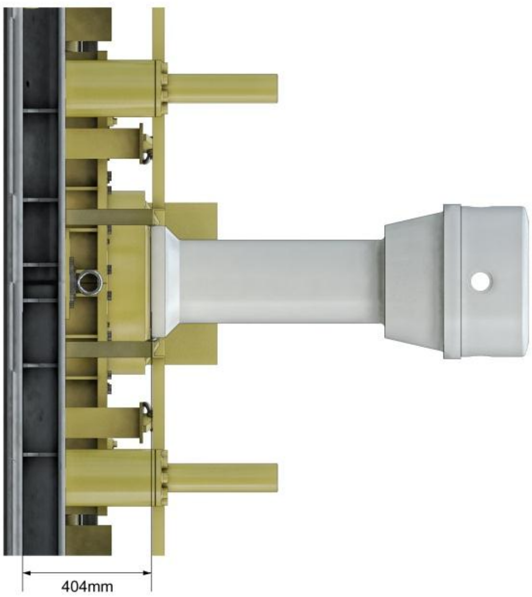
Appendix 1: Illustrations

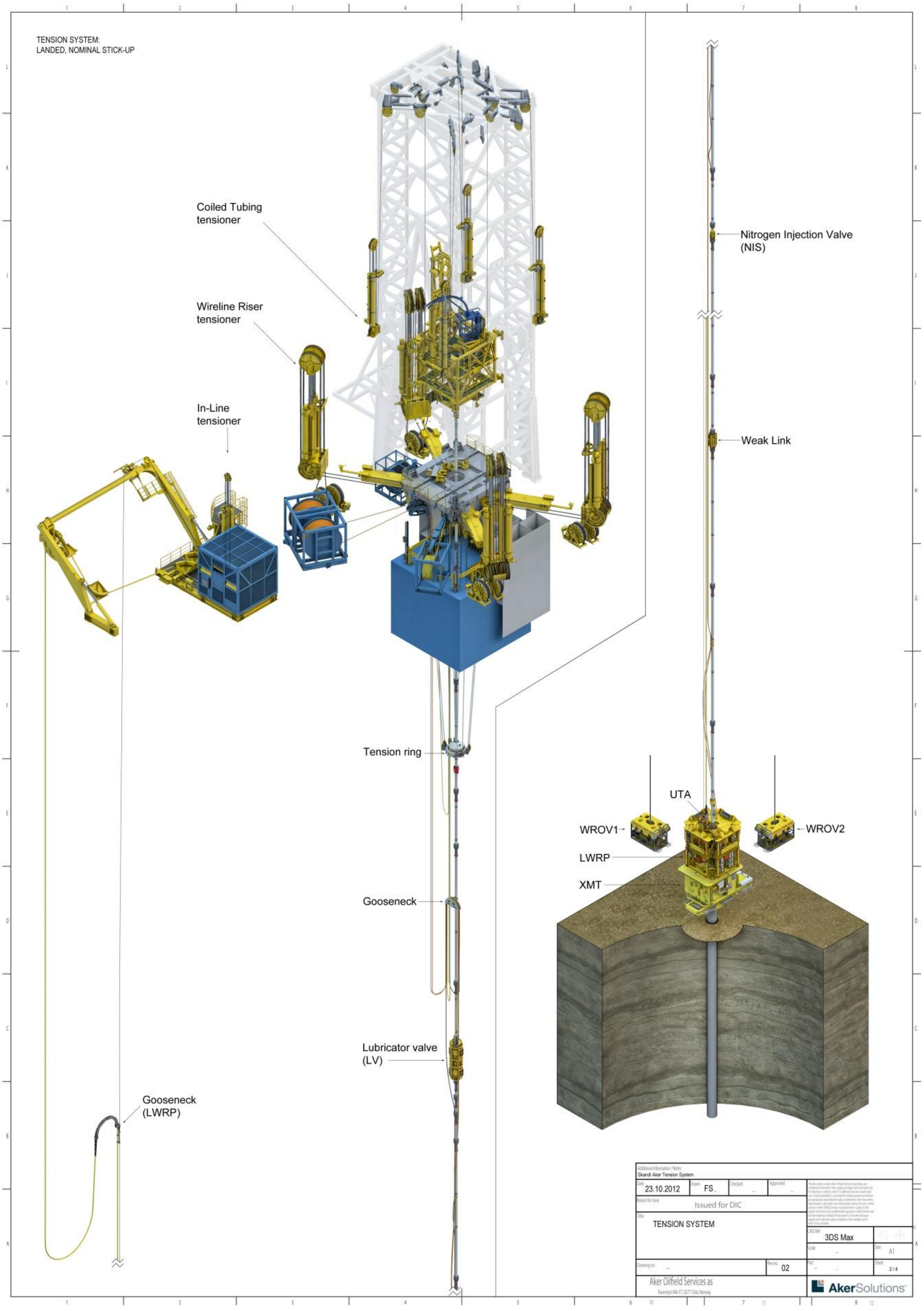



30198mm
1500
3000
High High Alarm
High Alarm
FWD
Test and inspection skid
Cameron adapter
EDP
WCP
Lower stress joint
Standard riser joint 50ft
Travelling Block, 450T hook
Travelling Block, 750T hook
~1590
~100


Looking port




404mm


TENSION SYSTEM:
LANDED, NOMINAL STICK-UP
Coiled Tubing tensioner
Wireline Riser tensioner
In-Line tensioner
Nitrogen Injection Valve (NIS)
Weak Link
Tension ring
UTA
WROV1
WROV2
LWRP
XMT
Gooseneck
Lubricator valve (LV)
Gooseneck (LWRP)
Additional Information / Notes
Skandi Aker Tension System
Date 23.10.2012
Drawn FS
Checked
Approved
Reason for issue Issued for DIC
Title TENSION SYSTEM
CAD Ref 3DS Max
Scale
Size A1
Drawing no
Rev.no. 02
Part
Sheet 2 / 4
Aker Oilfield Services as
Kanelyst Alle 57, 0277 Oslo, Norway
AkerSolutions

Appendix 2: Procedure

* RSR= Operation Manual RISER SYSTEM RETRIVAL, rev 02

The running sequence is edited after design of new equipment was made.

Running Sequence:

Step #	Procedural Description
	Prior operations: - Preparations for retrieval according to procedure are done (RSR p 15-19). - LWRP are lifted up from Xmas Tree hub according to (RSR p 20-23). - CT is lowered Present status: according to procedure (RSP p 24). - LWRP stack has been lifted of the Xmas Tree and are located approximately 4-7 meters above the Xmas Tree hub. - The vessel is moving towards safe handling zone. - The SFT is connected to the top of the Landing Joint (LJ) - Coiled Tubing PCE and Wireline PCE are all parked inside the CTTF - Deck personnel is ready to commence demobilizing the surface riser system - equipment - Tool Box Talk has been conducted describing the upcoming operations in detail. - CT stackup QServ arm is lowered to avoid collision with the tower. - This chapter can be done in parallel with chapter 6 below – Disconnection & Lift-off of the UTA (If it is to be removed)
	Rig down Landing Joint
1.	Prepare for rig-down LJ: - Ensure the slings from the SFT Collar to the Landing joint lifting collar are connected - CLOSE the two hatches using the hand operated winches. - CTTF tensioners are fully extended - Riser tensioners are active with reduced tension. Tower Main Winch holds most of the load - Remove hydraulic houses, electrical cable and T-bar from Riser Spider. - Override the elevator / spider interlock from the Cyber Chair - Perform Drop survey of Tower and CTTF - Lower Cursor Frame (LCF) is hung off on dogs in moon pool. slack off Cursor winch wires to allow additional clearance for CTTF. - Ensure DFMA in parked position. - Confirm that the Hydraulic work basket is properly parked - Confirm that the CTTF Access Basket position is parked in upper position
2.	Verify top tensioners is fully stroke out when SFT datum is in position (23m over work floor)
3.	Remove the Landing Joint centralizing Guide Bushing from Riser Spider (if used)
4.	Lift up CTTF such that the riser joint (RJ) below the LJ is aprox. 2m over Spider Jaw.
5.	Land the RJ below the Landing Joint in the spider and lock as per std procedure ref Attachment 26.3

6.	Disconnect landing joint (LJ) from riser joint (RJ): - Disconnect connections between landing joint and RJ in spider as per std. procedure ref Attachment 26.3 - Secure end of LJ with CWM PTA (pipe tile-in arm or use the WFMA as required) - Install hole protection cover on top of the RJ suspended in the spider - Install LJ pin protector
7.	Disconnect landing joint (LJ) from SFT: - Disconnect Landing Joint from SFT - Confirm that the two swivel locks are locked - Install SFT pin protector - Land LJ box end onto CWM Trolley with CTTF / SFT - Lower CTTF / SFT whilst the CWM trolley is pulling back until LJ is laying - horizontally on the trolley - Disconnect pick-up elevator to links from landing joint - Install LJ box protector - Disconnect pick-up elevator to links from SFT
8.	Before leaving the CTTF – perform a drops survey to ensure no loose items have been left behind **NOTE** – Any left tools / equipment left behind shall be put in a storage box that is properly secured to the structure
9.	Exit the CTTF via the CTTF Access Basket – elevate and park in upper parking position
	Install handling equipment (Lifting Yoke)
1.	Install Adapter - Connect adapter installation guide to adapter neck - Fasten guide slings to SFT - Follow same procedure as for connecting landing joint - Perform visual check to confirm the adapter is securely connected to SFT
2.	Install lifting yoke - Lay down the lifting yoke on work floor under SFT - Secure with wire between yoke and SFT - Lock and secure yoke to SFT - Raise yoke to vertical position - Make sure not to hit the Landing joint.
	Retrieval of Tension Joint
1.	Connect the yoke to the RJ neck as hung off in the spider
2.	Hoist the (50ft HW) riser slowly up with the elevator and lock in spider **NOTE**: The tension ring should now be accessible from the main deck
3.	Park TJ and TR by: - Follow point 6-19 - LOCK TR to dummy driver housing UN-LOCK TR from TJ
4.	Release TJ from Riser Tension Ring and pull TJ body up through WF
	Remove Riser Joints
	Follow procedure in attachment 26.3 Standard procedure for retrieving Riser Joints (RSP p 52)
1.	Lock LY to RJ
2.	Elevate riser system and lock&secure next riser joint in the spider jaw

3.	Secure RJ and prepare for disconnect
4.	Disconnect RJ
5.	Place RJ on CWM trolley - Use CWM arm to secure the RJ - Lower RJ (use the tower elevator system and lower all the surface equipment) Use CWM arm to guide RJ to trolley
6.	Repeat until desired length is reached

Retrieve Riser Joints:

Step #	Procedural Description – **Standard Procedure for Retrieving Riser Joints**
	Present status: according to procedure (RSP p 52). - LWRP stack has been lifted of the Xmas Tree and are located approximately 4-7 meters above the Xmas Tree hub. - The vessel is in safe handling zone. - Cargo Rain Crane (CRC) and Cat Walk Machine (CWM) are ready - The SFT is connected to the top of the Landing Joint (LJ) - Coiled Tubing PCE and Wireline PCE are all parked inside the CTTF - Deck personnel is ready to commence demobilizing the surface riser system - equipment - Tool Box Talk has been conducted describing the upcoming operations in detail. - CT stackup QServ arm is lowered to avoid collision with the tower.
	Retrieve Riser Joint
1.	Prepare for retrieve RJ: - All relevant tower equipment are available and working - Protective covers are installed on RJ-pin and box ends are available - The main winch carry all weight of riser stack during retrieval - Storage location is prepared - Catwalk Machine is in position - Confirm communication
2.	Connect elevator yoke to the riser joint hung off in the spider. (Simultaneously: Remove last RJ from CWM using CRC in the storage area) Make sure that: - Riser clams are removed - Umbilical and line reels are manned and set in lowest practical tension mode, ready to real in while the riser is being hoisted - Moonpool area is cleared from personnel
3.	Retrieve new riser - Pick up riser to approximately 0.5-1.0 meters above spider. - Unlock and Open spider - Check that riser can be elevated without any snagging against spider or Clamping platform - Elevate riser joint. Start slow-increase speed gradually- slow down when next RJ is through the spider. - Lift riser stack until next riser joint is approximately 0.5-1.0 meters above Work Floor. - Close, Lock & Secure the spider jaw. Visually confirm lock indicator

	- Lower down the elevator and hang-off riser joint in the spider. NOTE: Pay attention to snagging e.g. closely follow the weight indicator
4.	Disconnect the RJ connection - Turn RJ retaining sleeve until the four bolts are accessible - Undo the 4 riser connection dogs - Lift of the RJ - Guide the RJ with the CWM PTA and WFMA away from the well centre. (dropped objects) - Re-install the bolts on the RJ hanging in the elevator. Avoid dropped objects!
5.	Inspect and install protection caps - Inspect pin end seal and seal surface and install pin protector of <u>riser in spider</u> - Inspect box, seal, seal surface, dog position and install box protection <u>of riser in elevator</u> Remove Umbilical riser clamp in MP
6.	Position the RJ onto the CWM trolley - Skid CWM next to spider - Raise CWM lift arm - Hoist elevator to lift RJ onto CWM trolley while using PTA to secure box end - Land the riser on the CWM slide - Release CWM PTA - Move the CWM Slide back and lay down the RJ horizontally - Disconnect the elevator yoke from the RJ
7.	Connect the elevator to the riser element currently hanging in the spider
8.	Sore the RJ - Skid the CMW to the storage area - Lift of the Riser Joint using the Cargo Rail Crane gripper - Lay the RJ in the riser bay NOTE: Make sure that it is stored at a different position than it had when running the riser. This to ensure that the riser elements are rotated within the stack when running
9.	Repeat procedure until desired length is reached

NOTE1: TJ and TR are temporarily stored on deck and do not need to be removed.

NOTE2: When installing the riser system, the LV is placed further down than the normal standard procedure. This enables the team to only remove RJ above the LV.

NOTE3: The CT stackup QServ arm is lowered to avoid collision with the tower

Equipment list:

Equipment list: top to seafloor

1. Tower beam derrick
2. Elevator system frame/ Cursor Frame
3. Dynaplex hook
4. Lifting yoke (hydraulic provided by NOV)
5. Lifting yoke (connected to gimbal in CTTF)
6. Upper CTTF
7. Lower CTTF
8. SFT
9. Landing joint (LJ)
10. Riser Joint (RJ)
11. Tension Joint (TJ) with Tension Ring (TR)
12. RJ
13. Lubrication valve (LV)
14. RJ (heavy wall)
15. Nitrogen Injection System (NIS)
16. RJ (Light wall)
17. Weak link
18. RJ
19. Lower Workover Riser Package (LWRP)
20. X-mas Tree (XT)

Appendix 3: Unit Acceleration

Accelerations for handling equipment in tower. Extracted from document: 705-AKOFS-KZ-00-002 *Design Loads Skandi Aker*.

state:	Riser Removal Limit State *	Ultimate limit state Transit
Probability level		20 year return period
Significant wave height	<3 meters	
return period	3 hours	
spreading	cos^2	
Heading	+- 30 deg	+- 180 deg
Draft	8 meters	6-8 meters
Metacentric height	4 meters	0.5 meters
Accelerations	**Tower COG (21.48 m over deck)**	**Tower bottom ****
lognitudal (P)	0,65	3,43
transversal (R)	1,3	5,54
vertical (H)	0,61	4,75
Value combination	m/s2	m/s2
H + P	1,26	8,18
H + R	1,91	10,29
H + sq(P^2 + R^2)	2,06	11,27

* Conservative estimate. True Limit State is 2 Hs

** hang-off condition. Spider locked.

Dynamic load at significant wave height set to 2 meters

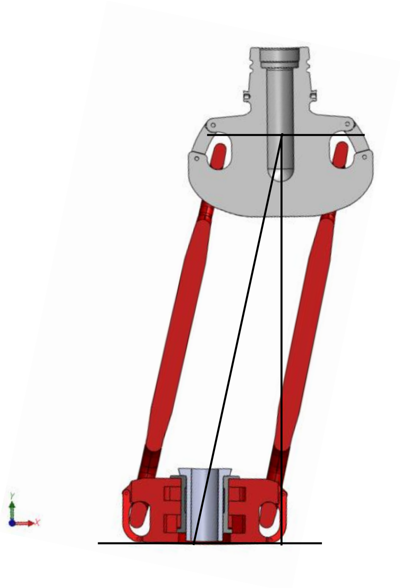
Force in still water condition			
Safe working Load hook	SWL-hook	450	mT
Tower equipment	W-equipment	30	mT
Safe Working Load	SWL	420	mT
load SWL	F-swl	4 120	kN
DAF still water operation	DAF	1,15	-
load still water operation	F-still water	4 738	kN
Additional force due to acceleration			
acceleration max	a-max	2,06	m/s2
mass top equipment	m top	5	mT
vertical acceleration	a-vertical	0,61	m/s2
mass riser stack	m riser stac	420	mT
Vertical dynamic force	F-2Hs	267	kN
Total force *	**F-tot**	**5 005**	**kN**
New DAF for 2 Hs **	**DAF**	**1,22**	

* conservative, Still water + additional load

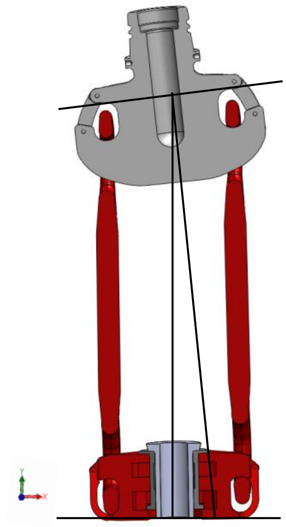
** rounded up

Appendix 4: Accidental heel evaluation

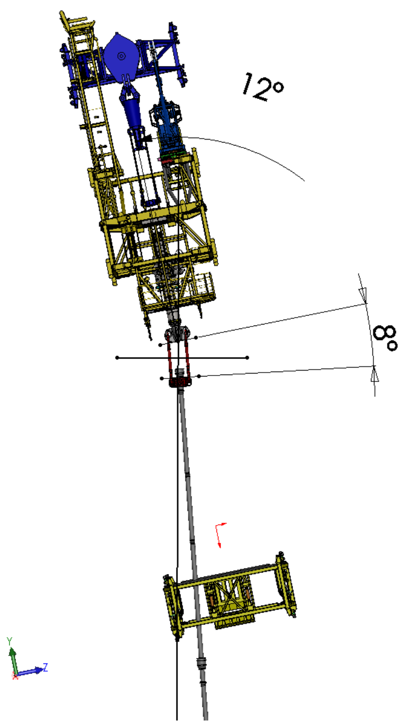
Model evaluation shows that the riser element will collide with the moonpool if the ship experiences approximately 12 degrees angel of heel. The yoke will not have problem with loosing contact if the relative angle between the adapter and the elevator/riser are small.



12 degree movement off center
→Both connections still intact



5 deg realtive movement
→ Loss of connection



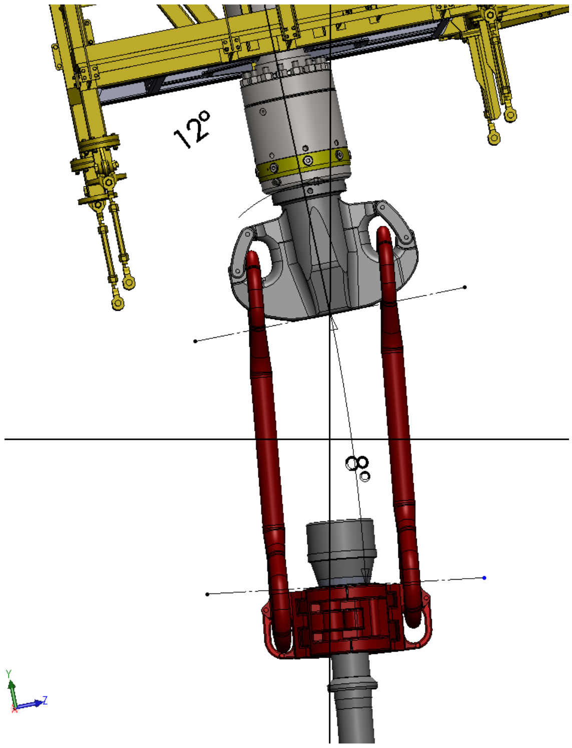


Illustration of vessel with 12 deg heel. Spider not locked. With spider locked max angle of the riser would be 4 deg. Then the riser element would bend under the locked spider.

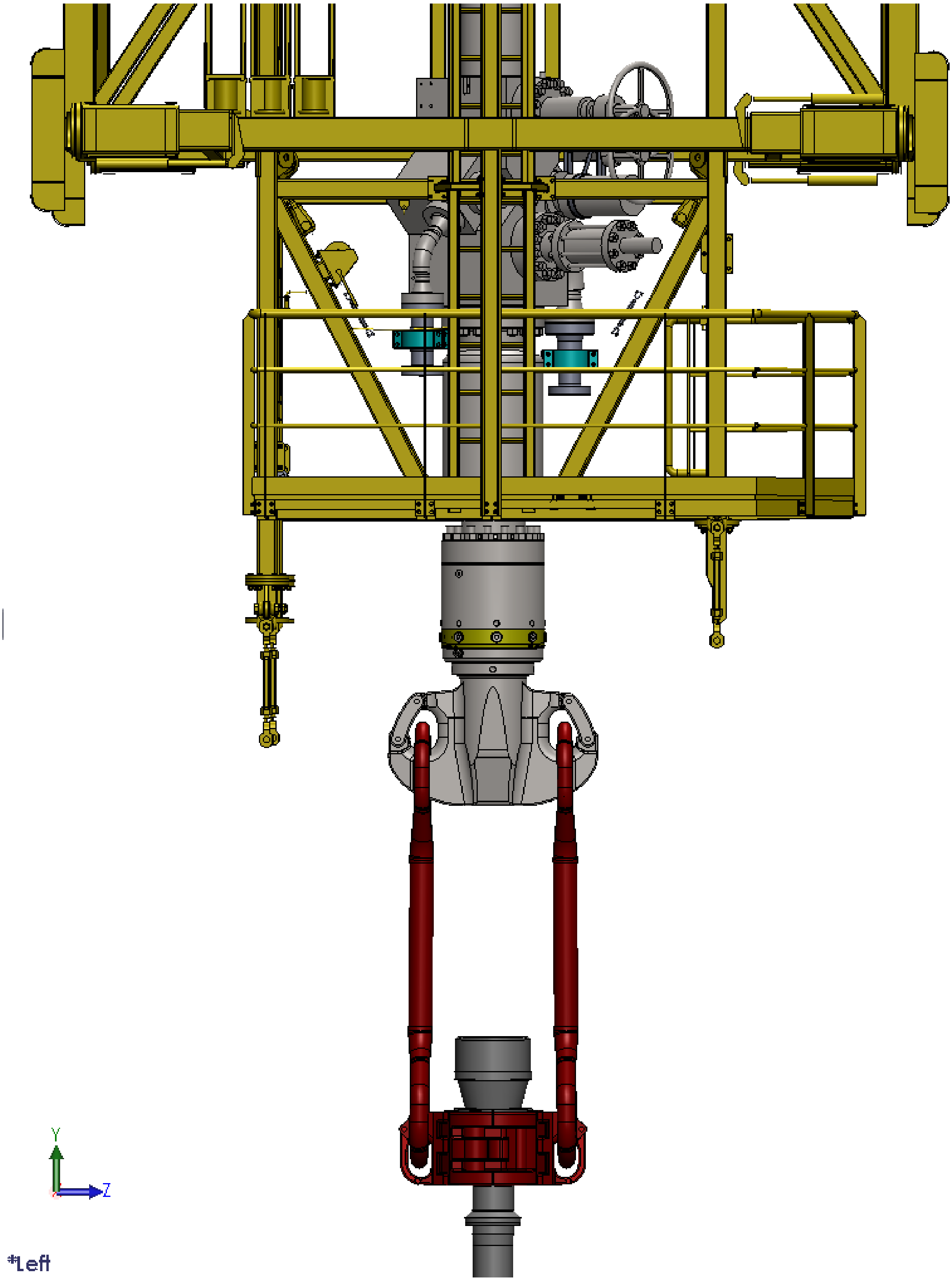


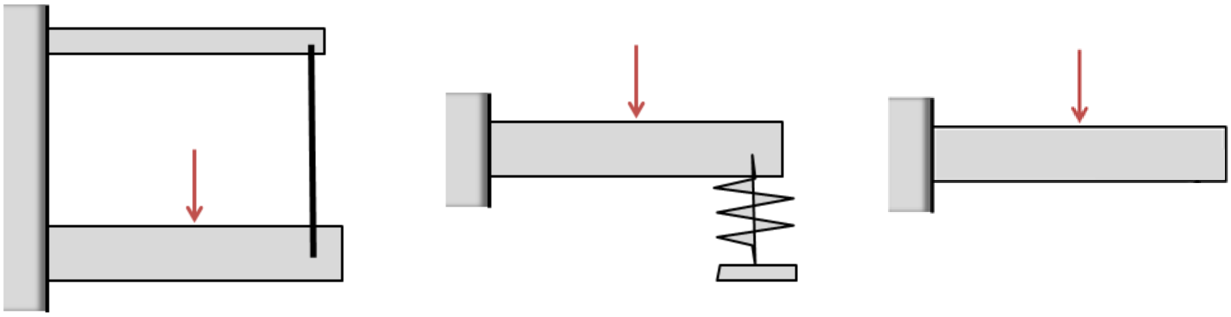
Illustration of new handling system connected to the Surface Flow Tree in normal condition.

Appendix 5: Design Summary

Reference specification	**Description**	**Value**	**Unit**
Class requirements		DNV-OS-E101	
Design specification		API 8C	
General data:	**Description**	**Value**	**Unit**
Design lifetime		20	years
Average time in use		20% of time above	
Distribution of operation mode	Time used are split in following modes: - Lifting entire riser stack - Handling riser element	% of the time above 50% 50%	
Design temperature	Ambient air temperature	-20 to 40	^oC
Sun exposure	Direct equatorial sun radiation		
Corrosion	Surface protection/coating	NORSOK	
Operation area	(see vessel Zone chart)	Zone 1 & 2	
Manning	The handling tool shall be manned during operation and can be un-manned during transit.		
Load data:	**Description**	**Value**	**Unit**
Load from existing elevator	Safe Working Load (SWL)	450	mT
	Rated Load (R)	550	mT
Equipment in tower	Equipment weight in tower	30	mT
New load case:	Safe Working Load (SWL)	420	mT
Indirect Loads:	**Description**	**Value**	**Unit**
Acceleration extreme operation: (Hs=3) (21.5 m above MD)	Longitudinal	0,65	m/s^2
	Transverse	1,3	m/s^2
	Vertical	0,61	m/s^2
Acceleration transit Ultimate: (Hs=17) (21.5 m above MD)	Longitudinal	3,43	m/s^2
	Transverse	5,54	m/s^2
	Vertical	4,75	m/s^2
Weather limitations:	**Description**	**Value**	**Unit**
Handling equipment operation	Hs = Normal operation	2	m
	Hs = Absolute Limit normal operation	3	m
Handling equipment transit	Hs = Ultimate Limit State transit	17	m

Appendix 6: Additional pad-eye calculations

The pad ears can be modelled as a two beam system connected with a rod on the edges. The upper beam and the rod can be simplified to a spring system. From this system it can be shown that the upper beam only takes a small fraction of the total load, and are therefore negligible.



Basis formulas:

Hooks law: $$\sigma = E\varepsilon \quad (1)$$

Stress: $$\sigma = \frac{F}{A} \quad (2)$$

Deformation: $$\delta = \varepsilon L \quad (3)$$

Beam deformation: $$\delta = \frac{FL^3}{3EI} \quad (4)$$

Spring stiffness $$F = k \cdot \delta \quad (5)$$

Calculating rod deflection ΔL_r

(1) and (2) combined: $$\sigma_r = \frac{F}{A_r} = E_r\varepsilon_r$$

Re-write and multiplied with L: $$\frac{FL_r}{A_rE_r} = \varepsilon_r L_r = \delta_r$$

Assuming small deformations (5) $$F = k_2\delta_2$$

Re-write together with (5): $$\delta_r = \frac{FL_r}{A_rE_r} = \frac{(k_2\delta_2)L_r}{A_rE_r}$$

Deformation upper rod: $$\delta_r = \frac{(k_2\delta_2)L_r}{A_rE_r} \quad (6)$$

Calculating upper beam stiffness k_2

Spring stiffness upper beam (5):

$$k_2 = \frac{F}{\delta_2}$$

Beam deformation (4):

$$\delta_2 = \frac{FL_2^3}{3E_2I_2}$$

Spring stiffness beam 2:

$$k_2 = \frac{\mathrm{F}}{\frac{FL_2^3}{3E_2I_2}} = \frac{3E_2I_2}{L_2^3} \tag{7}$$

Calculating lower beam stiffness k_1

Newton:

$$F_1 = F_2 \tag{8}$$

(7) together with (4):

$$k_1\delta_1 = k_2\delta_2$$

Assume small deformations:

$$\delta_1 = \delta_2 + \delta_r \tag{9}$$

Re-written and combine (8):

$$k_1 = \frac{k_2\delta_2}{\delta_1} = \frac{k_2\delta_2}{\delta_2 + \delta_r}$$

Insert ΔL_r (6):

$$k_1 = \frac{k_2\delta_2}{\delta_2 + \frac{(k_2\delta_2)L_r}{A_rE_r}}$$

Spring stiffness beam 1 in relation
to spring stiffness beam 2

$$k_1 = \frac{k_2}{1 + \frac{k_2L_r}{A_rE_r}} \tag{10}$$

Combine (9) with (6):

$$k_1 = \frac{1}{\frac{1}{k_2} + \frac{L_r}{A_rE_r}} = \frac{1}{\left(\frac{L_2^3}{3E_2I_2}\right) + \frac{L_r}{A_rE_r}}$$

Spring stiffness beam 1:

$$\boldsymbol{k_1} = \left(\frac{L_2^3}{3E_2I_2} + \frac{L_r}{A_rE_r}\right)^{-1} \tag{11}$$

Inserting different values show that k1 is small and the not loaded beam is negligible.