Subsea Tie-In
Design Solutions and Optimisation Methods

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IKM Ocean Design AS-2015

University of Stavanger

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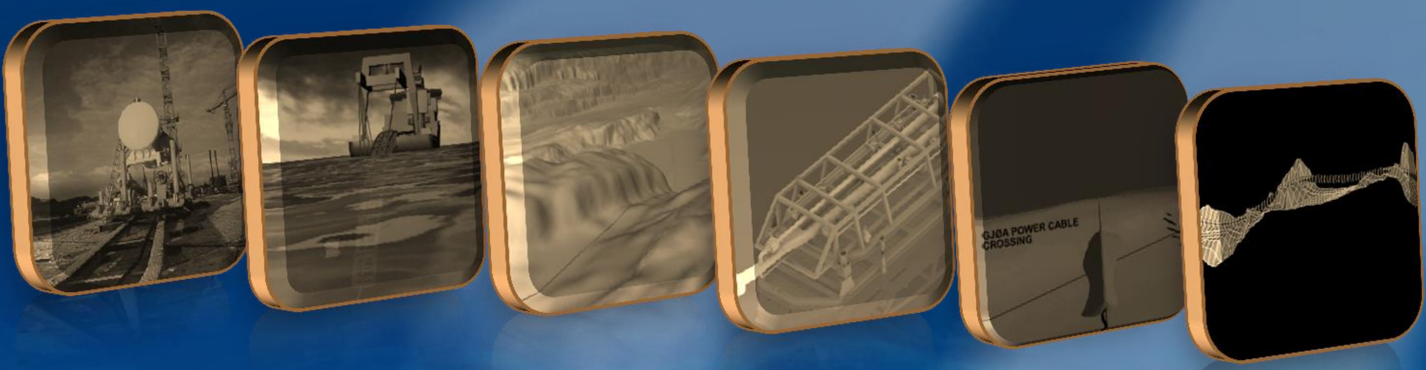

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Abstract

Subsea Tie-in Systems is used to connect pipes between subsea structures in the offshore oil and gas industry. Subsea Tie-in system has been developed for many decades by the industry, and is used throughout the world in many subsea oil and gas fields of today. Industry experience and various piping codes have been developed and used in the design over the years. However there has been a lack of recommended standard practice and guideline for designing such systems. In this thesis two computer software analysis packages commonly used in the industry for structural analysis of piping systems is explored and compared. A vertical spool design case is investigated by the use of finite element analysis. Relevant design load cases are identified and a design basis is established for the analysis. Relevant piping codes such as ASME and DNV are used in the design. Some of the main challenges which have a great influence on rigid spool design are the fabrication tolerances and metrology, which has to be accounted for in the design. This thesis gives proposal on how to implement statistical distribution of tolerances in the analysis by use of design exploration tools included in the ANSYS Software package. Advantages and disadvantages are described.

The thesis will present some theory and examples to gain a general understanding about the content to be presented. An Introduction of the most common Tie-in Systems and their basic configurations and shape is presented. Advantages and limitations are described. Recommendations and suggestions for future spool design solutions and load mitigations are given.

In this thesis a vertical spool has been analysed with a statistical and probabilistic approach for the metrology and tolerances, the results shows that it is beneficial to include such method in order to better document the safety level and the conservatism in the spool design. The approach also allows the engineer to make a better decision towards the optimisation process of the spool.

The thesis also shows that simple mitigation measures for a vertical spool such as pre-bending and introduction of a seabed support and buoyancy onto the spool has positive effects by reducing the resulting bending moments at connector ends, and can reduce the total stresses in the spool. The results also show that the vertical spool design is is very sensitive to VIV and hence fatigue capacity governs the design.

The vertical spool design has also been checked by use of the commercial piping software package AutoPIPE from Bentley. The results compared to the ANSYS analysis shows that there is a minimal difference in utilisations when using pipe beam element technology. The software is found to be feasible for usage in subsea spool design for small to moderate displacements and deformations, however for an optimised weight design it is recommended to perform a FEA with solid element models in ANSYS.

Preface

This thesis summarizes my post graduate master's degree at the Dept. of Mechanical and Structural Engineering and Material Technology at the University of Stavanger during the spring of 2015. The Thesis has been written in co-operation with IKM Ocean Design A/S which has been my employer for the last 7 years where I have been working as a Structural and Mechanical Engineer. IKM Ocean Design Specializes in design and engineering of subsea pipelines, subsea structures and Tie-in solutions. The company is a sub company of the IKM Group in Norway which is a major sub supplier to the oil and gas industry. During the years working for IKM two vertical Tie-in Spool systems for deep water applications projects have been proven to be of great challenge when it comes to design optimization, analysis techniques and strength verification in the project. Hence a requirement for a more standardized route and methods for these types of spool would indeed benefit future projects. A new DNV guideline for structural design criteria for rigid tie-in spools has been developed by Statoil and DNV. This guideline is in a preliminary version and has not been available for the author of this thesis but it is expected that this guideline will be released during 2015. The main objective of this thesis is to investigate standard solution and identify main challenges for engineering of subsea Tie-in Spools, and propose possible new solutions and recommendations for the commencing of such projects.

My Intention for selecting this subject was to learn more about subsea systems and computer analysing techniques such as finite element programs, and to explore solutions and methods based upon my project experience in IKM within this topic. I hope that people who read it will find it interesting and inspiring, and that the work contributes to give information and advice for future projects and development in the industry.

I would like to thank the management and administration at IKM Ocean Design AS for the opportunity to work on this thesis. Special thanks go to *MD. Peder Hoås*, *Tech. Dir. Per Nystrøm* and former *Dept. Mngr. Helge Nesse*, who made it possible for me to post graduate with this thesis in the IKM Company.
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I would also thank my supervisor *Dr. Hirpa G. Lemu* at *UIS* for given me guidance and comments on how to work with this thesis.

Finally I would like to give my *family* and my wife *Christina Andersen*, a special gratitude for being so patient and supportive for me during the working process over the years as a part time student at *UIS.*

Stavanger 12.06.2015

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Nomenclature

Latin symbols

$\bar{a}$	Intercept of the design SN-curve with the log N axis
A	Area
A_i	Internal area of pipe
A_e	External area of pipe
A_s	Cross sectional area of pipe
b	Constant polynomial
c	Corrosion allowance
c	Damping coeffcient
c_c	Critical damping
D_i	Inside diameter
D_o	Outer Diameter
D	Nominal diameter
D	Accumulated fatigue damage
D_k	Nodal Imposed displacement in k=x, y, z global direction
d	Distance
E	Young's Modulus
F	Force
F_{axl}	Axial force
F_1	ASME design factor for hoop stress
F_2	ASME design factor for longitudinal stress
F_3	ASME design factor for combined stress
F_{fric}	Frictional force
f_{osc}	Frequency forced oscillation
f_0	Eigen frequency
f_i	Natural frequency for i'th mode
f_u	Ultimate strength of material
f_ν	Shedding frequency
f	ASME Fatigue factor
F_{wall}	Axial pipe wall force due to pressure
F_{endcap}	Axial pipe endcap force due to pressure
g	Dimensional errors
$i_{o,i}$	Stress intensification factor (SIF) out of plane or in plane
I	Moment of inertia
K	Stiffness
k	Number of stress blocks
$[K]$	Stiffness matrix
K_s	Stability factor
L	Length
M	Bending moment
M_{add}	Added mass
$[M]$	Mass matrix

m	mass
m	negative inverse slope of the S-N curve
m_e	Effective mass
M_i	In plane bending moment
M_o	Out of plane bending moment
M_t	Torsional moment
N	Axial force
N	Number of cycles S-N Curves
n	Number of stress cycles
N_{eff}	Effective force in pipe
N_{true}	True force in pipe wall
P	Pressure
$P_c(t)$	Collapse pressure
P_e	External pressure
$P_{el}(t)$	Pressure at elastic capacity perfect tube
P_i	Internal pressure
P_{min}	Minimum internal pressure
P_o	Outer pressure or external pressure
$P_p(t)$	Pressure at plastic capacity
P_n	Probability during n-years
R_k	Nodal imposed rotations k= x, y, z global axis
R_e	Reynolds number
R_p	Return period
R	Mean radius
R_i	Internal radius
R_o	Outer radius
S_A	ASME stress limit flexural stress
S_b	ASME Longitudinal bending stresses
S_c	ASME Allowable stress at cold pipe
S_h	ASME hoop stress
S_h	ASME Allowable stress at hot pipe
S_L	ASME longitudinal stress
S_P	ASME Longitudinal pressure stresses
S_u	ASME Ultimate tensile strength
S_{axial}	ASME Axial stress
S_t	ASME Torsional stress
S	ASME Specified minimum yield strength
S	Sample standard deviation
T	ASME Temperature de-rating factor
t	Nominal pipe wall thickness
t_1	DNV definition for minimum pipe wall thickness
t_{min}	Minimum wall thickness
t_{Corr}	Corrosion allowance
t_{fab}	Fabrication tolerance
U	Current speed
V	Flow velocity

V_r	Reduced velocity
W_{sub}	Weight of pipe pr. meter
W_t	Pipe wall thickness
y	Constant polynomial
Y	Population random variable
Z	Distance in meter
Z_{nom}	ASME Section modulus, nominal wall thickness

Greek Symbols

α	Thermal expansion coefficient or
α_m	Allowable stress factor membrane stress
α_{fab}	fabrication factor
ε_{el}	Longitudinal strain
Δ	Displacement or difference
σ_a	Axial stress
σ_b	Bending stress
σ_i	Principal stress i=1,2,3
σ_j	2D-Coordinate stress j=x, y, z
σ_l	Longitudinal stress
σ_h	Mean hoop stress or circumferential stress
σ_{rr}	Radial through wall stress
$\sigma_{\theta\theta}$	Tangential stress or Circumferential stress
σ_y	Yield strength of material
τ_{ij}	Shear stress i=x, y, z j=x, y, z
ν	Poisson’s ratio
ν	Kinematic viscosity
θ	Temperature Difference
μ	Frictional coeffcient
γ_{HISC}	Material quality factor
ρ	Mass density
η	Usage factor
ζ_T	Total modal damping ratio
ω	Natural frequency

Abbreviations

AISC	American Institute of Steel construction
ANSI	American National Standards
APDL	ANSYS Parametric Design Language
API	American Petroleum Institute
ASD	Allowable Stress Design
ASME	American Society Of Mechanical Engineers
ASCE	American Society of Civil Engineers
BBRTS	Big Brother Remote Tie-In system
BPVC	Boiler pressure Vessel Code
BOP	Blow Out Preventer
CAD	Computer Assisted Drawing
CFD	Computational Fluid Dynamics
CFX	ANSYS computational fluid dynamics program
DNV	Det Norske Veritas
EQSLV	Equivalent Stress Level
DFF	Design Fatigue factor
DOE	Design Of Experiments
DOF	Degree Of freedom
DP	Design Pressure or Differential Pressure
FAT	Factory Acceptance Test
FEA	Finite Element Analysis
FEED	Front End Engineering and Design
FEM	Finite Element Method
FFRP	Flexible Fibre Reinforced Pipe
FMC	Food Machinery Corporation
GE	General Electric
GR	Gravity
GUI	Graphical user Interface
HCCS	Horizontal Clamp Connector System
HCS	Horizontal Clamp
HISC	Hydrogen Induced Stress Corrosion
ICCG	Incomplete Cholesky Conjugate Gradient
IPB	Integrated Production Bundle
ISO	International Organization for standardization
KC	Kollet Connector
KHK	Kouatsu-Gas Hoan Kyoukai, (The High Pressure Gas Safety Institute of Japan)
KP	Kilometer point
LRFD	Load Resistance Factoring Design
MAOP	Maximum Allowable Operating Pressure
Max	Maximum
MEG	Mono Ethylene Glycol

MF	Manifold
Min	Minimum
MIT	Massachusetts Institute of Technology
MOP	Maximum Operating Pressure
MSL	Mean Sea Level
Nom	Nominal
NEMA	National Electrical Manufacturers Association
NORSOK	Norsk Sokkels Konkurranseposisjon
NPD	Norwegian Petroleum Directorate
PC	Personal Computer
PCG	Preconditioned Conjugate Gradient Solver
PEEK	Polyetheretherketone
PLEM	Pipeline End Module
PLET	Pipeline End Termination
ROV	Remote Operated Vehicle
RP	Recommended practice
RTS	Remote Tie-In System
SCF	Stress Concentration Factor
SIF	Stress Intensification Factor
SMYS	Specified Minimum Yield Strength
SMTS	Specified Minimum Tensile Strength
TR	Technical Requirement
TFRP	Thermoplastic Reinforced Pipe
UBC	Uniform Building Code
ULS	Ultimate Limit State
UTIS	Universal Tie-in System
VIV	Vortex Induced Vibration
WB	Work Bench
WI	Water Injection
WRC	Welding Research Council
X-Tree	Wellhead “Christmas Tree”

1. INTRODUCTION

Subsea Tie-in solutions provided by most of the major actors in the subsea market provides various systems for connecting pipelines to manifolds, wells and Trunk pipe lines. These pipelines are usually called “spools” or tie in spool. This is usually a steel pipe oriented either vertically or horizontally with a connector system in each end, other types used is of a flexible types similar to what is used in risers. These spools are often designed to withstand large forces and displacements due to pressure and temperature in the pipeline during installation and operation; hence the requirement for flexibility and strength is one of the key design features. Various computer optimization techniques such as the use of FEA and CFD are utilized in order to analyse and verify strength of these spools towards numerous load combinations in order to document required design life and governing codes. Experience has shown that some of these solutions are sensitive to parameter changes such as:

- Flow and process data
- Material choice
- Metrology and fabrication tolerances
- Environmental factors.
- Size and shape
- Connector solutions

Typically main issues related to design of rigid spools can be listed as follows:

- Size
- Stresses
- Conflict between company standard and code requirements
- Lack of recommended practice
- Corrosion and (HISC) problems
- Insulation
- VIV
- Weight
- Fatigue
- Erosion
- Slugging
- Pressure loss
- Requirement for MEG inhibitors
- Sour service
- Seabed
- Size and limitation of connector systems
- Requirement for structural support equipment

In order to reduce project cost, time and complexity, (especially for deep water application and diver less tie-in system) the following topics should be studied such as:

- Efficient computer analysis and methods
- An early identification of critical values
- Alternative Tie-in solutions
- Reduction of complexity
- Reduction of vessel installation time.
- Reduction of cost by use of robust standard solutions.
- Better use and understanding of recommended design standards, company practices and codes.

1.1 Historical

Since the 1980's, when the subsea industry started moving into water depths where divers could not be used, the industry has been challenged to provide a simple cost effective method of connecting two lines without divers.
The industry has responded to this challenge providing innovative methods of doing first end and second end tie-in methods including:

- Stab & hinge-over',
- Rigid jumpers/spools
- Flexibles,
- Deflect and connect

A multitude of vertical and horizontal connectors & tools have been used. However, the use of rigid jumpers still remains the universal method of performing deep water pipeline connections, possibly due its extensive proven track record, its cost effectiveness and high reliability. However, this system still has significant drawbacks which include the requirement for metrology, topsides fabrication (which may or may not be on the critical path), installation with a multi-point lift and its limited capability to accommodate pipeline expansion and two tie-in operations. Ref. /1/
Some of the early projects during the 1980,s utilizing the deflect to connect approach was

- **East Frigg Project.** June 1988. Connection of 2 production manifolds to a central manifold by 2 bundles in 24" carrier pipes to provide buoyancy. Bundles connected by a first time diverless Deflect to connect method.
- **Troll Olje Project**. August 1995. Connection of 16" oil and gas export pipelines. First time diverless Deflect to Connect directly on pipelines by attaching weight and buoyancy.

1.2 Problem Description

A pipeline connection is normally used as a link between a pipeline, manifold, oil-well, storage tank, processing facility or other mechanical equipment used for the transportation of a fluid, gas, sand or a combination of all from one location to another. The pipeline link connection is called a spool which is an English terminology (in Norwegian it translates to "snelle", which is a device for reeling something on like a fishing reel). When we use the word spool in piping terminology it is understood as piece of pipe with necessary bends tees and flanges for connection to another system. In simple terms it is the pipe from flange to flange. The concept is relatively simple. As the pipes are heated and pressurized they expand and since the piping is restrained in some way in a piping system stresses are developed. For subsea pipelines the spools is usually an infield pipeline connection to a trunk exporting pipeline, manifold, oil wells or other subsea facility. The transport medium is:

- Produced oil
- Gas injection
- Water injection
- Multiphase flow (oil, gas and water)

Spools must have enough flexibility to withstand the expansion deflection from facilities such as:

- Pipeline and Risers connected to subsea structures or other processing unit.
- Oil-wells and manifolds
- Environmental forces

Reference is also made to the Master thesis of 2012 made by Espen Slettebø Ref. /2/ the thesis assesses key requirements related to tie-in spools by a detailed review about issues related to the design, fabrication, installation and operation of tie-in spools here the definition of the Tie-in spool is described as.

"Essentially spool pieces are short sections of pipeline that:

- Provide an interface between the pipeline and its connection point that bridges the inaccuracies associated with pipeline installation. For a tie-in spool to serve as intended, it needs to satisfy numerous different criteria. Principally it needs to make up the connection between the pipeline and the interconnecting part. For pipelines that are transporting hydrocarbons it is crucial that the connections are sealed. Containment of hydrocarbons is crucial to reduce the risk of pollution and ensuring safe transportation of hydrocarbons. Tie-in spools are measured, fabricated and installed after the pipeline has been laid. Mechanisms related to these operations, makes the tie-in spool a key piece of equipment in offshore field developments

- Allow the pipeline to expand during operation but also allow these pipeline expansion forces to be dissipated/reduced at the associated connection point. The tie-in spool also needs to be a flexible element. Pipelines expand because of temperature and pressure differences between installation and operational conditions. This expansion may be in the order of several meters. Depending on how the pipeline is constrained, expansion may cause the pipeline to buckle or by it extending in axial direction. The expansion is taken up by deflection of the tie-in spool. Simultaneously as the pipeline expands, forces are induced into the tie-in spool and the connector. Making sure that induced loads are below material and connector limitations is critical in design of tie-in spools.
  These key requirements can have a significant impact on the overall cost of a project. A too conservative design means an oversized tie-in spool. A too large tie-in spool increases the use of materials, hampers the manufacturing process and more importantly may limit the number of vessels that can install the spools resulting in a requirement for large costly heavy lift vessels or separate two vessels to transport the spools."

1.3 Scope and Limitation

This thesis major purpose is to investigate and present some of the standard solutions of the tie-in system as used by the major actors in the oil and gas industry. The thesis will utilize other studies, company experience, papers and thesis on this topic. The main objective is to analyse a vertical jumper spool by use of commercial finite element analysis software, and to study spool design such as:

- Investigate the effect of a flexible joint or seabed support in order to reduce moment and forces in a rigid spool.
- Optimize the computer analysis by parametric variation
- Comparison of computer models and software
- Study effects of statistical distribution of tolerances and deflections

The study will also include:

- Fabrication issues
- Development of design basis for analysis
- Theory
- Use of applicable standards and codes

Other topics such as:

- Conceptual ideas
- Further studies and development for Tie-in
- Limitations of Tie-in systems

The thesis will aim to propose recommendations for commencing of such projects and present the result of the case study. Engineering and analysis of subsea Tie in spools normally involves a large work scope to be investigated. In order to limit the work for this thesis, a limited number of load cases are checked, and the focus of the work presented here is mainly for vertical spool types.

1.4 Report Structure

Chapter 1 *(Introduction)*

The introduction contains the background information, to gain an illustrative understanding about the content of this thesis. The problem is stated followed by the purpose and scope of the thesis. A short thesis organization is also included (this section) to make navigation in the document simple for the reader.

Chapter 2 *(Background and Theory)*

This section contains presentations of some theory and examples as to gain an understanding of the basic principles and physical behaviour of piping system

Chapter 3 *(Tie-in Spools System)*

This section presents examples of typical subsea Tie-in systems. The section describes typical advantages and disadvantages for each system. Fabrication methods and considerations of tolerances are discussed.

Chapter 4 *(Connector and Tie-in Systems)*

This chapter describes the function of subsea connectors and the available tooling required for performing a subsea Tie-in. A general list of the most common systems used and the manufactures is given.

Chapter 5 *(Design basis)*

This chapter describes the basis for the design of the spool. The chapter describes data to be used in the design such as the use of governing codes and standards. The chapter also describes the important parameters such as materials, dimensions, loadings and limitations for the system.

Chapter 6 *(Spool optimisation and strength verification)*

This chapter describes the computer software tools used in the structural analysis of piping systems. The boundary condition and the computer model for the FEA are given and a description of the analysis method is outlined. Load cases for the spool is described and assigned to the analysis.

Chapter 7 *(Analysis Results)*
This chapter presents the analysis results from the ANSYS Design Explorer tool. Statistical distribution of the results are presented and discussed. An optimal configuration for the spool and sensitivity to imposed loading is studied
Chapter 8 (*Verification and Comparison of Results*)

This chapter investigates different computer models and compare results. The main purpose is to study if there are major differences between typical finite elements used in computer piping analysis. The chapter compares results from software typically used in the industry for piping analysis. (AutoPIPE).

Chapter 9 *(Spool Weight and Load Mitigation)*

In this chapter some ideas on how to minimize loading on the connectors for a vertical spool is investigated and results are presented. Typical subsea equipment used for mitigation of VIV and weight is presented.

Chapter 10 *(VIV Check of Spool)*

This chapter studies the effect and sensitivity of the spool to be excited by the sea current into a harmonic frequency with a spring seabed support. Modal analysis is performed by use of ANSYS. Typical recommended practice for the design check against VIV is discussed and a method for checking against fatigue is presented.

Chapter 11 *(Future Solutions)*

This chapter presents some developed and conceptual ideas for future subsea spool projects. The ideas are presented with the intention that it might have potential for cost savings

Chapter 12 *(Summary and Conclusion)*

The overall Summary, conclusion and recommendations from the work performed in this thesis are presented a recommended engineering practice based upon this thesis work is described. Suggestion for future studies on this topic is given.

2. BACKGROUND AND THEORY

2.1 Structural analysis of piping

Structural analysis of piping systems is of great importance to study as temperature, pressure and gravity forces is inducing stresses, strains and deformations in the pipe system when it is restrained. Furthermore as the piping system heats up and shuts down the piping system is exposed to changes in stresses, this causes a fatigue situation. For a piping system exposed to environmental forces such as current and waves typically for subsea pipes, VIV (Vortex Induced Vibration) can cause the pipeline to be excited into harmonic low frequency vibration. This can result in fatigue failure or unintentional high displacement ranges.

The designer must calculate the stresses allowed by a particular code. One of the significant differences between flexibility analysis and pressure design is that flexibility is related to stress range rather than a specific stress.

For subsea spool piping the most important parameters to study is the effect of:

- Pipeline expansion from pressure and temperature
- Tie-in forces
- Metrology and fabrication tolerances
- Environmental forces.
- Installation methodology

For pipelines which may vary from just a few hundred meters to several hundred kilometres it is also important to study the effects such as those listed below. These topics are thoroughly described in literature Ref. /10/.

- Pipeline lateral buckling
- Pipeline upheaval buckling
- Pipeline walking

For tie-in spools these effects are not relevant as the boundary conditions required for the phenomena is usually not present.

The pipelines are designed as to avoid buckle to be triggered at the end of a pipeline as this could potentially damage the spools. In theory the effects might be present in the spools if the effective force in a spool is of such a nature that large axial compression forces can be generated. Pipeline walking is a phenomena created by the in-balance of the effective axial force during start up and shut down and differences in the temperature gradient along the pipeline which changes the location of “virtual” anchor (restraining point) along the pipeline.

The acceptance criteria for both spools and pipelines are usually a strain and stress based criteria set forth by a piping code such as DNV, ISO or ASME.

2.2 Wall thickness design

Figure 2-1 Show the basic theory from solid mechanics. The figure shows the equilibrium balance for a pressurized pipe section exposed to internal and external pressure. The external pressure P_o plus the pipe wall tension force ($2S_Ht$) has to balance the internal force $P_{i.}$ The so called “Hoop” stress or the tangential stress is the dimension criteria for pipe wall design. The mean hoop stress is expressed as:

$$\sigma_h = \frac{p_i D_i - P_o D_o}{2t} \tag{2.1}$$

Where

D_o= Outside diameter
D_i= Inside diameter
σ_H= Mean hoop Stress
P_o=External pressure
P_i=Internal pressure
t= pipe wall thickness

Other variants of the hoop stress can be found such as Barlow (2.2) and the DNV-OS-F101 (1996) formula (2.3):

$$\sigma_h = \frac{p_i D}{2t} \tag{2.2}$$

$$\sigma_h = \frac{(p_i - p_o)(D_o - t_1)}{2t_1} \tag{2.3}$$

Here t_1 = Thickness -corrosion -prefabrication tolerances at operation

The ASME B31.8 Hoop stress formula for D/t>30:

$$S_h = \frac{(p_i - p_e)D}{2t} \tag{2.4}$$

t= nominal wall thickness

And for D/t< 30

$$S_h = \frac{(p_i - p_e)(D - t)}{2t} \tag{2.5}$$

Thin wall vessel (R/t >10) Roark’s formulas for stress and strain

$$\sigma_h = \frac{p_i R_i}{t} \tag{2.6}$$

Thick wall vessel Lamés equations

$$\sigma_{rr} = \frac{A}{r^2} + 2C \tag{2.7}$$

$$\sigma_{\theta\theta} = \frac{-A}{r^2} + 2C \tag{2.8}$$

Subjected to an internal pressure P gives the following solutions for the constants A and C

$\sigma_{rr} = -P$ at inner radius r=R_i and
$\sigma_{rr} = 0$ at outer radius r=R_o

$$A = \frac{PR_i^2}{(R_0^2 - R_i^2)}$$

$$C = \frac{PR_i^2 R_o^2}{(R_0^2 - R_i^2)}$$

So the expression becomes at given radius r:

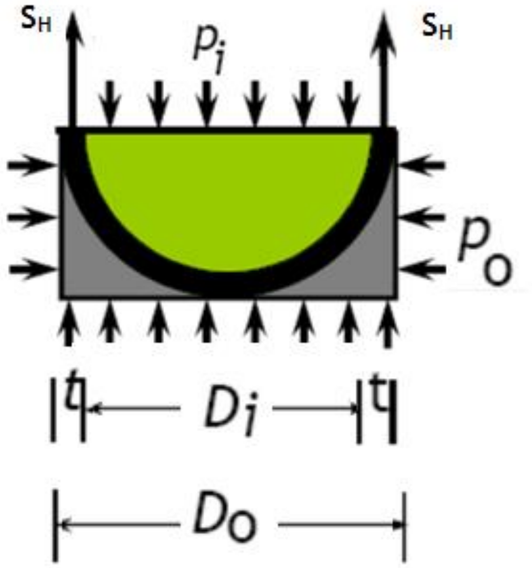
Tensile hoop stress

$$\sigma_{\theta\theta} = \frac{PR_i^2}{\left(R_0^2 - R_i^2\right)}(1 + \frac{R_0^2}{r^2}) \tag{2.9}$$

Compressive radial stress:

$$\sigma_{rr} = \frac{PR_i^2}{\left(R_0^2 - R_i^2\right)}(1 - \frac{R_0^2}{r^2}) \tag{2.10}$$

A comparison of the formulas is calculated in Appendix A3.4



$$P_0\,D_0 + 2s_H\,t = p_i D_i$$

Figure 2-1 Force balance in a pressurized pipe section pr. unit length

Most of design codes require that the hoop stress is less than the yield stress with a safety factor SF or a fraction of the yield stress named F_1 often found in American piping codes. The factor is dependent upon several factors such as:

- The location of the system
- Pressure
- Fluid type and service
- Fabrication methods.

This factor is then multiplied with the yield strength to get the allowable stress (ASD).

- $F_1\sigma_y$
- $\sigma_h \leq F_1\sigma_y$

The design factor or usage factor was taken as 0.72 for pipelines and 0.60 or less for risers. The figures date back 70 years to time when standards of pipe manufactures, welding and construction were of a less quality than what we have today Ref. /4/ .Today the limit in codes for utilising the material capacity in pipelines is much higher such as the plastic strain limit found in the DNV–OS-F101 code Ref. /7/.(It is here worth mention that the code does not give any limit state criteria for strain in pipe bends which is typical for spools).

The DNV code requires a strict control regime for the manufacturing and installation process.
The formula used in the code for wall thickness design in the LRFD design method is considering first term of the equations in Sec 5 D200 of the code (the other terms is for test pressure and mill test pressure)

$$t = \frac{D}{1 + \frac{2}{\gamma_{sc}\gamma_m(P_{li} - P_e)} \cdot \frac{2}{\sqrt{3}} \cdot \min(f_y, \frac{f_u}{1.15})} \tag{2.11}$$

Where:

D= Outside diameter

γ_{SC}= 1.138 (safety class normal)

f_u=Ultimate strength

P_{li}=incidental pressure

γ_m =1.15

f_y=Yield strength

t= pipe wall thickness

P_e=External pressure

The minimum wall thickness is the subtraction of the following components:

$t_{min}=t-t_{corr}-t_{fab}$

t_{corr} =Corrosion allowance

t_{fab}= Manufacturing tolerance on wall thickness

2.3 Collapse of pipe wall under external pressure

In deep water, collapse under external pressure drives the wall thickness design. The theory of collapse mechanism is a complex interaction between elastic circumferential bending and plastic bending and initial out-of roundness. The theory for pipe wall or axisymmetric shell under external loading can be found in various literatures. A complete thesis on this topic can be found in the doctoral thesis of 2009 by Rita G. Toscano ref /8/.

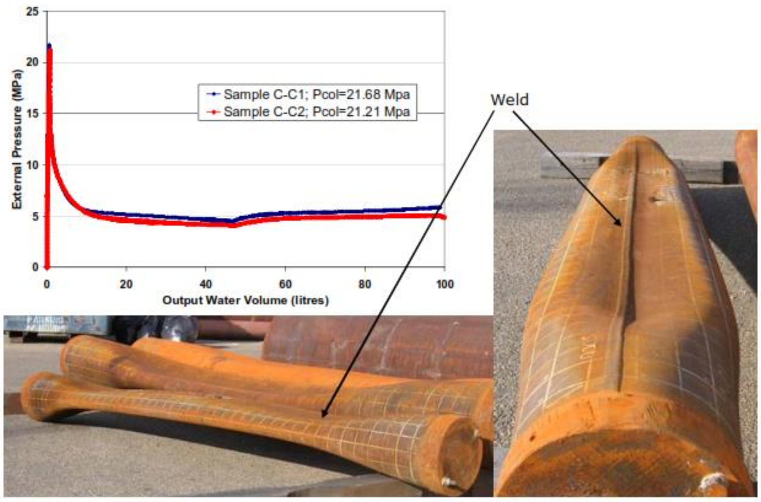


Figure 2-2 12" Pipe collapse curves and post collapse, Ref. /8/

Analytical solutions and investigation on this topic is also presented in the thesis of King, Ken Hiroshi Ref. /9/. In general FEA techniques are investigated with experimental test and analytical expression. For

practical engineering the most used formula for checking against collapse is the third degree polynominal expression presented in DNV-OS-F101 ref /7/.
External pressure shall meet the following criterion:

$$P_e - P_{min} \leq \frac{P(t)}{1.1\gamma_{sc}\gamma_m} \tag{2.12}$$

Characteristic resistance:

$$(P_c(t) - P_{el}(t)) \cdot ((P_c(t)^2 - P_b(t)^2) = P_c(t) \cdot P_{el}(t) \cdot P_p(t) \cdot f_o \cdot \frac{D}{t} \quad (2.13))$$

Where:

$$P_{el}(t) = \frac{2 \cdot E \cdot \left(\frac{t}{D}\right)^3}{1 - \nu^2} \tag{2.14}$$

$$P_p(t) = f_y \cdot \alpha_{fab} \cdot \frac{2 \cdot t}{D} \tag{2.15}$$

$$f_o = \frac{D_{max-}D_{min}}{D} \; (>0.5\%) \tag{2.16}$$

Where:

$P_{el}(t)$= Pressure at elastic capacity perfect tube
$P_p(t)$= Pressure at plastic capacity
$P_c(t)$=collapse pressure

E =Youngs Modulus
α_{fab}=fabrication factor
ν= poisson's ratio

The solution to the collapse pressure $P_c(t)$ is given in Sec 13 D700 of the code.

$$P_c(t) = y - \frac{1}{3}\mathrm{b} \tag{2.17}$$

Where the solution to the constants y and b is given in the code and can be calculated.
For local buckling collapse check with external pressure and bending moment strains the pipe must be checked in accordance with Sec 5. D600 in Ref. /7/
Other formulas for collapse due to external pressure can be found in the offshore code API RP111 section 4.3.2.1 Ref./32/

2.4 Longitudinal stress

Longitudinal stress is statically indeterminate and depends primarily from two effects which is the temperature and the poisson effect.
Longitudinal stress depends on how the pipeline moves longitudinally.

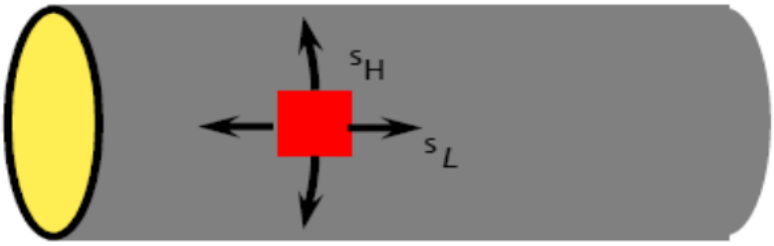


Figure 2-3 Stress component in a pipe

Note $S_H=\sigma_h$ and $S_L=\sigma_L$
The longitudinal strain is given by stress strain relation for a linear isotropic material:

$$\varepsilon_l = \frac{\sigma_l}{E} - \frac{\nu\sigma_h}{E} + \alpha\theta \qquad (2.18)$$

The first term is the longitudinal strain and the second term is the hoop strain and the last term is the thermal strain.
Where

ε_{el}= Longitudinal strain
σ_h= Hoop stress
σ_l= longitudinal stress
E= Youngs Modulus
α =Thermal expansion coefficient
θ=Temperature Difference
ν= poisson's ratio

Considering a complete axial constraint (ε_{el}= 0) and the hoop stress for a thin wall pipe expressed as:

$$\sigma_h = \frac{PR}{t} \qquad (2.19)$$

Where:
R= mean radius
t=wall thickness
Inserted into equation (2.18) and solved for the longitudinal stress gives the following expression:

$$\sigma_l = \frac{\nu PR}{t} - E\alpha\theta \qquad (2.20)$$

As shown the longitudinal stress has two components the first is related to pressure and the second is related to temperature. The pressure component is positive (tensile) and temperature component is negative (Compressive). The nature of resultant longitudinal stress depends on the relative magnitudes of pressure and temperature increase.

2.5 Combined stress and von Mises Equivalent Stress

Most of the design codes in allowable stress design (ASD) use the von Mises yield criterion for checking against yielding. From solid mechanics we know that yielding in isotropic material occurs when the tri-axial principal stress reaches the yield limit of the material expressed as:

$$\sigma_{eq} = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1} = \frac{f_y}{\gamma_m} \tag{2.21}$$

Where f_y is the yield limit of the material and γ_m is a material factor larger than 1.0.
For a plane stress state where σ_3=0 and the principal stress expressed in terms of coordinates stresses x, y the expression reduces to:

$$\sigma_{eq} = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau_{xy}^2} \tag{2.22}$$

The x-direction is the hoop stress and the y-direction is the longitudinal stress component and the last term is the shear stress. Maximum allowable combined stress varies with the codes and what type of operational phase. But normally 90% of SMYS (Specified Minimum Yield strength) is a common factor for allowable usage.

2.6 Pipeline expansion

Due to the operating temperature and operating pressure the pipeline will expand at its two ends.
The three main reasons contributing to the end force and expansion leading to the lateral upheaval buckling and walking are:

1. Temperature
2. Pressure
3. Poisson contraction associated with pressure effects

The thermal strain and pressure difference between installation and operation in an unrestrained pipe causes expansion as given by equation (2.18).The end cap force occurs at any curvature along the pipeline and contributes to the longitudinal stress see Figure 2-4 and Figure 2-5 the force is expressed as:

$$F_{end} = \mathrm{p_i}A_i - \mathrm{p_e}A_e \tag{2.23}$$

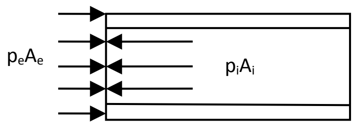


Figure 2-4 End Cap Force

Where:

p_i= internal pressure
A_i= Internal area of pipe
p_e=External pressure
A_e= External area

The effect of the end cap pressure is shown in the figure below where the pipeline expands and deflects the pipe spool.

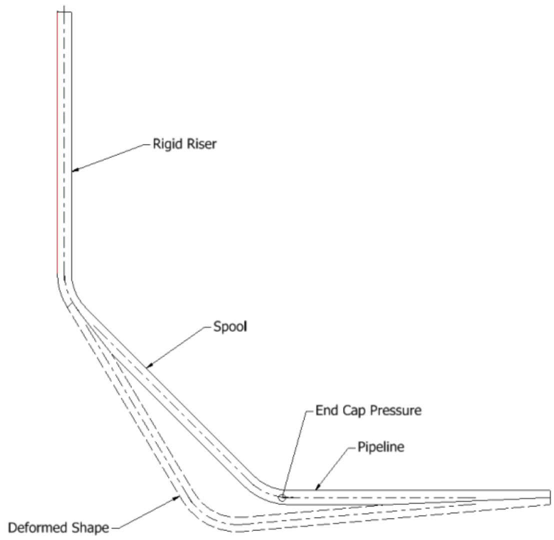


Figure 2-5 Pipeline end expansion

At partially restrained area the longitudinal stress is dependent upon the soil friction acting on the pipe. The longitudinal stress then becomes:

$$\sigma_l A_s = F_{end} - F_{fric} \tag{2.24}$$

And the frictional force is given as:

$$F_{fric} = \mu W_{sub} Z \tag{2.25}$$

Where:

F_{fric}= Frictional force
$\sigma_l A_s$= Pipe wall force
μ =Soil frictional coefficient
W_{sub}=Weight of pipe pr. meter
Z =Distance in meter

Figure 2-6 illustrates the frictional force between the pipe and the soil.

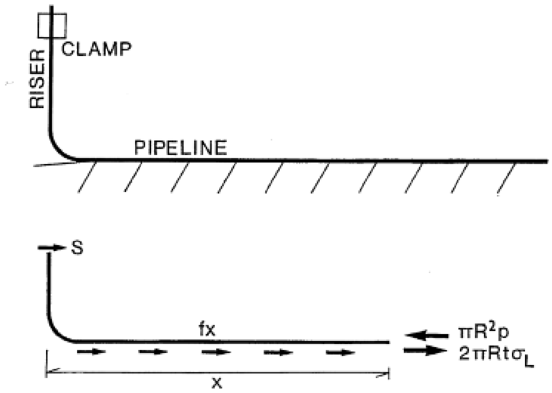


Figure 2-6 Frictional force from soil acting on pipeline

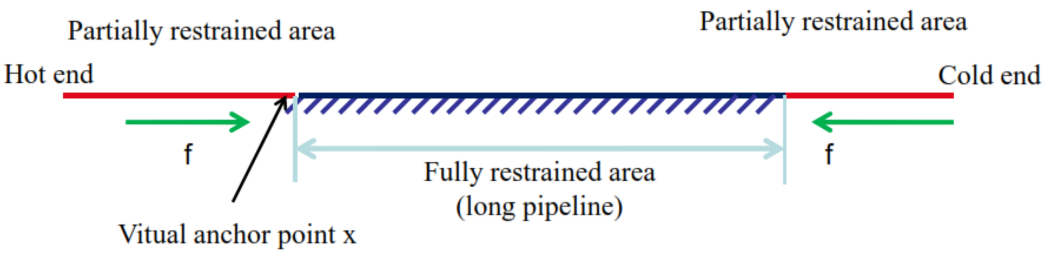


Figure 2-7 Anchor point of fully restrained pipeline

The distance required in order to make the pipeline fully restrained can now be calculated based upon (2.23),(2.24)and (2.25)and the following static equilibrium equation:

$$F_{wall} + F_{fric} - F_{endcap} = 0 \tag{2.26}$$

The distance to fully restrained pipe is called virtual anchor length and becomes:

$$Z = \frac{(F_{endcap} - F_{wall})}{F_{fric}} \tag{2.27}$$

Figure 2-7 shows a pipeline with a hot and cold ends, the distance between the hot and cold end is fully restrained.

The effective axial force is a very important concept in pipeline design. If the expansion is restricted an effective axial force will arise given by the following equation Ref. /10/and /11/:

$$N_{eff} = N_{true} - p_i A_i + p_e A_e \tag{2.28}$$

For a deformed pipe the governing differential equation for deformation is given as:

$$EI\frac{\partial^4 v}{\partial x^4} - (N - p_i A_i + p_e A_e)\frac{\partial^2 v}{\partial x^2} = 0 \tag{2.29}$$

The effect of the lateral pressure can be seen as a lateral force as shown in the Figure 2-8.

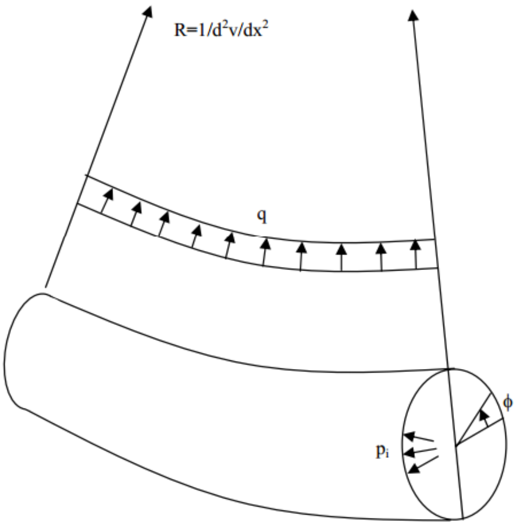


Figure 2-8 Deformed Pipe section with internal pressure and bending radius, Ref. /11/

This effective axial force is an equivalent force of the cross section including both pipe wall stress (true wall force) and the internal and external pressure; this governs the global structural behaviour buckling etc. The true force is the actual force as measured by a strain gauge or as by integrating the stress over the cross-section area. Other forces like bending moments and shear force are omitted for clarity as they will not enter the calculation of the effective axial force and the effect of the pressure

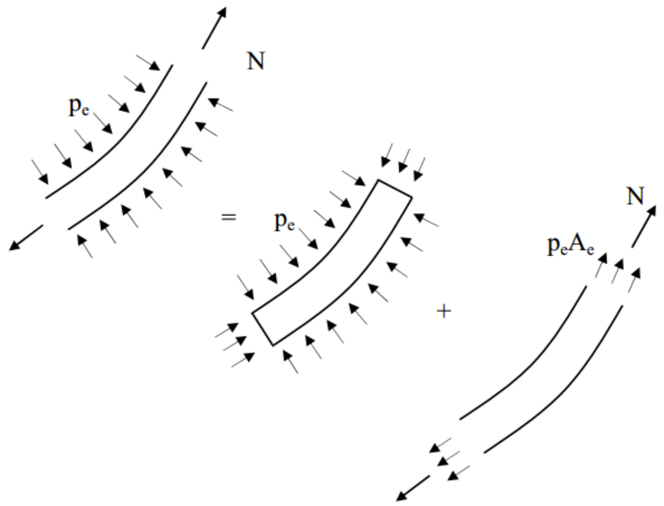


Figure 2-9 Equivalent physical system for external pressure, Ref. /11/

As seen from Figure 2-9 the section with an axial force, N, and the external pressure, p_e, (left figure) can be replaced by a section where the external pressure acts over a closed surface and gives the resulting force equal to the weight of the displaced water, the buoyancy of the pipe section (middle figure), and an axial force equal to N + p_eA_e. Considering the effect of the external pressure in the way as shown does not change the physics or add any forces to the pipe section. "However, it significantly simplifies the calculation. The alternative would be to integrate the pressure over the double curved pipe surface. Note also that the varying pressure due to varying water depth over the pipe surface needs to be accounted for in order to get the effect of the displaced water, the buoyancy".

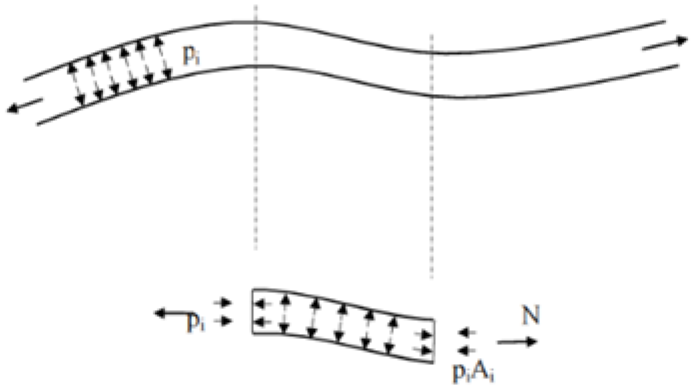


Figure 2-10 Equivalent physical system -internal pressure Ref. /11/

"A similar consideration, as for the external pressure, may be done for the internal pressure. As seen in Figure 2-10 considering a section of a pipeline with internal pressure, the external forces acting on this section is the axial force, N, and the "end cap" force, p_iA_i. Again other sectional forces like bending moment and shear forces are omitted for clarity. As the pressure acts in all direction in every point in the liquid, the internal pressure will always act on a closed surface. Further, the pressure at the cut away section ends will act as an external axial load in compression. From these considerations of the external and internal pressures acting on a pipeline section it becomes clear that the effect of these may be accounted for by the so-called effective axial force equation" Ref./11/:

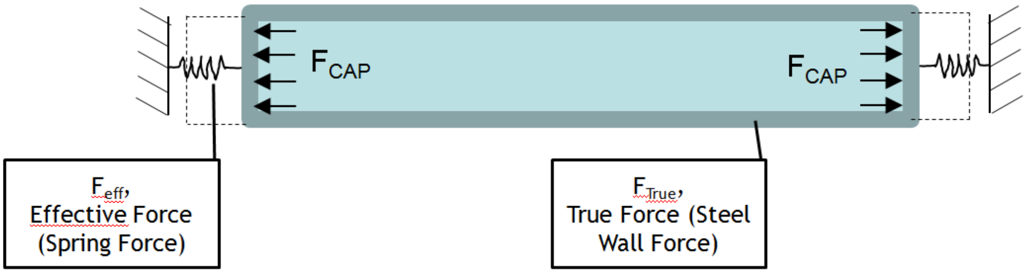


Figure 2-11 Simplified system Effective force closed cylinder

Consider the simplified system of Figure 2-11 with a pipe of closed ends subjected to internal pressure and springs at each end then the relationship between the effective axial force and the true force becomes:

$F_{eff}=F_{true}-F_{endcap}$

The effective force is governing the structural response of the system and is of great importance as to checking if the force is of a magnitude to trigger an upheaval or lateral buckle.

Figure 2-12 shows that the effective force varies with the pipeline length. One reason for this is that the temperature is not constant along the pipeline.

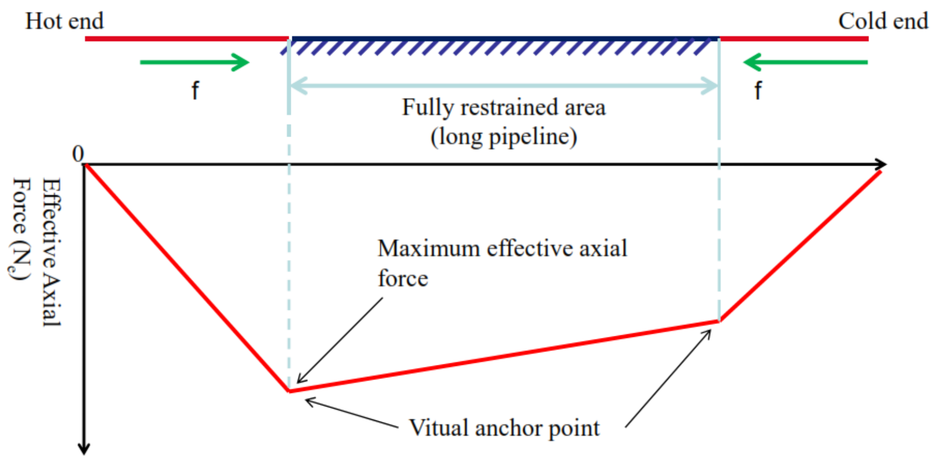


Figure 2-12 Effective force for a long pipeline

2.7 Flexibility of piping system

The flexibility of a piping system can be demonstrated with a simple calculation known as the "guided cantilever method" Ref. Pre-study Chapters 1.4". The principle is shown with beam theory.

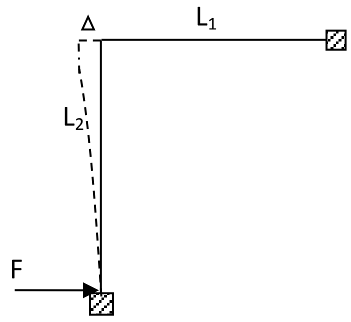


Figure 2-13 simple restrained pipe flexibility design

As the pipe heat up and is pressurized the expansion of the pipe induces stresses and forces due to its restrained boundary condition.

Figure 2-13 shows a simple 2D plane frame. The frame is a statically undetermined problem as there are 9 degrees of freedom (3 at each beam joint). The expansion is applied in the axial direction of L1 representing the heat up an imposed displacement of the pipe. The corner at the deformation is not allowed to be rotated (rigid corner).We wish to calculate the reaction length of leg L2 and the moment in order to check the stresses in the beam.

The solution for this problem can be solved by using the "direct method" with formulas for deformation for simple beams to establish a stiffness matrix for the beam element Ref. /13/. The beam element with 6 D.O.F for L_2 is shown in the following figure.

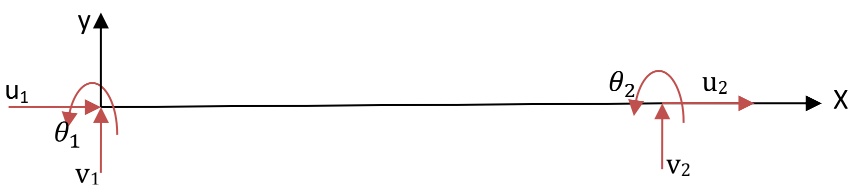


Figure 2-14 beam element with 6 D.O.F

Equation (2.30) shows that reaction force matrix $[\boldsymbol{R}]$ for the beam element is defined as the stiffness matrix $[\boldsymbol{K}]$ multiplied by the displacement vector$[\boldsymbol{D}]$

$$[\boldsymbol{R}] = [\boldsymbol{K}] \cdot [\boldsymbol{D}] = \frac{E}{L^2}\begin{bmatrix} AL & 0 & 0 & -AL & 0 & 0 \\ 0 & 12I/L & 6I & 0 & -12I/L & 6I \\ 0 & 6I & 4I & 0 & -6I & 2IL \\ -AL & 0 & 0 & AL & 0 & 0 \\ 0 & -12I/L & 6I & 0 & -12I/L & -6I \\ 0 & 6I & 2IL & 0 & -6I & 4IL \end{bmatrix}\begin{bmatrix} u_1 \\ v_1 \\ \theta_1 \\ u_2 \\ v_2 \\ \theta_2 \end{bmatrix} \tag{2.30}$$

The column displacement vector $[\boldsymbol{D}]$ represents the degree of freedom at the nodes (u_1, v_1, θ_1 u_2, v_2, θ_2) and by introducing the boundary condition for the beam (u_1=0, v_1,=0 θ_1=0 u_2=0, and θ_2=0) and v_2 =Δ and by eliminating the row and a column which are zero. The following stiffness of the beam then becomes:

$$K_{beam} = \frac{-12EI}{L^3} \tag{2.31}$$

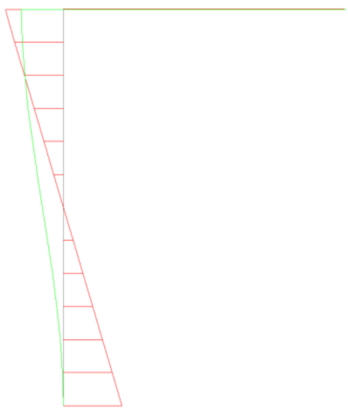


Figure 2-15 Bending and deflection diagram for frame

Figure 2-15 shows the resulting bending moment and the deflection diagram for the frame. Consider an example of a high temperature on L_1 (250°C or an expansion rate of 4in/100ft) and an imposed axial displacement of 1.5 inches which results in a total expansion of Δ=2.3in=58.4mm. Introducing the pipe data (Ref. Pre-study to master thesis):

<u>Pipe data:</u>

Pipe outside diameter	D_o=4.5in (114.3mm)
Wall thickness	t = 0.237in (6mm)
Displacement	Δ=2.3in (58.4mm)
Young Modulus	E=27.9x10^6psi (1.924 x10^5MPa)
Stress limit	S_A=15000psi (103.4 Mpa)

$$F = K_{beam} \cdot \Delta \tag{2.32}$$

Bending stress:

$$\sigma = \frac{My}{I} \tag{2.33}$$

Where y= $D_o/2$ and M=FL/2

Inserting (2.31), (2.32) into (2.33) and rearranging for L gives the following equation for required length L_2 to be within the given stress limit and give adequate flexibility:

$$L = \sqrt{\frac{3E \cdot \Delta \cdot D}{\sigma}} \tag{2.34}$$

Inserting values as given above and $S_A=\sigma$, equation (2.34) then gives the following value:

$$L = \sqrt{\frac{3 \cdot 192400 \cdot 58.4 \cdot 114.3}{103.4}} = 6105mm\ (20.03ft)$$

And the reaction force then becomes:

$$F = \frac{2 \cdot \sigma \cdot I}{\mathrm{R} \cdot L} = \frac{2 \cdot 103.4 \cdot 2.995 \cdot 10^6}{54.11 \cdot 6105} = 1875N\ (422lbf)$$

As seen from the calculations the flexibility of the piping is dependent upon geometry and the allowable stress in order to be within safe limits between the heat up and cool down cycles.

2.8 V.I.V in Pipelines

VIV or Vortex Induced Vibrations are common problems in offshore structures and often occur when structural components are in a free span and submitted to a flow from wind or seawater. When the flow separates from a large section of the structure surface due to its geometry (*bluff structure*), vortices are generated and shed alternately from one side to another side of the structure. This alternating shedding of vortices induces forces and can cause the structure to start oscillate. When the frequency of the vortex shedding is close to the natural frequency of the structure, large amplitude resonant oscillations may occur. This is a known problem for free spanning pipelines and risers. The VIV effect can generate large amplitude forces and this can lead to fatigue failure of the structural component. Figure 2-16 shows an illustration of how the pipe is displaced in the vertical and horizontal direction due to the vortex shedding generated from the sea current.

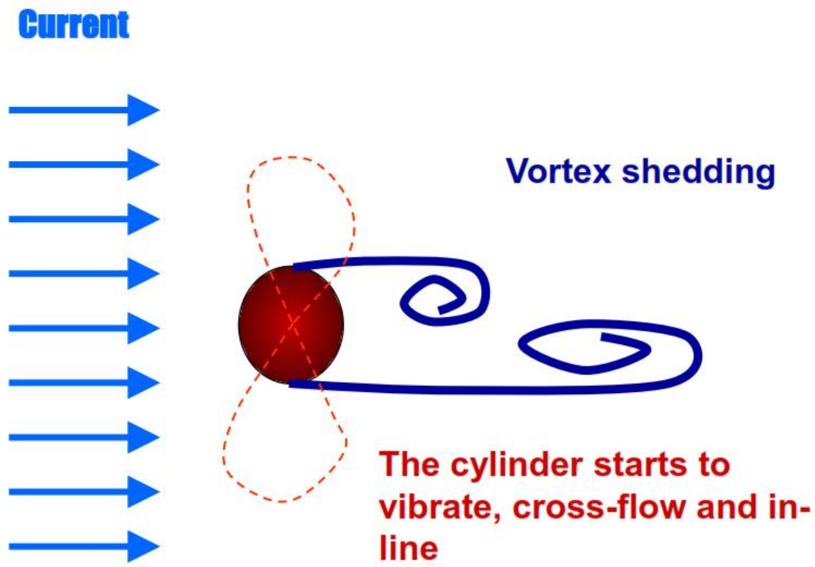


Figure 2-16 Vortex Induced Vibrations

The pipe can start to oscillate in the vertical direction (crossflow) and in the horizontal direction (In-line) in an *8* number pattern. Figure 2-18, shows that the oscillation is in phase or it can have a *D* shape pattern as seen in Figure 2-19 where the oscillation is out of phase. The behaviour and the pattern of the vortices are very much dependent upon the flow regime. The Reynolds number R_e is often used as a measure between the laminar and turbulent flow, see Figure 2-17 and is given by equation (2.35). The eigenfrequency of the pipe is also influenced by the change of added mass of the displaced water and this will also influence the vortex shedding frequency. Three other important key parameters, each linked to the frequency is the Strouhal number *St*, the reduced velocity $U_{R,}$ and the non-dimensional frequency $\hat{f}$. Ref. equation (2.36) to (2.38).

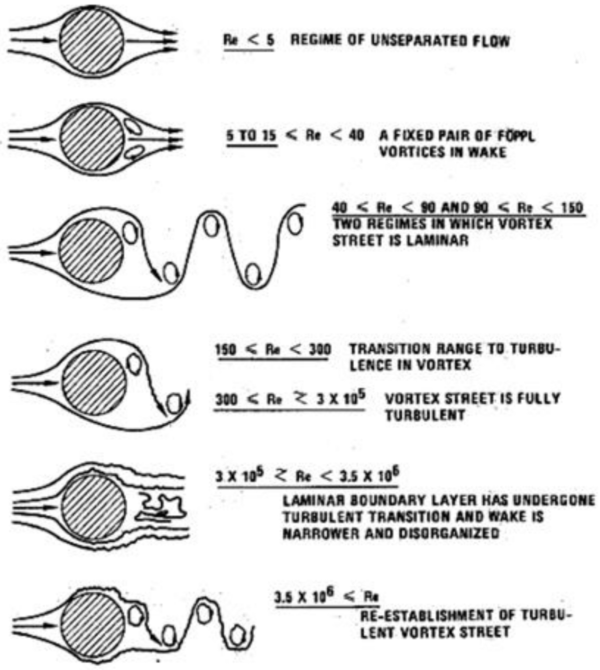


Figure 2-17 Regimes of fluid flow across smooth circular cylinder Ref. /23/

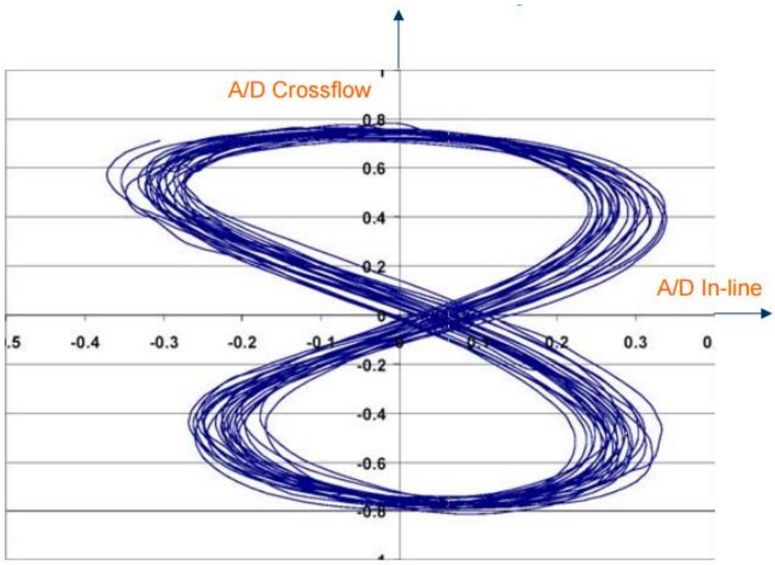


Figure 2-18 In-line and cross-flow oscillations-in phase [SINTEF]

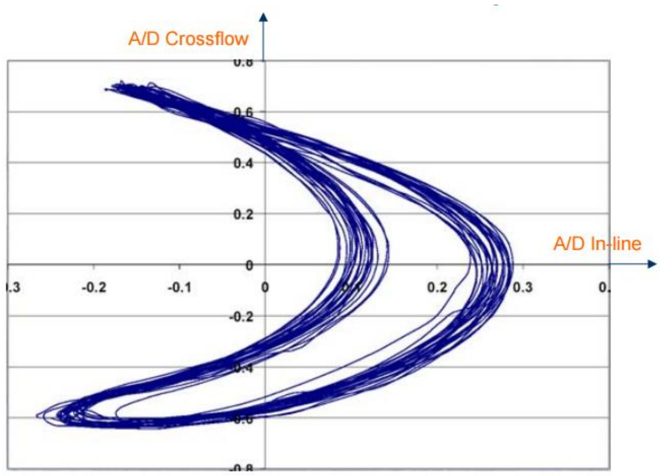


Figure 2-19 In-line and cross flow oscillations-out of phase [SINTEF]

From fluid dynamics theory the Reynolds number is defined as:

$$R_e = \frac{UD}{\nu} \tag{2.35}$$

And the Strouhal number:

$$S_t = \frac{f_v D}{U} \tag{2.36}$$

The Strouhal number is the frequency from a fixed cylinder.

Reduced velocity:

$$U_R = \frac{U}{D f_0} \tag{2.37}$$

The reduced velocity is from still water free oscillation tests

Non-dimensional frequency:

$$\hat{f} = \frac{f_{osc} D}{U} \tag{2.38}$$

Which is a Oscillation (response) frequency for forced oscillation tests

The parameters in the above formulas are:

U= Current speed	f_{osc} =Frequency forced oscillation
D= Diameter of pipe	f_0 =Eigenfrequency
ν= Kinematic viscosity	f_ν = Shedding frequency

These formulas are often found in design codes and recommended practices for checking pipes against VIV effects which can lead to fatigue failure.

3. TIE-IN SPOOLS SYSTEMS

3.1 Objective and functionality

A Tie-in spool is a prefabricated piece of pipe whose main objective is to act as a coupling between a pipe, manifold, wellhead, PLET, PLEM or other subsea structure. The main function is to transport hydrocarbons or other processed medium between the underwater facilities in an offshore oilfield area. The spools is designed with large flexibility, in order to withstand forces induced such as:

- Expansion from pipelines, wellheads, manifolds processing facilities etc.
- Installation loads.
- Fabrication tolerances and metrology
- Environmental loads

The spools are equipped with a mechanical connector system in each end either diver less or diver assisted system in order to mechanically seal the spool end.
A typical spool assembly is shown in Figure 3-1 and Figure 3-2 .

Figure 3-1 Vertical spool jumper lift (Gulf Island Fabrication for BP)

Figure 3-2 Horizontal spool lift (Stord Leirvik-Thaijournal.wordpress.com)

Figure 3-1 shows a vertical spool configuration mainly used in waters with depth greater than 300m. These spools seldom protection from trawling, and are typically used in connection with X-trees, manifolds etc. Observe the difference for the installation spreader bar structure for the horizontal spool lift versus the vertical lift as shown in Figure 3-2 . Typical subsea applications of vertical and horizontal Tie back spools are shown in Figure 3-3 to Figure 3-5.

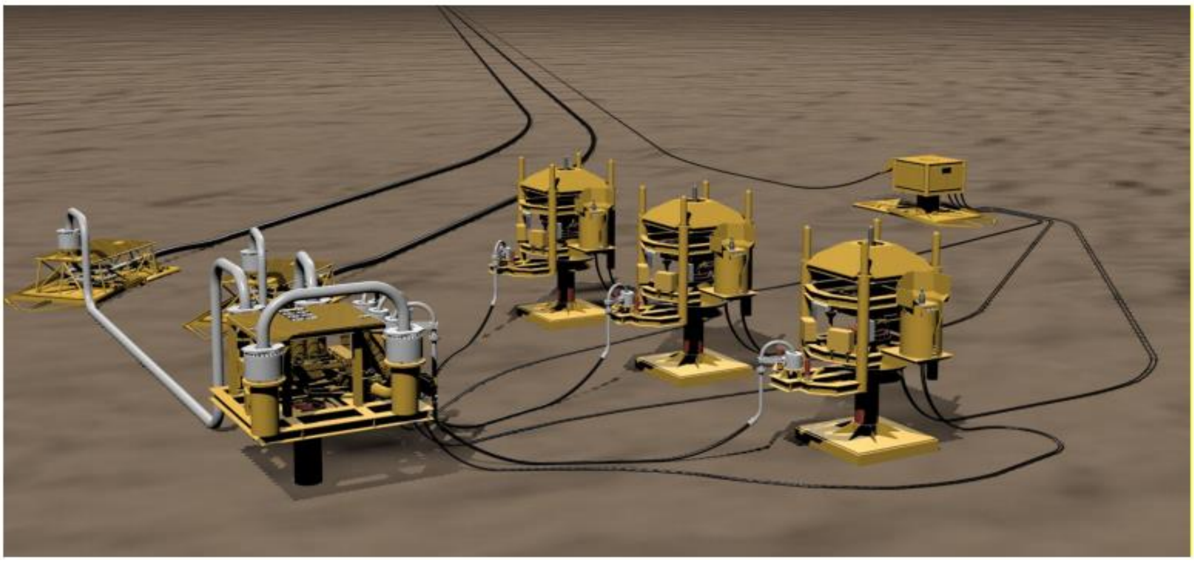


Figure 3-3 Typical Gulf Of Mexico Subsea Tie-Back Ref. /5/

Figure 3-4 Typical Tie-Back in the Norwegian continental shelf Ref. /5/

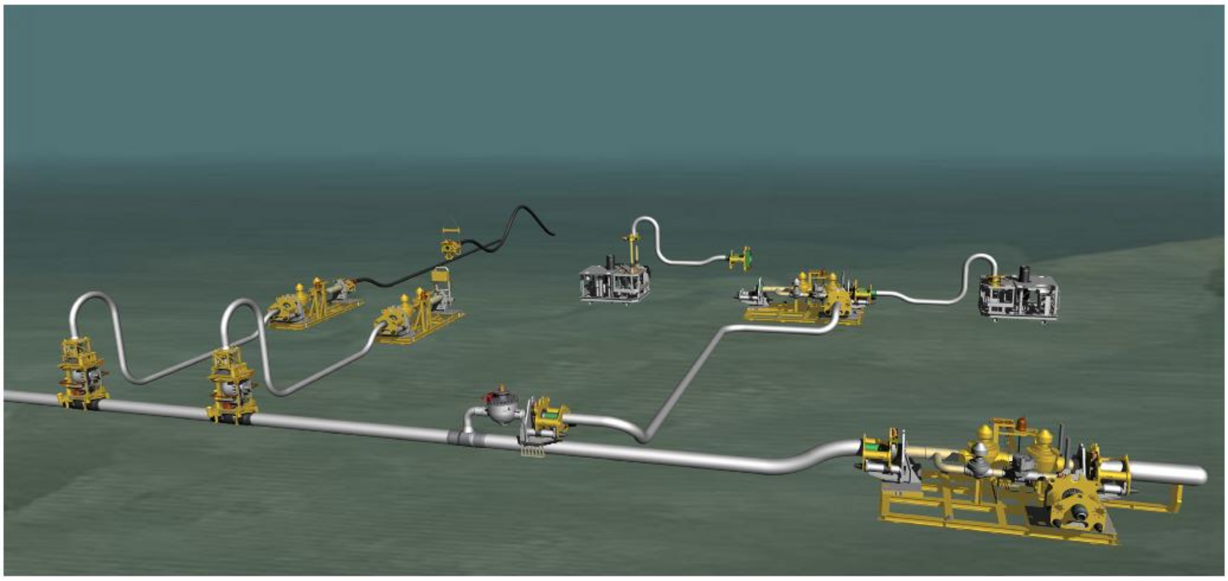
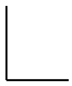
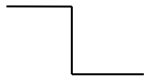
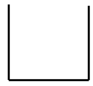


Figure 3-5 Spool's connected to subsea structures (PLEM, X-tree and Tee'- FMC Technologies)

3.2 **Configurations and geometrical shapes of spools**

Table 3-1 gives a rough classification for basic shape for rigid spools normally used in subsea spool design. A more detailed comparison with experience data is given in Table A6-1

Table 3-1 Spool Shapes

Name	**L-Shape**	**Z-shape**	**U-shape**	**N-shape**	**M-shape**
Shape					
Expansion range	≤0.5m	≤1m	≤1m	≤1.0m	≤2m
Area of usage	Horizontal spool resting on seabed perpendicular connection between hubs	Horizontal spool resting on seabed angular or perpendicular connection between hubs	Horizontal spool resting seabed parallel connection between hubs	Deep water Vertical spool arrangement with free span used between up-facing hubs.	Deep water vertical spool arrangement with free span used between up-facing hubs.
+/-	-Low flexibility +low weight -High installation cost -Protection cover usually required	+Medium flexibility +low weight -High installation cost -Protection cover usually required	+Medium flexibility +low weight -High installation cost -Protection cover usually required	+ Medium flexibility -Heavy weight -Buoyancy required -VIV Sensitive + Low installation cost -Snagging potential	+ High flexibility -Heavy weight -buoyancy required -VIV Sensitive + Low installation cost -Snagging potential

Horizontal spools are considered for applications within diving depth where snagging loads can be a threat (trawl or anchor etc.). It can also be used in region of significant hydrodynamic effects. Diver less horizontal Tie-in system can be used for the entire water depth range. Vertical systems are also available and are mainly used in deep water developments but have limitations on a max span range usually not more than 30 to 40meters.

3.3 **Tie-in System Determination**

When selecting and designing a Tie in system there are many considerations to be taken. Experience has shown that the following parameters are of great importance.

- Area location and water depths (diver assisted or diverless)
- Environmental loads
- Trawl frequency (protection requirements)
- Installation vessel (requirement for deck area and crane capacity)
- Connector system (Track record)
- Well stream and process data (medium, slugging, pressure and temperature profiles)
- Requirement for intervention and monitoring
- Weight limitations
- Hub to Hub distance
- Fabrication contractor
- Metrology contractor (Survey and measuring system)
- Design Code requirements
- Method of metrology
- Seabed condition
- Size of equipment and tools
- Tolerances
- Vessel limitations
- Pigging requirements
- No. of Tie-ins to be performed
- Pipeline expansion
- Material choice
- Region of the world for the installation
- Design life

When selecting a system some decision gates or risk description of each system can assist in the early planning of a project. JP Kenny presented a paper on this in 2008 at the OPT conference Ref. /6/ and presented risk evaluations according to criteria rated with colour code as shown in the Table 3-2.

Table 3-2 project risk classification

Risk Description
High-may limit the application of this system in some cases
Medium-risk needs to be assessed on project basis
Low-proven reliable services

IKM Ocean Design AS project experience is implemented into this comparison. The text is marked with a *cursive* text and the comparison is shown in Table A6-1 . The evaluation shows that *vertical spools* are classified as *high risk* towards connector load capacity, increased complexity due to free span, can be sensitive to snagging, possible high risk for seal damage and it can be difficult to perform pigging operations.

For *horizontal spools* the limitation is as for vertical spools for the connector capacity towards bending moments and forces. However the spools are considered to have an overall lower risk compared with vertical spools. It is worth mention here that horizontals spools normally require longer offshore installation time and can give higher project costs.

3.4 Spool Fabrication

The fabrication of spools is normally based upon the metrology survey report which describes the location and required dimension between the two connecting hubs. The distance between the PLET and pipeline is measured and then the pipeline engineer designs a spool that will connect the two hubs.

Considerations relating to fabricating of spools are:

- Location of fabricator and yard size requirement
- Fabricator qualifications (ISO 9000 qualification, DNV OS-F101 approved, NORSOK Qualified, etc.)
- Material purchasing
- Production and standard requirement for testing.
- Lead time for special items
- Welding sequence
- Number of bends
- Coating methods
- Fabrication tolerances
- Size and weight
- Number of bends
- Requirement for fabrication stands
- Requirement for test stands and test equipment
- Requirement for lifting aids
- Logistics regarding shipping of spools
- Interface control

The above mentioned points are of importance to map and study in an early phase usually when fabrication specifications and procedures are developed for the project a comparative overview is given in Table 3-3.

Table 3-3 fabrication design considerations

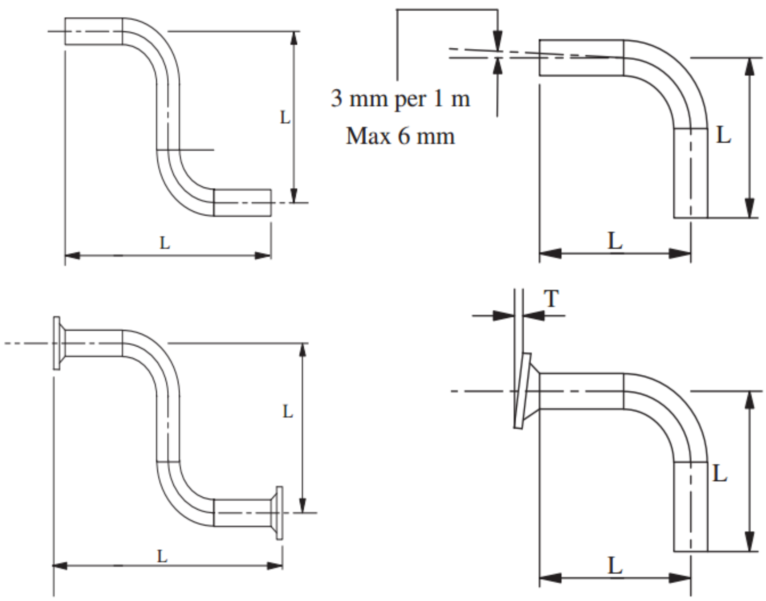
Issue	Horizontal Tie-in Spool	Vertical Tie-in spool
Size and weight	Large footprint, limits amount of spools to be transported to field.	Small footprint, scaffolding requirement needed.
Geometry	General fewer bend on horizontal spool but complexity varies	Generally more bends are related to vertical spool complexity varies
Fabrication and shipping stands	Usually less weight for support steel as stands do not need to support inboard test hub.	Requirement for large weight of supporting steel as well as tilting function of hub

3.5 Piping Fabrication Tolerances

There is a much stricter control requirement when it comes to pipe and riser fabrication versus ordinary steel constructions usually less than 0.1 x wall thickness or maximum 3mm Ref. /14/. The high demand for strict tolerances and dimensional requirement is usually set forth in the piping codes and is one of the keys to allow high utilisation of piping materials used in subsea pipe laying and subsea installation. Welds in pipelines are normally made with symmetrical weld groove with welding from the outside of the pipe. The pipes can be filled with shielding gas or backing rings can be used before welding the pipes. The main issues related to tolerances can be listed as:

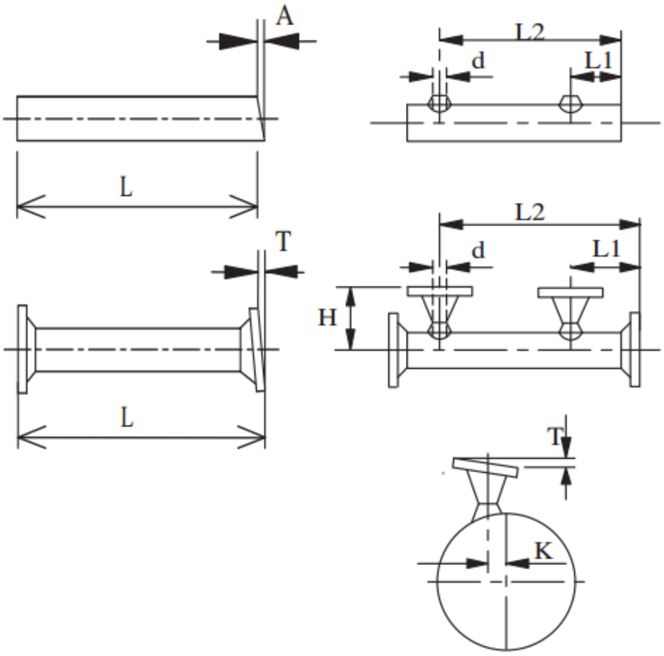
- Wall thickness tolerance
- Out of roundness
- Weld offset
- Ovality
- Concentricity
- Angular tolerances
- Length tolerances
- Bend thinning from induction bending

Figure 3-6 and Figure 3-7 from the NORSOK Standard Ref./27/ shows tolerances for pipe spool fabrication typically found in offshore topside process applications. Stricter tolerances when required are usually specified in drawings and procedures.



L [m]	Δ L	T
	Tolerances [mm]	
≤ 6	± 5	± 1,5
> 6	± 10	± 1,5

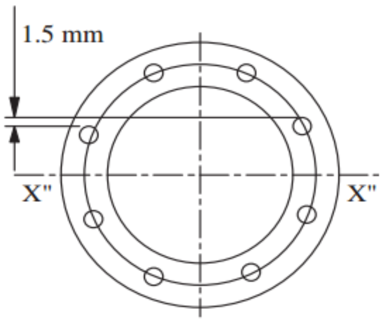
Figure 3-6 Tolerances for prefabricated piping assemblies, Ref. /27/



L [m]	L	A	T
	Tolerances [mm]		
≤ 6	± 3	± 1,5	± 1,5
> 6	± 5	± 1,5	± 1,5

d = nominal diameter

	2 in ≤ d ≤ 10 in	12 in ≤ d ≤ 20 in	d >20 in
	Tolerances [mm]		
L1	± 3	± 5	± 5
L2	± 3	± 5	± 5
H	± 3	± 3	± 3
T	± 1,5	± 1,5	± 1,5
K	± 2	± 3	± 3



NOTE 1 Before reworking any spools contact engineering department in order to check complete isometric.
NOTE 2 Cut to fit requirement to be stated on fabrication isometrics (typically 100 mm).
NOTE 3 Bolt holes on flanges shall straddle the horizontal or vertical lines or plant north/south centre lines when orientation is not given on drawings.

Figure 3-7 Tolerances for prefabricated piping assemblies, Ref. /27/

3.6 Probabilistic Assessment of Fabrication Tolerances

In this chapter we take a look at the engineering practice of using the fabrication tolerances in spool design. It is common in projects to include linear and angular tolerances of the spool in order to asses if the lock in of the connectors at the hub will generate large stresses in the spool. The question which arises is how much of the tolerances should be incorporated into the design and how much is taken care of by the governing piping codes. The answer to this is not straight forward for an optimised spool design. For normal topside or manifold piping design, the fabrication tolerances are usually incorporated into the piping design formulas. Such as the NORSOK standard Ref. /28/ and ASME B31.3 code Ref. /17/. Below is given some discussion from the book *guide to ASME B31.3* Ref. /26/ where the question regarding alignment of pipe flanges is discussed:

> ASME B31.3 provides some good practice guidelines with respect to flange boltup in para. 335. This includes requiring repair or replacement of flanges with damaged gasket seating surfaces, uniformly compressing the gasket during flange boltup, and using only one gasket between seating surfaces.
>
> Paragraph 335.2.3 requires that the bolts extend completely through their threads. However, it provides that the bolt is considered to be acceptably engaged if it is short of being completely through the nut by one thread or less. Thus, if the Owner wishes to have the nuts completely engaged, as many do, this would have to be specified in the engineering design.
>
> Perhaps one of the most frequently violated provisions of the Code is the flange alignment tolerance in para. 335.1.1(c). This requires that, before bolting up, flange faces shall be aligned to the design plane within 1 mm in 200 mm (1/16 in./ft) measured across any diameter, and that bolt holes shall be aligned within 3 mm (1/8 in.) of maximum offset. The first requirement relates to cocking of one flange relative to the other and the second relates to offset or torsional misalignment. This means that each flange can be misaligned 1 mm in 200 mm relative to the design plane. Thus, the flanges could be misaligned relative to each other by as much as double the amount. Furthermore, the design plane is not required to be the same for each flange (e.g., in a system where there is intentional misalignment to achieve cold spring). However, this would have to be the intention of the engineering design.
>
> This requirement became an issue on a project where the gaps between flanges for small-bore [e.g., DN 50 (NPS 2)] pipe flanges were being measured with feeler gages to check the misalignment. Interpretation 15-07 resulted, with the following question and reply:
>
> > "Question: In accordance with ASME B31.3c-1995 Addenda, para. 335.1.1(c), prior to bolting up a flanged joint, may the flange faces be out of alignment from the design plane by more that 1/16 in./ft (0.5%), provided the misalignment is considered in the design of the flanged assembly and attached piping in accordance with para. 300(c)(3)?"
> >
> > "Reply: Yes."
>
> Thus, some greater misalignment can be tolerated if it is provided for in the engineering design. It is quite reasonable to expect that greater misalignment than permitted by para. 335.1.1(c) can be accepted in small-bore piping, particularly if it is not connected to load-sensitive equipment. On the other hand, the Code alignment provisions are generally not tight enough for larger piping connected to load-sensitive equipment. An appropriate test of whether the alignment is acceptable is to check the machinery alignment with and without the piping bolted to it.

Based upon the above answer from the ASME Committee, larger misalignment is allowed as long as it is considered in the design. This would then have to be considered in the analysis. An interesting paper regarding fabrication tolerances is found in Ref./29/ here the following problem description is given:

"Structural reliability is normally assessed by considering its converse, failure probability. Analysts often qualify this by referring to "notional" failure probability. This is partly because the target values that designers aim for are generally so small that calculations are sensitive to assumptions about the forms of the tails of the frequency distributions used in the mathematical models of loading and resistance variables, and these tails are usually poorly defined.

"Much of the complexity of reliability analysis methods arises from the fact that the contributions of uncertainties in all the design parameters are combined within complex engineering design equations. However, the basic principles can be used quite directly to address practical problems, such as assessing the magnitude of the locked-in stresses due to the interaction between fabrication tolerances and assembly methods. In this case, interest is focused towards the most likely outcome rather than on rare events, and the mathematical concept is therefore more robust."

The paper discusses the frequency distribution of dimensional errors such as the normal distribution the Central Limit theorem and the rectangular distribution, it concludes with that the outcome is normally distributed even if the input is of a rectangular distribution hence the total effect of the tolerances has uncertain bounds which can only be described probabilistically.

It further gives some guide on how to set the limits for the tolerances or dimensional errors by using the RMS or root mean square of the errors the following as the number of the variable X_i becomes large the frequency approaches the normal distribution. And if the un-factored design condition should be taken to be a characteristic value with 5% probability of exceedance this is obtained by taking a range between +/- 1.65 standard deviation from the mean position. Thus the characteristic effect of dimensional errors is given by the following equation.

$$1.65s[Y] \approx \frac{1.65}{\sqrt{3}}\sqrt{(\sum_{i=1}^{n}(\Delta g)^2} \approx \sqrt{(\sum_{i=1}^{n}(\Delta g)^2} \tag{3.1}$$

Imperfections in element dimensions can be introduced into the structural analysis models by means of member extensions or end rotations. These can be linearly superimposed on other structural effects, so the effects of dimensional errors can be analysed independently of permanent or transient loadings.

The paper presents an analysis of a roof where over 400 different locations with the effect of tolerance was analyzed. The results where compared to the structural building code and the ULS factor of 1.4. The result shows that the sagging moments was increased by 8% in the ULS case due to fabrication tolerances and considerably increase in the lateral moments by 100 %.

The paper concludes with the following:

"Computers allow us to compute load effects for structures that have much greater complexity and indeterminacy than those that were designed using traditional methods. There is a danger that engineers may lose sight of the important load paths when they refine their calculations to reduce apparent overdesign. In so doing, ductility and tolerance towards imperfect erection procedures may be reduced, and it may become increasingly unwise to trust that redistribution of local overloads will safely take care of fabrication tolerances.

No matter what the design specifications might state, the products of different fabricators will have different patterns of variation, and a realistic prediction of the overall effects of fabrication tolerances will require knowledge of real processes."

3.7 Summary

A shown in this chapter there are many parameters regarding spool and pipe design which must be considered. A Pre-Study of the planned project is advisable to commence in order to determine the best spool design. The Pre-study or FEED should focus on the following key topics:

- Subsea routing and location of spool
- Spool type
- Installation time and constraints
- Use conservative data, previous experience is important
- Define optimum spool lengths (minimum and maximum)
- Identify critical values early in the project small changes can give rise to high cost later in project

For detail design :

- Establish design basis for analysis and design
- Perform piping analysis (FEM)
- Evaluate the effect of environmental loading, process medium and flow data
- Methodology for fabrication and installation
- Operational aspects
- Installation analysis

For the piping analysis it is advisable to incorporate a statistical tool for assessing the fabrication tolerances in order to establish a reasonable safety level in-line with the code requirements. One question is on how the project should handle the standard deviation based upon the critically of the system i.e. should the level be in the range 1.65 - 3 times the standard deviation. How will the probability of exceedance be quantified (5% or 10%)? Most of the analyzed spools today are based upon “worst case” scenario which is a very rare event but should perhaps be analyzed with the most probably expected tolerances and loading. Different techniques for random probabilistic and parameter correlation effects are available and can be included in simulation software (Monte Carlo simulation is one example)

4. CONNECTOR AND TIE-IN SYSTEMS

This chapter is dedicated to an introduction of the different subsea connector designs which is used throughout the subsea industry today. The requirement for developing of remote connector systems for pipelines and spools has been a growing business, as the oil industry has moved into deeper waters.

4.1 Connectors

The principle of connecting a pressurized pipe end to another pipe end can be best demonstrated by the classical bolted flange to flange connection with a seal located between the flanges see Figure 4-1.

Figure 4-1 Bolted Flanged Connection (VECTOR SPO Compact Flange)

The connection between the pipe joints has to sustain in addition to the internal pressure also the external forces see Figure 4-2 such as:

- Bending moments
- Torsion
- Axial forces

The forces are generated in the piping system and the joint must be able to provide a leak free connection with a given design safety margin often dictated by standards service requirements and codes. Misalignments in angular and linear directions due to inaccuracy can occur during assembly and operation, it is important that the connector can handle the misalignments in order to maintain a tight seal.

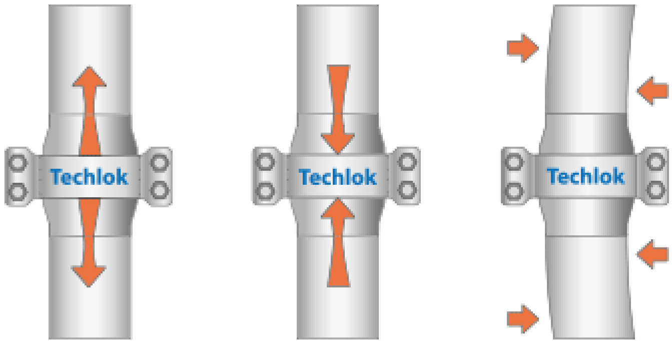


Figure 4-2 Clamped Connection subjected to external forces (Techlok by VECTOR)

Figure 4-3 Clamp Connector (Techlok by VECTOR)

The clamp connector see Figure 4-3 and Figure 4-4, uses the same principle as a bolted flanged connection. The gasket is placed between two flanges and an enclave is placed around it which is the tightened up by bolts the angled surface creates a compressional force against the seal surfaces. The internal pressure then energizes the seal lips

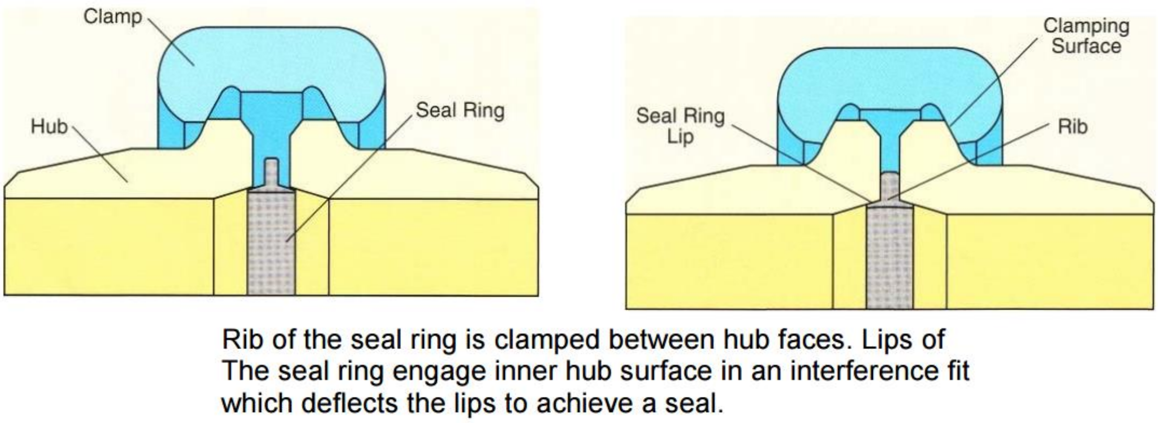


Figure 4-4 Clamp Connector (Grayloc)

The subsea connectors used for Tie-in spool applications needs to go through an extensive qualification programme in order to achieve correct certificates. Connectors need to be tested for all loading types that it might be subjected for typical capacity charts are developed for the specific connections. In a typical subsea connection a pipe piece called “Hub” is made to connect the ends together see Figure 4-5 the “Hubs” are divided into female and male hubs, the male connects the female and are machined to tight tolerances. A seal is placed between the hubs and the connection is tightened up by a mechanical external drive screw that energizes the connection. The hubs are welded to the pipe ends of the manifold and a transition piece called “pup-piece” is usually made as the transition between the hub and piping. The pup piece allows for final length adjustments.

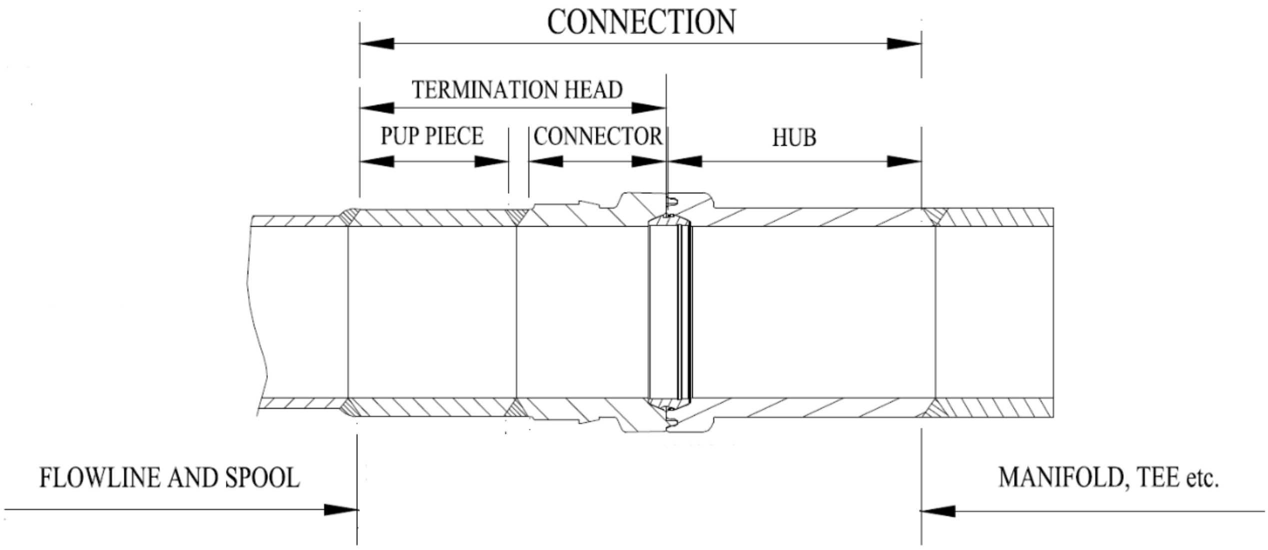


Figure 4-5 Typical Hub Connection (FMC Design)

The Hubs on the manifold subsea structure is typical connected to a stiff steel block called "Tombstone" see Figure 4-6, the tombstone's main purpose is to transfer most of the incoming spool forces into the subsea structure such that the process piping on the subsea structure is not overstressed by Tie-in loads.

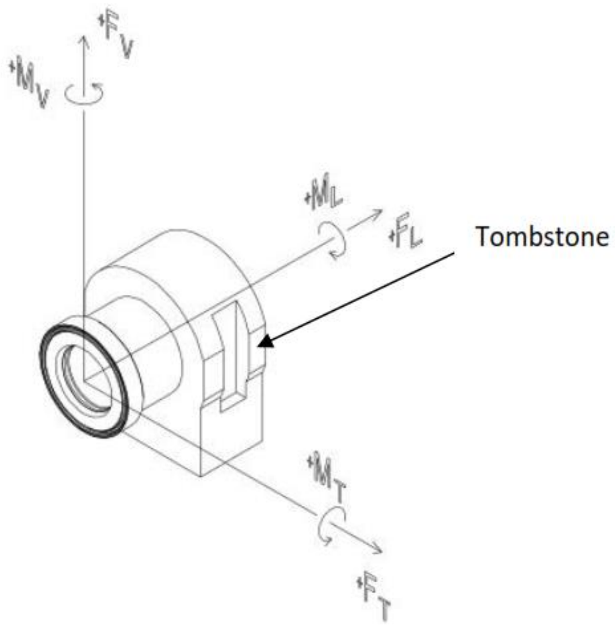


Figure 4-6 Hub located in Tombstone subjected to Spool Forces

The subsea connector is typically provided with an ROV operated external drive screw that energises the clamp. Here there are many suppliers (Vector, FMC, and Aker) see Figure 4-7 and Figure 4-8. The limiting factor of these clamps is often the misalignment capacity and the bending moment capacity. The Pipe size often dictates what type of connector to use in the design. An exploded view of the different main part in the optima connector is shown in Figure 4-9.

Figure 4-7 ROV Operated Subsea Connector (Optima VECTOR)

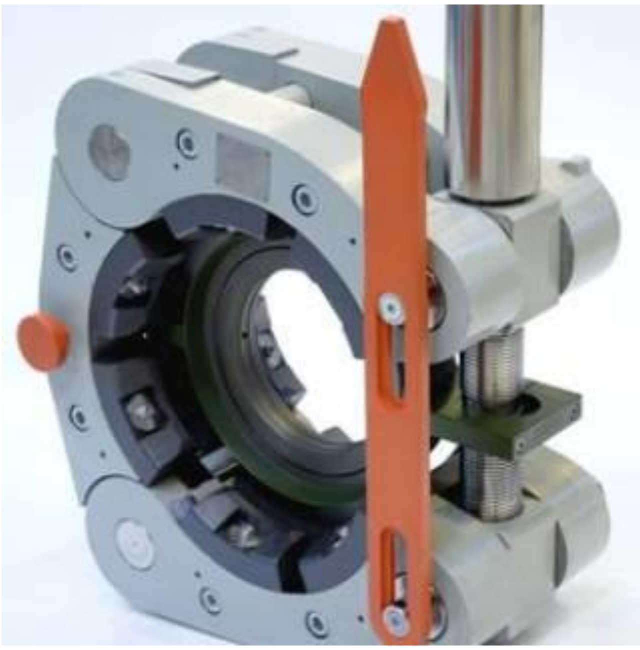


Figure 4-8 ROV Operated Pipe clamp Connector (AKER)

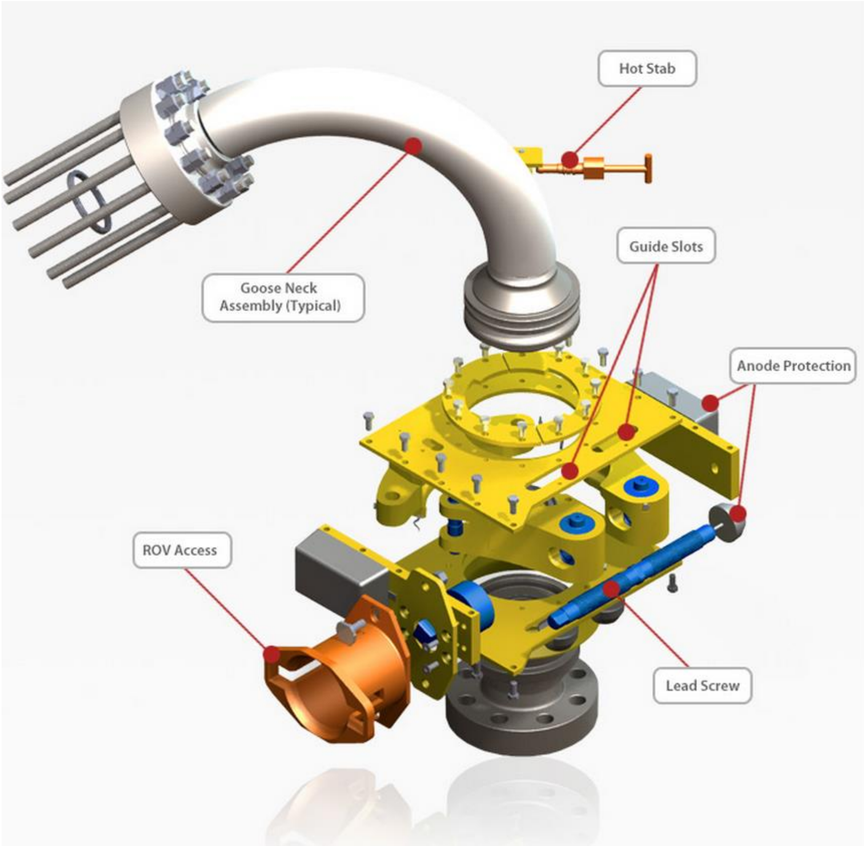


Figure 4-9 Optima connector exploded view (Vector)

The collet connector design is often found in vertical spool connections. The collet connector consists of a body and a hub. On the hub individual collets are mounted in a circular pattern. Figure 4-10 shows the FMC KC connector, outside the collets a cam ring slides axially along the collets length to either lock or unlock the connector. The seal is made by compression of a metal gasket between the body and the hub. The collet connector has the ability to align hubs that are misaligned

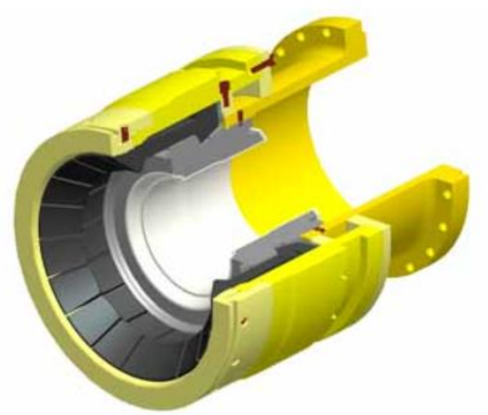


Figure 4-10 KC 4.2 Connector (FMC)

Figure 4-11 Collet Connector KC 4.2 high pressure and multibore (FMC)

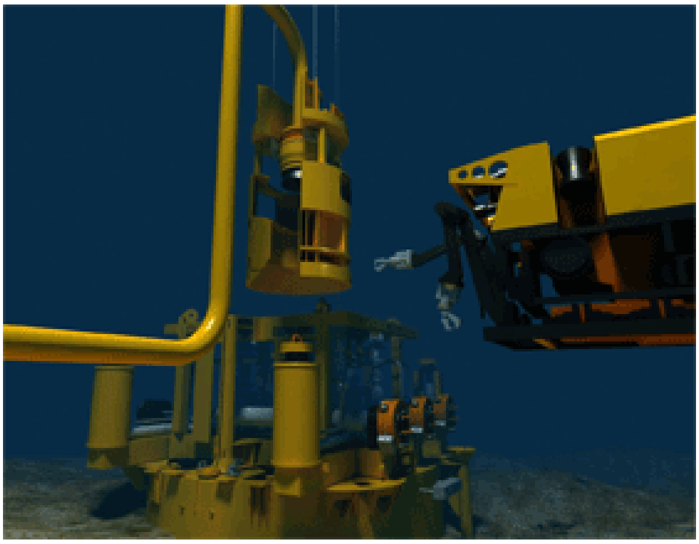


Figure 4-12 Vertical connection collet connector (FMC)

4.2 Tie-in Systems

This chapter gives a brief description of the common types of Tie-in systems used in the subsea market of today. For horizontal tie-in connections, the pipeline spool has to be connected to the subsea structure by use of a connection system. Various techniques and systems are developed by different subsea companies over the years see table below:

Table 4-1 Subsea Tie-In System's Manufacturing Companies

FMC Technologies	Vetco (GE Oil & gas)	Aker Solutions	Nemo
• Rovcon MK.I • UTIS. • Ucon-H.	• Icarus. • HCCS.	• RTS. • BBRTS. • HCS.	• Thor.

A comparison for each system based upon advantages and disadvantages is given in Table 4-2
The principle of each system is more or less the same. The pipe or spool is connected to the structure by means of hydraulic stroking/ winching or a combination of both. The system anchor's itself to the structure and pulls/pushes the pipe into position and then activates the mechanical connector clamp to make up the connection the old Icarus system uses a winch system which dock onto the porch see Figure 4-13and Figure 4-15.

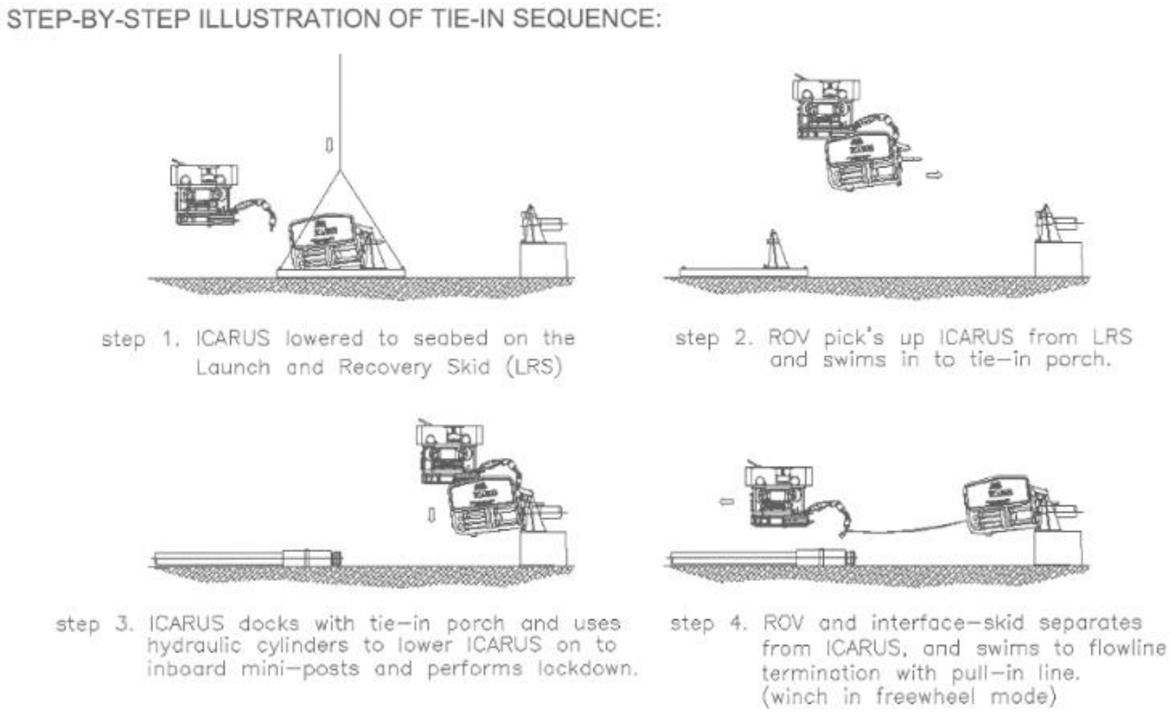


Figure 4-13 Icarus Tie-in System step 1 to 4 (GE-Oil & Gas-Vetco)

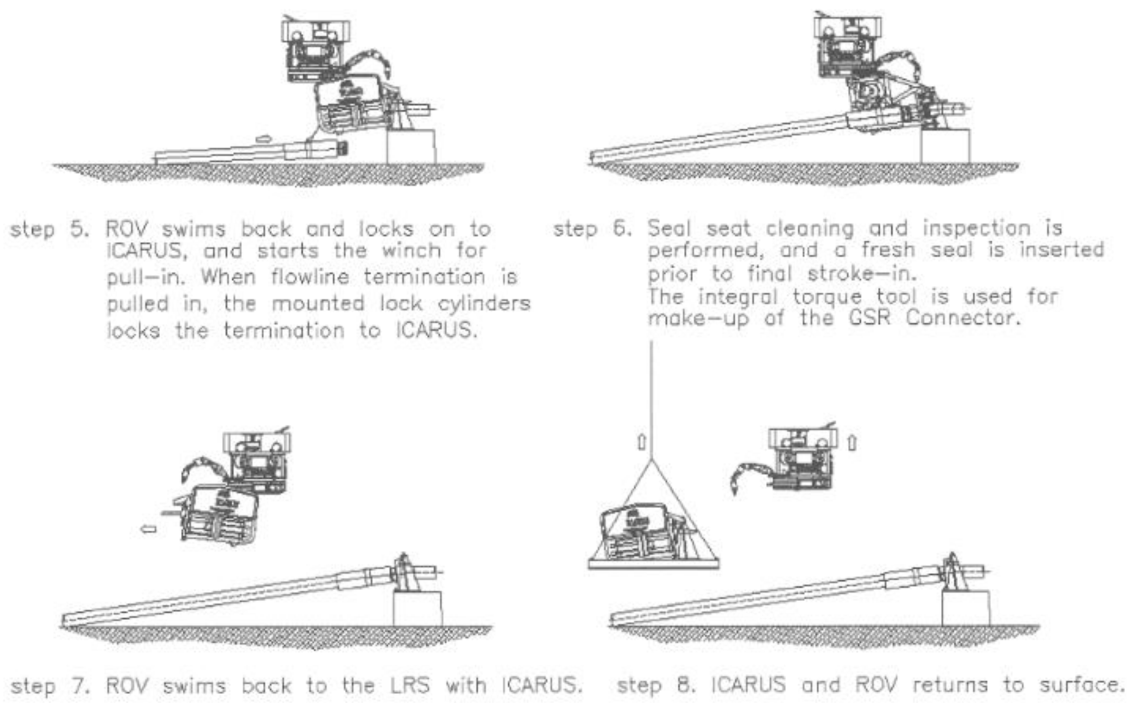


Figure 4-14 Icarus Tie-in System step 5 to 8 (GE-Oil & gas-Vetco)

Table 4-2 Comparison Tie-in systems

Tie-in System	Advantages	Disadvantages
HCS	• Can perform all Tie-ins • Compact Tie-in Interfaces • Stroke-in tool very well suited for spool Tie-In. • Well suited for tie-in of modules/integrated pig launchers etc. Inside structures	• Stroking tool design may give higher risk of compression in flexibles • Limited track record
BBRTS	• Long track record • Can perform all Tie-ins • Powerful	• Heavy and large interfaces • Tie-in Tool heavy • Installation vessel has to be extremely close to platform due to installation method
RTS	• Long track record • Lightweight tie-in interfaces • Tool neutral in water	• Limited force/moment capacity,

Tie-in System	Advantages	Disadvantages
ROVCON MK.II	• Long track record • Lightweight tie-in interfaces • Can perform all infield tie-ins • Tool neutral in water	• Large overall tool size requires a high clearance, possible conflicts with existing infra structures. • Requires a straight pipe section due to high lifting height of termination head. • High seabed clearance can give challenges for long spool frespan.
UCON-H	• Can perform all tie-ins (infield and GEP) • Powerful • Stroke-in tool very well suited for spool tie-ins	• Stroking tool design may give higher risk of compression in flexibles • Limited track record
HCCS	• Can perform all Tie-ins • Powerful • Stroke-in tool very well suited for spool Tie-In • Well suited for tie-in of modules/integrated pig launchers etc. Inside structures	• Stroking tool design may give higher risk of compression in flexibles
ICARUS	• Long track record • Lightweight tie-in interfaces • Tool neutral in water • Can perform all infield tie-in	• Large overall tool size requires a high clearance, possible conflicts with existing infra structures. • Requires a straight pipe section due to high lifting height of termination head. • High seabed clearance can give challenges for long spool free span.

The most common used Tie-in Systems for new oil field developments used today is:

- HCS (Horizontal Connection System)-Aker Solutions
- HCCS (horizontal Clamp Connection System)-Vetco
- Ucon-H (Universal connection-Horizontal Tie-in)
- Thor –Nemo (Acquired by FMC Technologies)

Figure 4-15 shows the Thor tie-in System which is one of these newly developed systems used in subsea fields.

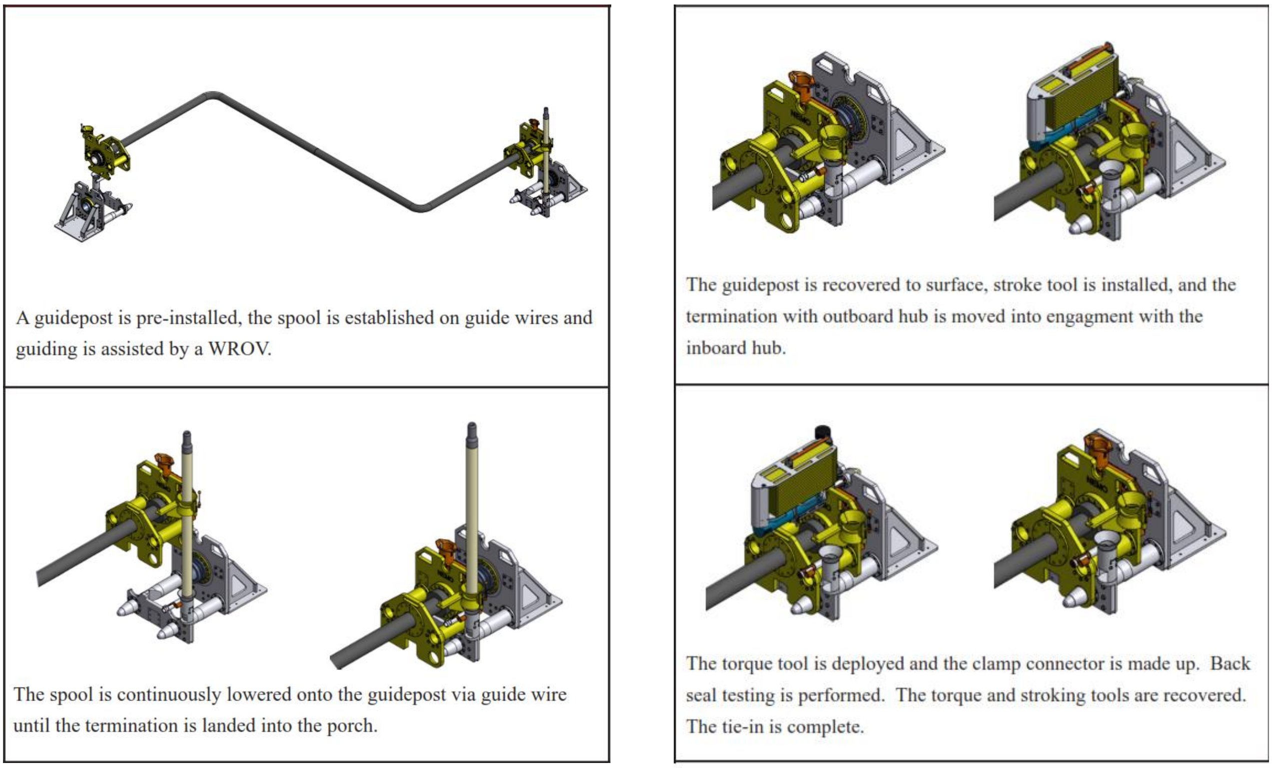


Figure 4-15 Installation sequence for Thor Tie-in System (FMC-NEMO)

5. DESIGN BASIS

The purpose of the design basis is to outline the general design premises for the piping design. The design basis identifies relevant standard codes and regulations to be followed. The design basis outlines the different design criteria for the piping analysis and spool design. It defines the basic load cases and load combination for the systems. The design basis in this thesis is based upon an earlier deep water subsea project.

5.1 Applicable codes and regulations

In the design basis it is common to list the governing standards and rules to be used in the pipe design. The use of standards and codes shall ensure that the requirements from governmental rules and laws is fulfilled and to give guidelines for a safe design., see Table 5-1 Codes, standards and regulations used for the design of spools and piping:

Table 5-1 Codes, standards and regulations for pipes

Item	Standard / Regulation	Document Title
1.	Norwegian Petroleum Directorate (NPD)	Regulations to the Petroleum Act, FOR-1997-06-653
2.	ISO 13628-1 (API 17A)	Design and Operation of Subsea Production Systems - Part 1 general Requirements and Recommendations
3.	ISO 13628-6	Design and Operating of subsea Production Systems- part 6 : Subsea Production Control System
4.	ISO 13628-15	Petroleum and Natural Gas Industries Design and Operation of Subsea Production Systems part 15
5.	ASME B31.8	Gas Transmission and Distribution on Piping Systems(Chapter VIII) 2012
6.	ASME B31.3	Process Piping 2012
7.	API RP 1111	Design, Construction, Operation, and maintenance of Offshore Hydrocarbon Pipelines, API Recommended Practice 1111 ,fourth edition, December 2009
8.	DNV-RP-F112	Design Of Duplex Stainless Steel Subsea Equipment Exposed to Cathodic Protection 2008
9.	DNV-RP-C203	Fatigue Design of offshore Steel Structures
10.	DNV-OS-F101	Submarine Pipeline systems 2013
11.	DNV-RP-F105	Free Spanning Pipelines, 2006

The reference deep water subsea project for this thesis used the Allowable Stress Design method (ASD) instead of the Limit State Design (LRFD) as outlined in the DNV codes.
The spool design principles shall be based on requirements and recommendations described in *ISO 13628-1*
The piping shall be designed according to *ASME B31.8 Gas transmission and distribution piping systems Chapter VIII offshore gas transmission*. This code is based upon the allowable stress design criteria and is applicable for systems with design temperatures in the range of -29°C to 232°C. For external pressure collapse check the pipe shall be checked against the limits as given in *DNV-OS-F101 Submarine Pipeline systems 2013* or *API RP 1111 Design, Construction, Operation, and maintenance of Offshore Hydrocarbon Pipelines*. Piping material in Super Duplex shall be evaluated in accordance with *DNV-RP-F112 Design Of Duplex Stainless Steel Subsea Equipment Exposed to Cathodic Protection, 2008*

5.2 Material Data

Table 5-2 shows material values usually used for subsea spools the 25%Cr. Super duplex material has a far better corrosion, erosion resistance and a higher material strength than carbon steel pipe. The minor side of using super duplex is that the full capacity may not be fully utilized due to risk of hydrogen Induced stress corrosion cracking (HISC) caused mainly by cathodic protection on subsea systems. Other potential sources for corrosion such as sour environments with high content of hydrogen must be considered when a material choice is made. The choice of material for the analysis case is a 6 inch water injection pipe in *grade X65 material* which is a carbon steel pipe.

Table 5-2 Piping material data

Parameter	25%Cr.Duplex - UNS S32750	X65
Yield strength (20 °C) SMYS	545 MPa	450
Tensile strength (20 °C) SMTS	750 MPa	531
Density	7850 kg/m^3	7870 kg/ m^3
Young's modulus	2.0 x 10^5 N/mm^2	2.0 x 10^5 N/mm^2
Linear expansion	13.5 x10^{-6}/°C	16 x 10^{-6}/°C

5.3 Pipe Dimensions

In addition to material data the design codes and standards define the applicable pipe dimensions for the spool, Table 5-3 shows the dimensions of the spool used in the reference project:

Table 5-3 Spool piping geometry

System	Outer diameter [mm]	Pipe wall Thickness [mm]	Schedule No.	Inner Diameter [mm]	Bend Radius [mm]	Bend wall thinning	Wall thickness tolerance
6" WI	168.3	18.3	160	131.7	457.2	10%	12.5%

The schedule No. is a number in accordance with ANSI standard for pipe wall thickness. The bend wall thinning values are typically given by the manufactures for induction bends and can vary between 2 to 20%

dependent upon size radius and wall thickness of the pipe bend. The wall thickness tolerance is according to ANSI and ASME standard for pipes.

5.4 Environmental Data

For the design case the following parameters for the subsea location yields:

Table 5-4 Environmental data

Parameter	Data
Subsea ambient temperature	4.3°C
Seawater density	1026 kg/m^3
Max Current speed (Operational)-Omnidirectional	0.7 m/s
Max. design water depth collapse	900 m
Min. design water depth to obtain design gauge pressure	700 m

Reference is also made to Table 10-5 for current distribution

5.5 Design parameters

For the design and analysis of the tie-in spool the applicable design parameters is given in Table 5-5 below and is based upon reference project values. In accordance to Statoil TR1230 Ref /19/ the test pressure requirements is given as:

- ASME B31.8 = 1.5 x (design pressure less the theoretical external pressure from static head at actual water depth (counted to MSL) but limited at maximum 1.4 x design pressure or
- ASME B31.4 = 1.25 x design pressure

Table 5-5 Design data

Parameters	6" Water Injection Jumper
Ambient temperature, subsea	Min 4.3° C, Max 21.5 ° C
Design temperature	Min -29 ° C, Max 100 ° C
Max operational temperature	34 ° C
Design collapse pressure 900 m water depth	90 bar
Density of water injection fluid	1026 kg/m^3
Internal differential design pressure (DP) at water depth 700m	345 bar
Corrosion allowance	3 mm
Erosion allowance	0 mm
Hydro test FAT pressure (1.5 x DP)	517.5 bar
Subsea Test pressure (1.25 x DP)at MSL -700m	431.3 bar
Design Life	25 years

5.6 Spool configuration

The spool configuration investigated is of the vertical ½ M type which has previous been used and installed in a deepwater project. The connector is of the vertical type landing on up-facing hubs and is closed by engaging the mechanical screw which clamps the hubs together against the seal.
The configuration and the dimension of each leg is given in Table 5-6.

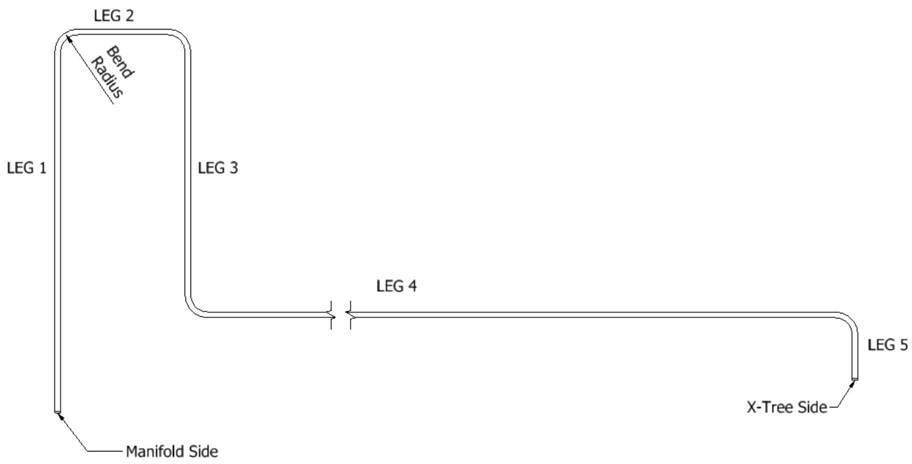


Figure 5-1 Jumper Spool Shape

The final geometry will vary within the range given by an installation tolerance and the field layout. A set of predefined length is chosen as to keep the required possible lengths to a minimum. The bend radius is R=457mm for all bends in the spool.

Table 5-6 Spool Jumper Configurations

Type	Spool length [m]	Leg 1 [mm]	Leg 2 [mm]	Leg 3 [mm]	Leg 4 [mm]	Leg 5 [mm]	Height Difference [mm]
Max	30	7267	3000	6510	27000	757	0
	30	8117	3000	6510	27000	757	850
	30	8767	3000	6510	27000	757	1500
Nom	24	7267	3000	6510	21000	757	0
	24	8117	3000	6510	21000	757	850
	24	8767	3000	6510	21000	757	1500
Min	18	7267	3000	6510	15000	757	0
	18	8117	3000	6510	15000	757	850
	18	8767	3000	6510	15000	757	1500

5.7 Installation, settlement, Spool fabrication and Metrology Tolerances

The hub to hub fabrication and metrology tolerances given in Table 5-7 will be used in the jumper analyses based upon a deep sea water project. These variables tend to vary from project to project, and are dependent upon the measuring method used, contractor's fabrication quality and experience.

Table 5-7 Installation Tolerances and Settlements

	Parameters	Data
Manifold (MF)	Position	2.0 m Radius
	Vertical Angle	± 2.5 °
	Vertical Position	+ 0.3 m
	Long Term Settlement	- 130 mm

The jumpers will be fabricated based on the field metrology report after installation of the Manifold and X-Tree. Thus, the installation tolerances are covered for by changing the jumper geometry in the analyses to find the most unfavourable configuration, governing for the jumper design. However, the settlements and the fabrication and metrology tolerances will be unknown and will have to be considered in the jumper analyses by applying various load combinations to cover the worst, most unfavourable cases. Table 5-8 and Table 5-9 show the values to be used in the analysis of the spool.

Table 5-8 Deflections and settlements

Location	Value	Description
X-Tree Deflections - Horizontal	216 mm	Deflection when using BOP
X-Tree Deflections - Vertical	25 mm	Tree well expansion
Manifold Settlement - Vertical	130 mm	Assumed settlement

Table 5-9 fabrication and metrology tolerance

Tolerance	Translation, relative hub-to-hub distance (mm)		Rotation (deg)	
	ΔX	Δy	Rx	Rz
Metrology Inaccuracy	±25	±25	±0.25	±0.25
Fabrication	±6	±6	±0.25	±0.25
Total Tolerances (Metrology + Fabrication)	±31	±31	±0.5	±0.5

Figure 5-2 shows the location of the spool tolerances

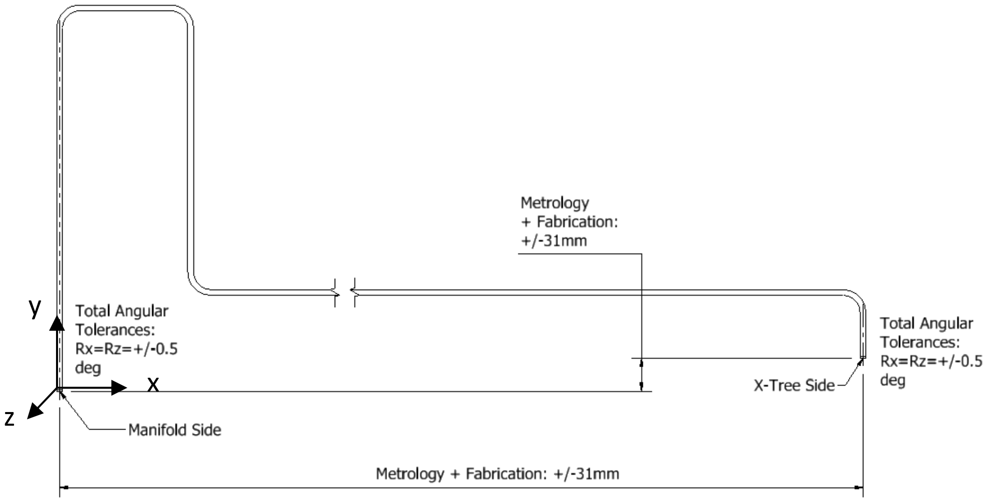


Figure 5-2 Jumper spool tolerances

In the analysis the longitudinal deflection and fabrication tolerances is assumed to be of a uniform distribution which is used when the only information known about the parameter variability is the upper and lower bounds. For the angular tolerances a normal statistical distribution is applied. (Refer to chapters 6.2). The reference project did not account for any statistical scatter in the tolerances and imposed deflections. A range of predetermined max/min for deflections and tolerances was combined and analysed together with the acting loads.

5.8 **Load Cases**

During the different phases from construction to the end of design life the jumper piping will be subject to different loads. The following loads and or load cases should be considered as potential loads during the jumper life cycle.

- Pressure testing (Onshore and subsea)
- Lifting/Lowering through splash zone
- Connection
- Operation
- Slug loads
- Shutdowns
- Snag loads
- Retrieval
- Earthquake
- Vortex Induced Vibration (VIV)
- Fatigue Evaluation
- Erosion
- Seal replacement
- Dropped object loads

However, the investigation of this study is limited to only the following cases:

- Pressure testing onshore and subsea
- Connection/operation and seal replacement (in-place)
- VIV fatigue evaluation

For a more comprehensive load investigation readers of this thesis are encouraged to study Jone Lutro's Master thesis Ref. /3/. In his thesis the main topic is a LRFD design of a vertical spool in accordance with DNV-OS-F101 Standard.

5.9 Design Code Check

5.9.1 Code formulas

Pipe wall thickness

In accordance to ASME B31.8 the allowable stress limits for Internal design pressure for the nominal wall thickness for a given design pressure is given by the following equation:

$$W_t = \frac{(P_i - P_e)}{2S} \cdot \frac{1}{FT} + c \tag{5.1}$$

Where:

Wt	=Pipe wall thickness	[mm]
P_i	=Internal design pressure	[N/mm^2]
P_e	=External pressure	[N/mm^2]
S	= Specified minimum yield strength	[N/mm^2]
F_1	= Design factor	[-]
T	= Temperature de-rating factor	[-]
c	= Corrosion allowance	[mm]

The design factor F1 depends on actual Location Class for the piping. The subsea manifold is defined as part of the pipeline and the design factor for hoop stress is F1 = 0.72 (Table A842.2.2-1). The temperature de-rating factor T is equal to 1 for temperatures up to 121° C (Table 841.1.8-1 for carbon steel only)

Hoop Stress

The hoop stress due to internal design pressure must fulfil the criteria given in ASME B31.8 section A842.2.2 for D/t ≥30

$$S_H = (P_i - P_e)\frac{Dt}{2t} \leq F_1 ST \tag{5.2}$$

And for D/t≤30:

$$S_H = (P_i - P_e)\frac{D - t}{2t} \leq F_1ST \quad (5.3)$$

Where:

p_i	=	Internal design pressure
p_e	=	External pressure
D	=	Outer diameter
S	=	Specified minimum yield strength
F1	=	Hoop stress design factor, here F1 = 0.72
T	=	Temperature de-rating factor
t	=	Nominal wall thickness

In piping design usually the fabrication tolerances is to some degree accounted for in the stress limits and the stress intensification factors (SIF) as given in the piping code (Ref. /15/ Table E-1 and Table A842.2.2-1) . The stress limit for wall thickness design in ASME was historically developed based upon the wall thickness tolerance of 12.5% and a safety margin for hydro testing of 1.25xMOP to achieve 90% of SMYS resulting in a max usage factor of 0.72.

Longitudinal stress

For subsea pipelines the longitudinal stress shall fulfil the following

$$|S_L| \leq F_2S \quad (5.4)$$

Where:

S	=	Specified minimum yield strength
F_2	=	Longitudinal stress design factor (Table A842.2.2-1), here F2 = 0.8
S_L	=	Maximum longitudinal stress

The maximum longitudinal stress is given as the sum of the longitudinal pressure stress, the longitudinal bending stress and the axial stress due to sustained and thermal loads:

$$S_L = |S_P| + |S_b| + |S_{axial}| \quad (5.5)$$

Where:

S_L	=	Longitudinal stresses
S_P	=	Longitudinal pressure stresses
S_b	=	Longitudinal bending stresses
S_{axial}	=	Axial stress

The bending stress is calculated by use of the following formula:

$$\sigma_b = \frac{\sqrt{(i_iM_i)^2 + (i_oM_o)^2}}{Z_{nom}} \quad (5.6)$$

Where:

M_i= In plane bending moment
M_o= out of plane bending moment
Z_{nom}= Section modulus, nominal wall thickness

$i_{o,i}$=Stress intensification factor (SIF) out of plane or in plane

Combined stress

The combined stress shall not exceed the value given by the maximum shear stress equation (Tresca combined stress) A842.2.2 (c)

$$2\left[\left(\frac{S_L - S_h}{2}\right)^2 + S_t^2\right]^{1/2} \leq F_3 S \tag{5.7}$$

or alternatively, the value given by the Maximum distortional energy theory (von Mises combined stress)

$$\sqrt{S_h^2 - S_h S_L + S_L^2 + 3S_t^2} \leq F_3 S \tag{5.8}$$

Where:

S_C	=	Combined stress
S_t	=	Torsional stress
S_L	=	SL_{max} or SL_{min}, whichever is greater in magnitude
SL_{max}	=	Maximum longitudinal stress (Sa+Sb) or (Sa-Sb),
S_b	=	Bending stress
S_t	=	Tangential stresses ≈ Torsional stress=M_t/2Z (M_t= Torsional moment)
Z	=	Section Modulus
S_H	=	Hoop stress
F_3	=	Combined stress design factor (table A842.2.2-1), here F3 = 0.9

The wall thickness must be reduced with corrosion allowance, mill tolerance and erosion allowance for platform piping and risers when calculating combined stress. The spool is considered as part of the pipeline and hence the nominal pipe wall thickness is used in the combination stress check

5.9.2 HISC Stress limits

If a pipe is made from Duplex or Super Duplex then Hydrogen Stress Induced Cracking can be a potential problem and the pipe must be checked in accordance with Ref. /32/ Section Sec. 4, D302. The RP or recommended practice presents HISC as a separate failure mode, which should be analyzed for duplex components used subsea. The design requirements in the RP include stress and strain limits, as well as a number of other factors. It is Important to note that *the RP does not* give load factors to the design this must be considered separately.

The first step in evaluating protection against HISC is a screening criterion where the longitudinal stress from the piping analysis is used together with a calculated stress concentration factor. From this step it is possible to identify the most utilized spool configuration, and hence what type of geometry to use in further analysis in the event of this screening criterion should fail. The next step consists of a linear elastic analysis by use of FEA. Stress classification lines are added to the geometry. From the FEA analyses the membrane and membrane plus bending stresses are extracted and compared to given limits.

When the linear elastic analysis criterion is not met, a non-linear analysis is required. Here the Neuber`s rule is used as a first approximation, and the corresponding non-linear stresses and strains are calculated and compared to given limits.

The final step in a detailed HISC evaluation should all above criterions fail is a full elastic plastic model with as close to real geometry as possible, and different material curve for all material used in the construction. This analysis would give a deeper insight in the structures true physical behavior. The first criterion to check for piping analysis is given by eq. (5.9) and is:

$$SCF \times \sigma_l < (SMYS - Derating) \times \gamma_{HISC} \times \alpha_m \tag{5.9}$$

Where:

SCF = Stress Concentration Factor (Welds, transitions, etc.)

σ_l = Longitudinal stress extracted from the pipe element used in the analysis

SMYS = Specified Minimum yield Strength

De-rating =Temperature de-rating for yield strength at operational temperature

γ_{HISC} =Coarse austenite spacing or fine (0.85 or 1.0 Table D.2)

$\alpha_m = 0.8$, (Ref. /32/ Sec. 4, part 302)

A design where the membrane stress in the component is below 80% of $\gamma_{\mathrm{HISC}} \cdot \mathrm{SMYS}$ is acceptable. The peak stress can be disregarded. (Ref. /32/ figure 2). The criterion is for detailed FEA linear elastic analysis of the given geometry.

5.9.3 Code Stress Limits

The stress criteria are applicable for the jumper spool design see Table 5-10 . The equivalent and the longitudinal stress criteria are applicable for smooth pipe cross sections without welds, grooves & fillets transitions etc. The stress intensification factors (SIF) are included in the code formulas and must always be checked where applicable. The HISC criterion is for the max allowable membrane stress Ref. Table 5-11 and section 5.9.2

For hydrostatic testing of pipe the ASME B31.8 code does not specify any stress limits and the committee is silent regarding this case see interpretations volume 16 in Ref. /15/. A factor of 0.96 x SMYS is used for the check towards the combination stress as described in Statoil specification given in Ref. /19/. The limit for hoop stress and combination stress is also given in Section 7 E100 of Ref. /7/. Here the min of (0.96xSMYS or 0.84xSMTS) is specified for hydrostatic mill pressure test which is based upon minimum wall thickness t_{min}. If a standard API pipe is purchased with the 12.5% fabrication tolerance on the wall thickness, it seems reasonable to use the 0.72xSMYS for the hoop stress criteria so that the pipe is not utilised more than 90% of SMYS during the test for the minimum wall thickness. However the benefit of raising the pressure test to 100% of SMYS is shown in Figure 5-3 and is given in publications published by John F. Kiefner and Willard A. Maxey found in the pipeline handbook Ref. /34/. Here curves for 9 flaws depth to wall ratios are given. For example consider the horizontal line in Figure 5-3 for 100% SMYS here the longest surviving defect that is 50% through the wall can only be about 3.5 inches. Compare that length to the length of the longest possible 50%-through flaw at the MAOP which is 10 inches.

Hence smaller flaws are assured by even higher pressure. In short the higher the test pressure is above MAOP the smaller the possible surviving flaws are.

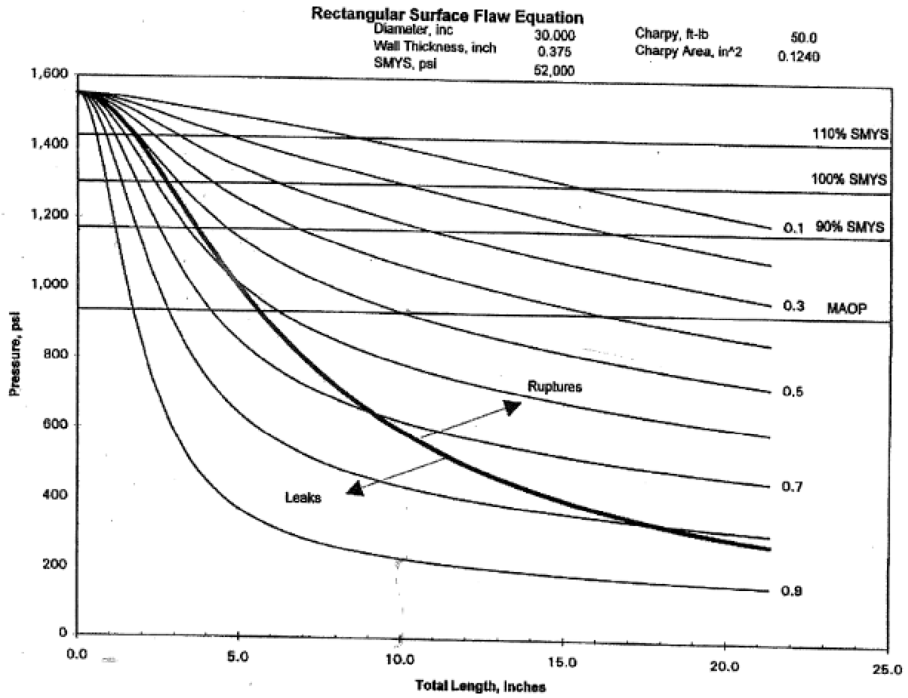


Figure 1. Impact of test pressure levels on margin of safety.

Figure 5-3 Impact of test pressure levels on margin of safety Ref. /34/

Table 5-10 and Table 5-11 list the allowable stresses for the spool design based upon the code formulas as given in this chapter. Table 5-11 is included as information only since the pipe in the analysis is of carbon steel *grade X65.*

Table 5-10 Allowable stress piping X-65 grade

Criterion	Reference	Temperature 0°C -121°C (250^{o}F or less)
Yield Strength	ASME B31.8	450 MPa
Hoop Stress	ASME B31.8	324 MPa
Longitudinal Stress	ASME B31.8	360 MPa
Combination Stress-Operational	ASME B31.8	405 MPa
Combination Stress-Hydro testing	Statoil TR1230	432 MPa

Table 5-11 Allowable stress Super Duplex piping

Criterion	Reference	Temperature			
		20°C /25°C	**30°C /44°C**	**70°C /75°C**	**105 °C /112°C**
Yield Strength	DNV-OS-F101	545 MPa	526 MPa	495 MPa	465 MPa
Hoop Stress	ASME B31.8	392 MPa	379 MPa	356 MPa	335 MPa
Longitudinal Stress	ASME B31.8	436 MPa	421 MPa	396 MPa	372 MPa
Combination stress	ASME B31.8	491 MPa	473 MPa	446 MPa	419 MPa
HISC	DNV-RP-F112	357 MPa	350 MPa	340 MPa	335 MPa
Combination Stress-Hydro testing onshore	Statoil TR 1230	523 MPa	505 MPa	NA	NA

6. SPOOL OPTIMISATION AND STRENGTH VERIFICATION

6.1 Finite element program ANSYS

For study of subsea flow lines and tie-in spools the most utilized FEA software used today is the ANSYS software package. The ANSYS software package is a large tool with many capabilities. The program offers many functions and can be used for many areas spanning from structural /mechanical fluid problems to advanced areas within physics. Readers can visit the ANSYS Website for a comprehensive description of each analysis packages
Some of the main products are:

Structural analysis:

- ANSYS- Multiphysics
- (Includes all of finite element disciplines)
- ANSYS Mechanical
- ANSYS Structural
- ANSYS Professional
- ANSYS DesignSpace
- ANSYS ACT
- ANSYS Rigid Body Dynamics
- ANSYS Composite PrepPost
- ANSYS nCode DesignLife
- Explicit Dynamics
- ANSYS Explicit STR
- ANSYS Autodyn
- ANSYS LS-DYNA

Fluid dynamics:

- ANSYS Fluent
- ANSYS CFX
- ANSYS CFD
- ANSYS CFD-Flo
- ANSYS CFD Professional
- FLUENT for CATIA V5
- ANSYS CFD-Post
- ANSYS Icepak
- ANSYS Polyflow
- ANSYS Vista TF
- ANSYS BladeModeler
- ANSYS TurboGrid

In this thesis the ANSYS Structural version 15 is used. This version has the capability to handle linear and non-linear analysis, buckling, contact and dynamic analysis problems. The package includes the ANSYS Workbench and ANSYS APDL classic graphical user interface. Several add-ins is provided to the package such as CAD interface with most of the common parametric design tools (Inventor, Solidworks, CATIA, Pro engineer etc.) in addition to the ANSYS Design modeller.

ANSYS has a large library of different element types with a variety of boundary conditions, material properties, and other relevant data. The user can determine to utilise the powerful APDL programming language and make scripts similar to what students learn by using the software CALFEM in finite elements topics. The APDL Programming language takes time to learn and to programme the input script, and has to some degree the disadvantages of being a source of programming error due a large number of options that has to be carefully checked towards the ANSYS documentation or other software. The benefit of APDL is that it is quick to change parameters, especially for simple beam and shell constructions and also to perform large numerous load combinations in one analysis run. That is also why most of the pipeline engineering is conducted with this type of programming. A simulation of a long pipeline uses contact elements and pipe elements typically CONTA175, PIPE288 PIPE289 (for Version 15 or higher). Figure 6-1 shows the vertical spool modelled in ANSYS Classic.

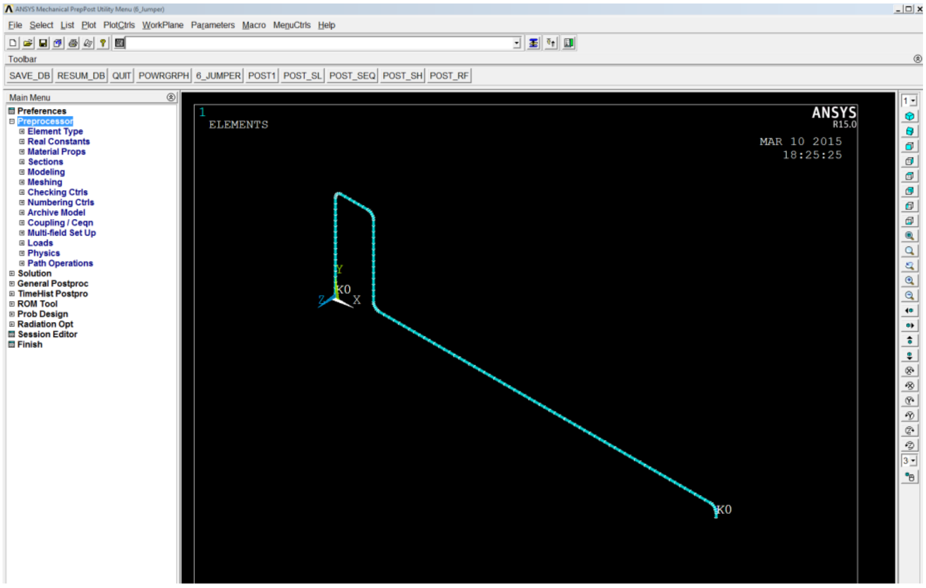


Figure 6-1 ANSYS Classic GUI

The other cousin of ANSYS shown in Figure 6-1 is the ANSYS Workbench shown in Figure 6-2 which allows a user to enter a more Windows operated GUI without the in depth knowledge of all the programming techniques. This GUI is quite Intuitive and is supplied with many advanced features including the advanced automatic mesh generator for complex shapes and constructions. This tool is highly utilised in mechanical and CFD analysis. The program allows the user to also include APDL Programming. The software includes all of the capabilities as listed above.

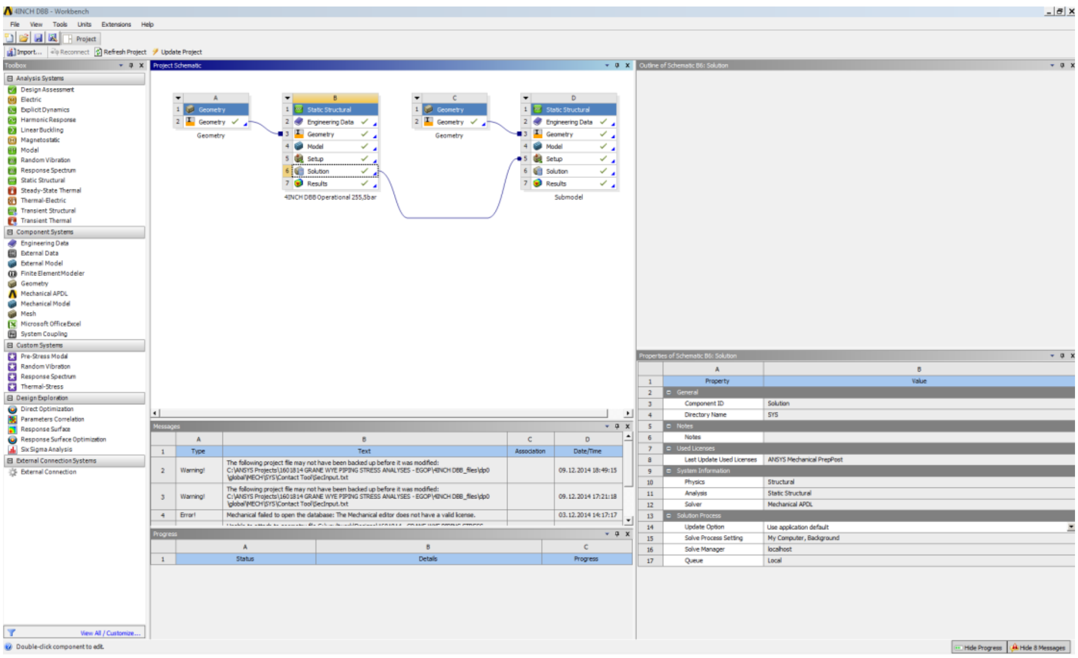


Figure 6-2 ANSYS Workbench environment

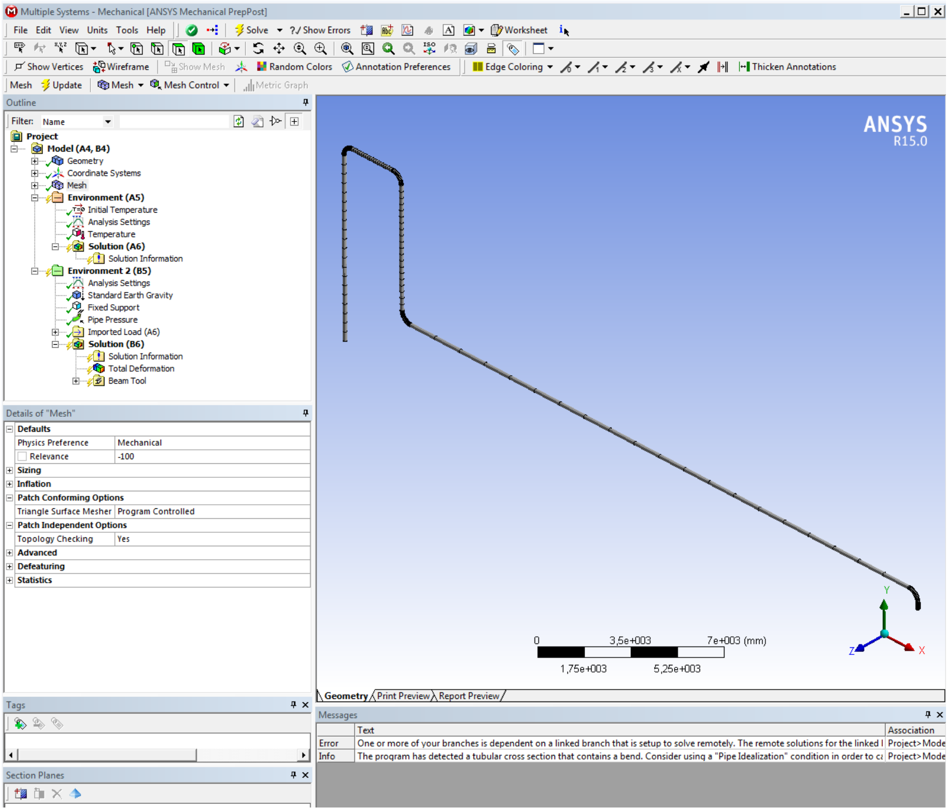


Figure 6-3 ANSYS Workbench Pipe model

In this thesis the add-in tool included in the ANSYS Workbench Environment called ANSYS DesignExplorer is used. This tool allows you to vary and study the effect and sensitivity of design and load parameters. Here the user can specify a type of probability function for each parameter and conduct design experiments and load combinations. The add-in also has a statistical feature named “six sigma analysis” a tool that allows the user to show probabilities for a safe design. The tool also includes response surfaces where sensitivities of parameters can be studied. An optimisation tool is also provided which lets the user explore the best candidates for the design within the user specified design limits. The graphical layout is shown in Figure 6-4

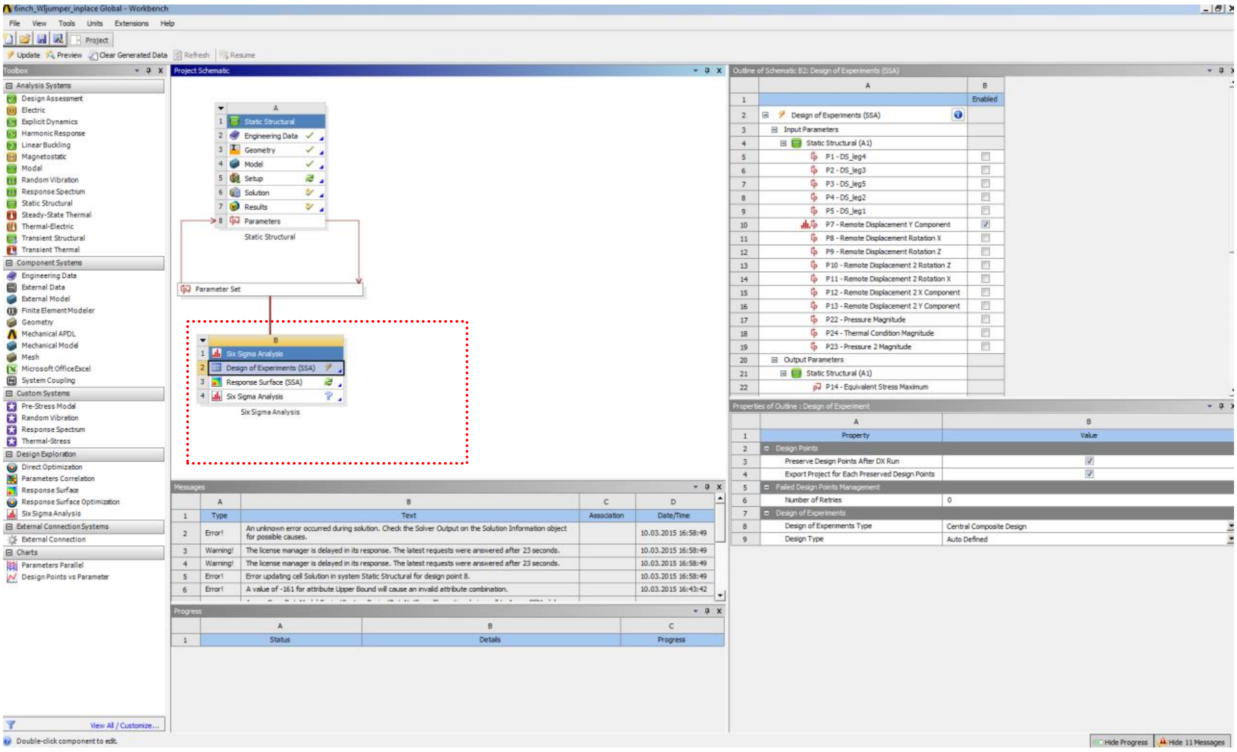


Figure 6-4 GUI ANSYS WB Design Explorer

The tool provides many advanced functionalities and is beyond the scope of this thesis but some of the features is mentioned here see (see Appendix 5) for a quick intro.

Useful tools utilised is:

- Design of experiments:
  -Variables are assigned a statistical distribution (Normal, Uniform, log-normal, triangular, Weibull ,Beta etc.)
  - Design points (combinations) are generated automatically to explore the parametric space efficiently
- Six Sigma analysis:
  -Probability distribution for the output parameter (stress, reaction forces etc.)
- Response surface
  -Which parameters contribute most and identification of sensitivities to input parameters

The optimisation tool is mainly to screen out the best candidates for the design based upon limit constraints such as safety factors, minimum stress and weight etc. For the spool the final dimension is unknown in the early phase of the project and hence a selected type and range of possible candidates needs to be checked.

6.2 Analysis Description

As described in chapters 5.6 and 5.7 the spool has to accommodate many possible deflections, lengths and rotation combinations. There are 6 DOF at each end with 2 possible variations. In theory this gives a total of 2^{12} =4096 combinations for the deflections and rotations. So for the predetermined length Max, Nom and Min, which is the target length range for the final geometry with 3 configurations each, results in a large amount of combinations. (4096 x 9 = 36864 possible combinations for each spool). Hence engineering judgment and experience is required in order to constrain some of the possibilities here.

The following assumptions for the boundary conditions are used for the "in-place model" in order to reduce the number of possibilities see Figure 6-5:

Dz=0, Since this spool is of the vertical type the main stiffness is in the x-y plane and hence the out of plane deviations and tolerances is assumed to have a minor influence on the spool.

Ry=0, Since the length dimension in the x-direction is much >> than the imposed tolerance rotation among Y-axes it is assumed to have a minor influence on the spool stresses.

Dx_{MF}=0, The longitudinal variation of the spool is imposed at one end (X-tree end, Dx_{XT}) It is assumed that fixating one end and pulling the spool is the same as pulling two ends apart for the same amount of tolerance.

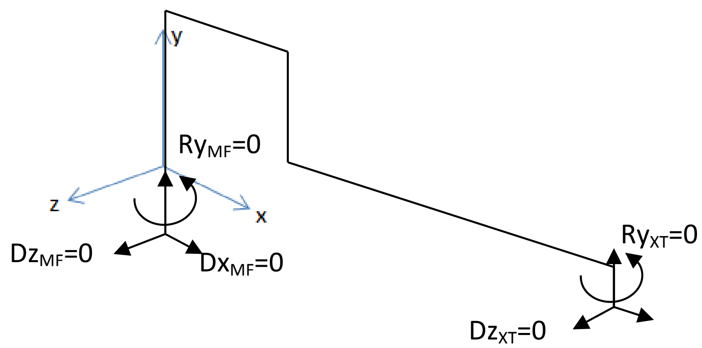


Figure 6-5 Boundary conditions spool "In-place model"

As a first step the analysis is divided into several sub cases in order to capture the sensitivity of the imposed deflections and rotations of the geometry. Here statistical distribution is given for the tolerances and deflections. The piping code specifies that production tolerances and installation tolerances always must be considered in the design calculations.

In accordance to chapter 3.6 a statistical distribution of the tolerances should be applied in the analysis. Table 6-1 and Table 6-2 show the distribution values used in the analysis.

Table 6-1 Statistical Distributions

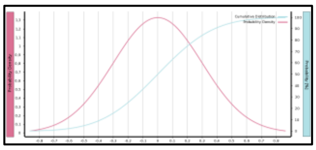
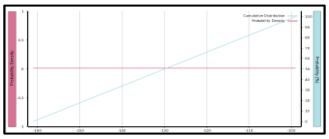
Parameter	**Distribution type**	**Commentary**
Angular tolerances	Normal distribution 	A mean value with a standard deviation is assumed. It is assumed that there is a 95% probability of hitting the max/min angular tolerances of +/- 0.5deg with a 100% dimensional control.
Imposed deflections and linear fabrication tolerances	Uniform distribution 	Here a linear uniform distribution is assumed for the max and min limits. Deflection + linear fabrication tolerances are added as no information about the dimensions and deflection is given. The greatest uncertainty in this assumption is the expected imposed deflection.

Table 6-2 Distribution values

location	**Dx**	**Dy**	**Rx**	**Rz**
Manifold side	0=fixed	=-130mm +/-(25mm+6mm) =-130+/-31mm	μ_{mean}=0 Std.=0.3	μ_{mean}=0 Std.=0.3
X-tree side	=+/-216mm+(25mm+6mm) = +/-247mm	=-25mm +/-(25mm+6mm) =-25mm+/- 31mm	μ_{mean}=0 Std.=0.3	μ_{mean}=0 Std.=0.3

Figure 6-6 and Figure 6-7 shows the detail of the distributions for the rotations and deflections and their probability functions.

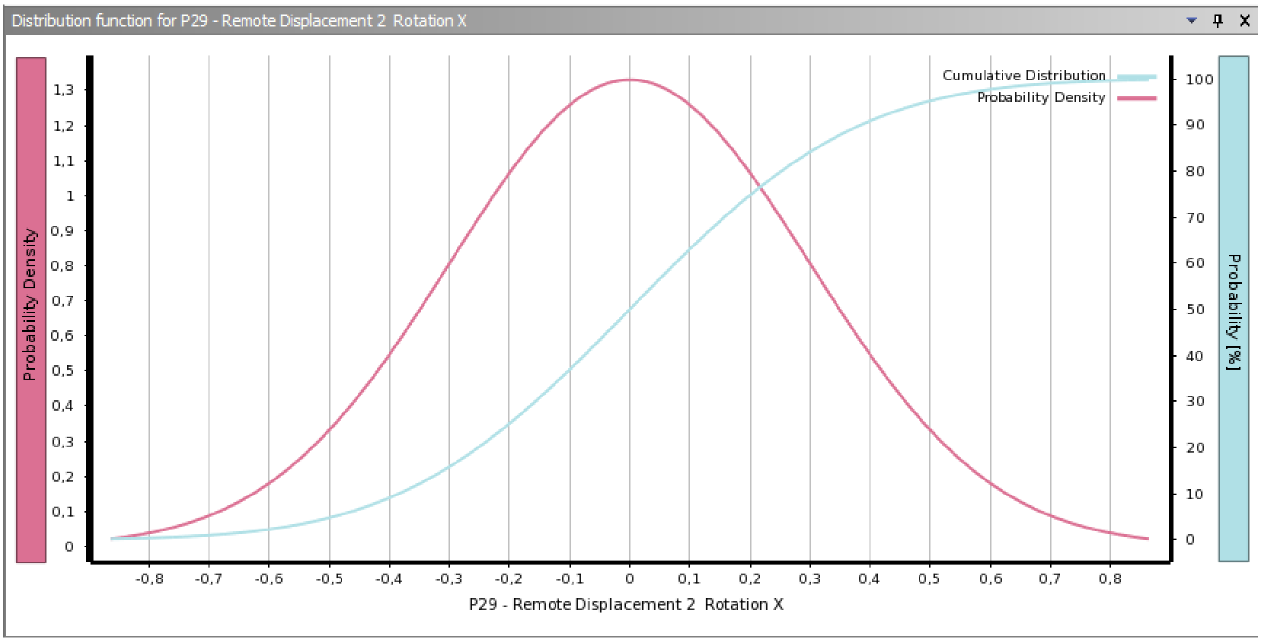


Figure 6-6 normal distributions for rotations

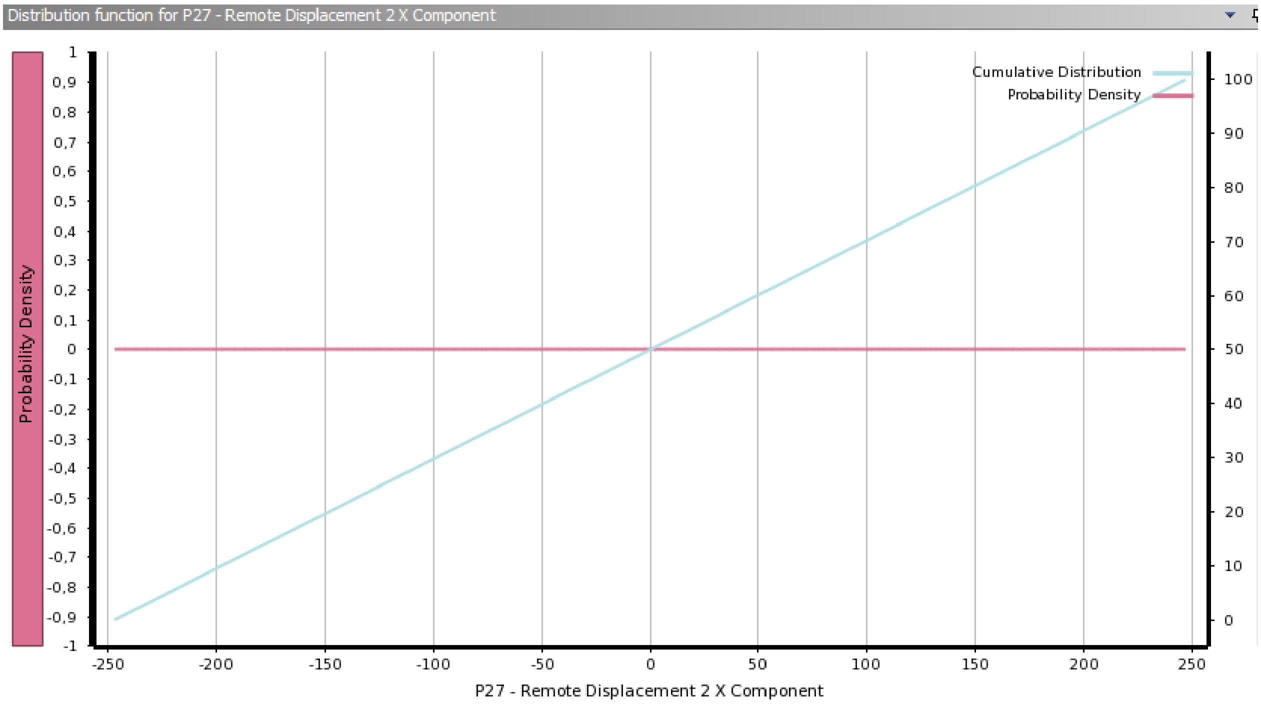


Figure 6-7 Linear max/min distribution of imposed deflections

6.3 Finite Element Model Description

The jumper spool geometry is modelled in Autodesk Inventor where a parametric setup has been established for the length of the legs. A surface is then extruded along the path with the pipe profile. In order to simulate the end cap effect of a pressurized system a solid lid is modelled in each end of the spool. This simulates the interfaces to the thick wall “hub” located in the connector which is welded onto the pipe. The assumption made here is that the stiffness of the “hub” and the connector is much larger than the spool pipe wall, and hence the boundary condition can be treated as approximately ridged.

The model is then transferred in to the ANSYS workbench environment see Figure 6-9, where a model is set up and meshed with shell elements and solid elements. A finer mesh in the contact area between the lid and the shell element is required to get sufficient contact nodes and to avoid errors and convergence problems. The meshing tool requires some trial and error in order to get an acceptable quality. The software provides many advanced settings for meshing and tools for checking the quality such as “mesh metrics”. For the study of the spool a coarse model is chosen as to limit the amount of elements and time to run each analysis. The element property can be reached by the FE tool provided, and is described in the ANSYS documentation through the help menu. As an alternative to modelling in Autodesk Inventor for geometry, one could use either the ANSYS Design Modeller or construct the geometry in the ANSYS classic environment. The modelling process is shown in Figure 6-8.

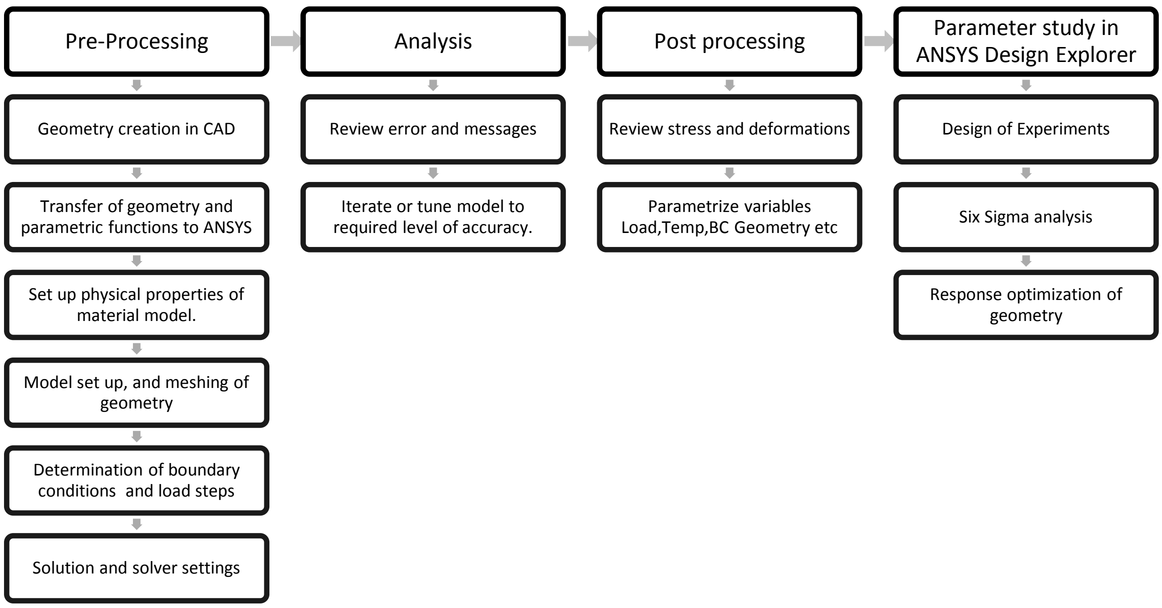


Figure 6-8 ANSYS FEA Flow chart

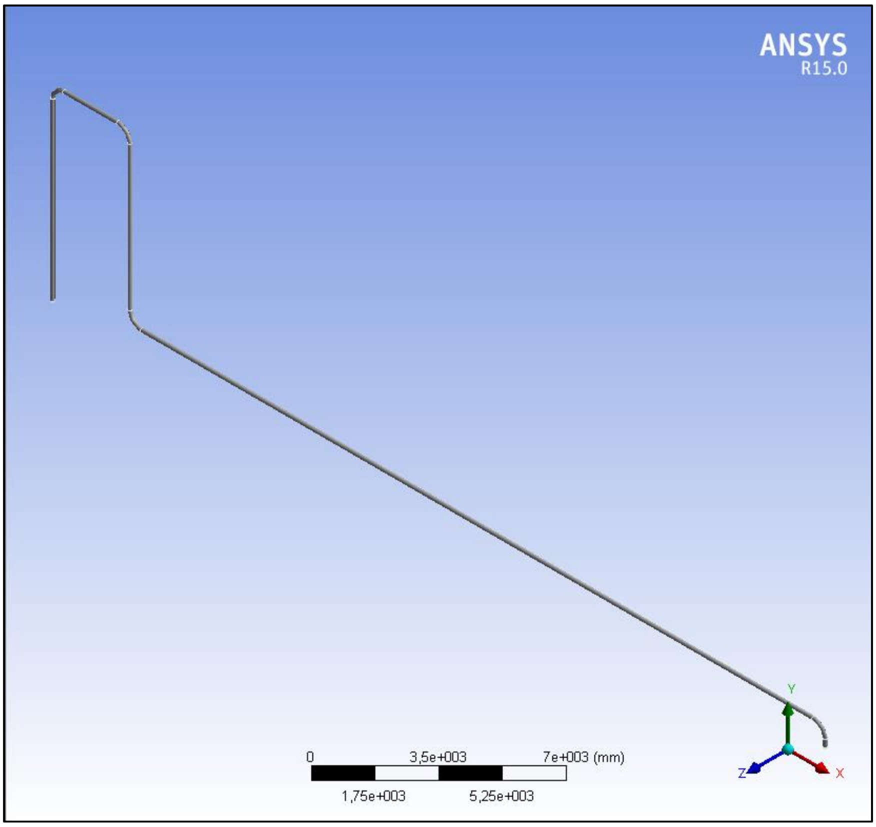


Figure 6-9ANSYS FEA Model

Figure 6-9 and Figure 6-11 shows the geometry and the mesh used in the analysis. Figure 6-10 shows the overall quality of the mesh in the model. The Element Quality option provides a composite quality metric that ranges between 0 and 1. This metric is based on the ratio of the volume to the sum of the square of the edge lengths for 2D quad/tri elements, or the square root of the cube of the sum of the square of the edge lengths for 3D elements. A value of **1** indicates a perfect cube or square while a value of **0** indicates that the element has a zero or negative volume. The shell element generated here is called Quadrilaterals with 4 nodes and has an overall quality of 0.95 to 0.98 which is considered to be very good. A total of **16565** nodes and **16241** elements are generated in the model.

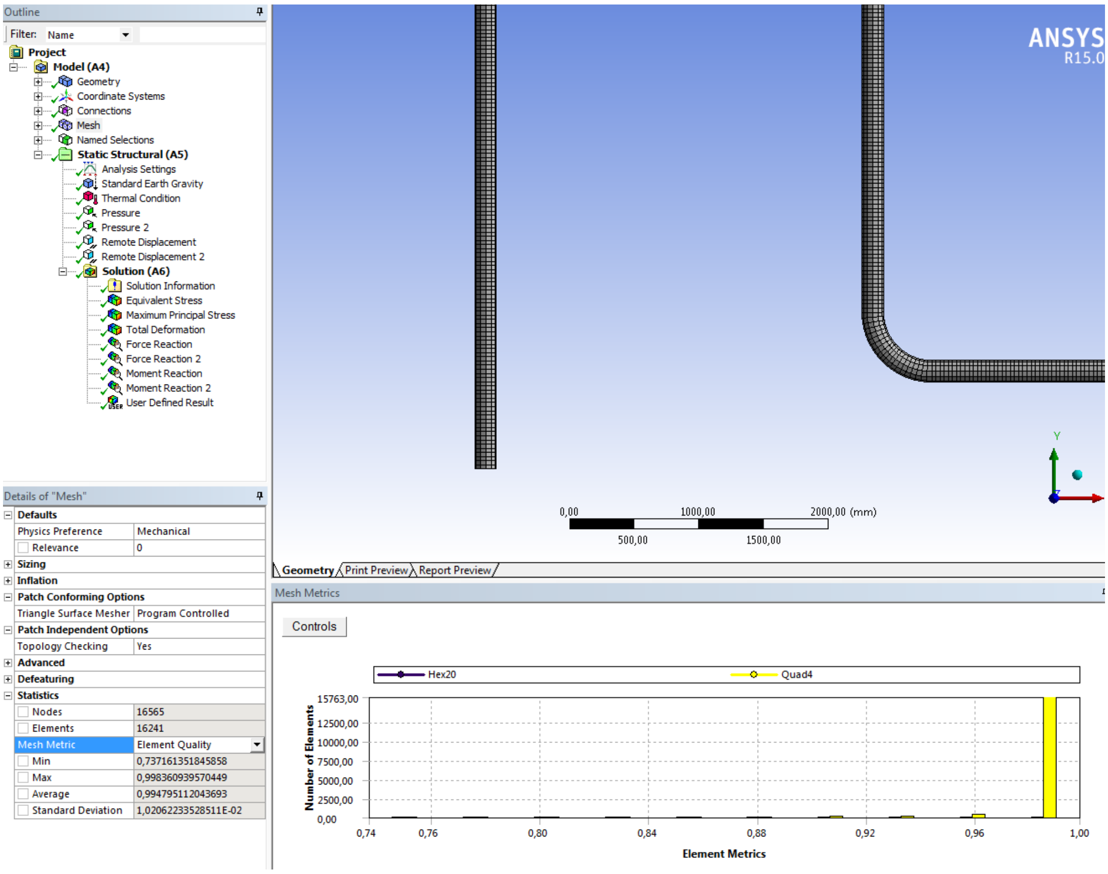


Figure 6-10 Element quality metrics

The element geometry is set up by ANSYS. ANSYS is using an MESH200 command (Ref. ANSYS documentation) to set up the geometric mesh without any element attributes such as material properties, real constant. Then the geometric mesh is automatically assigned element types. The following elements is used

- SHELL181
- SOLID186
- TARGET170
- CONTA174
- SURF184

Shell element property 181 is shown in Figure 6-12

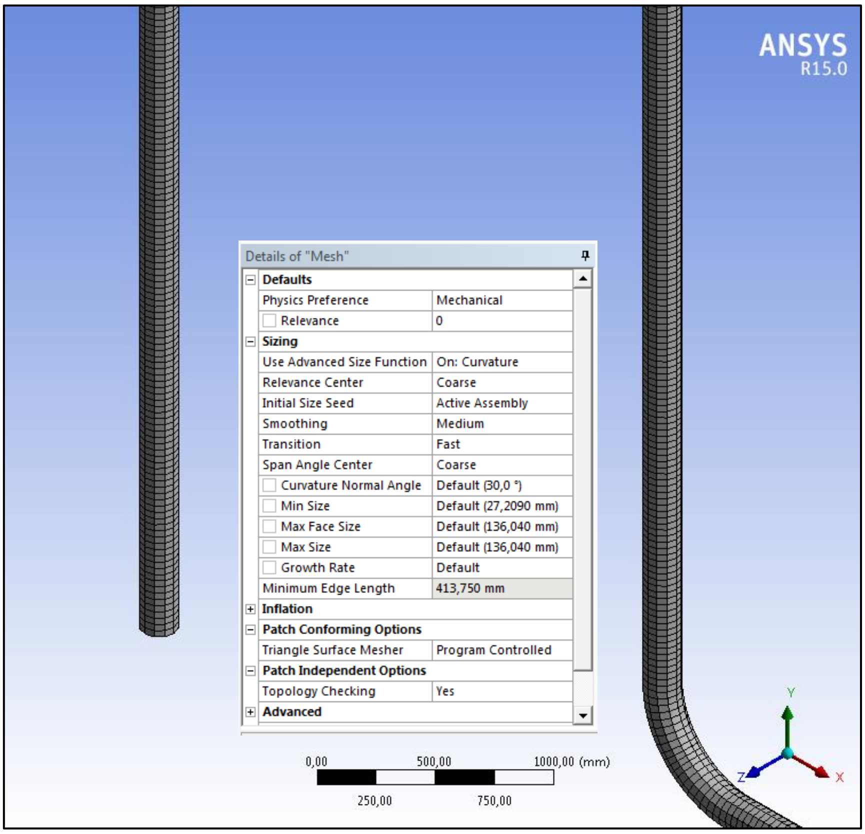
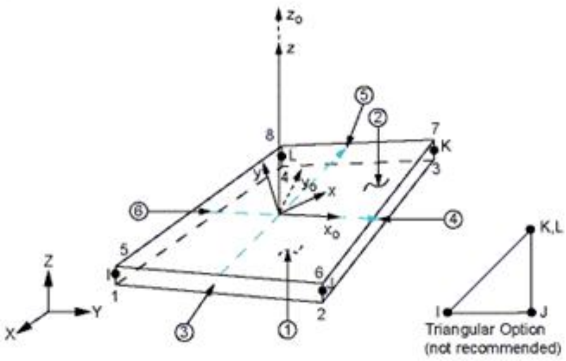


Figure 6-11 ANSYS FEA model Shell Elements

As seen from Figure 6-11 and details of "Mesh", minimum element size used by the software is 27 mm. Since this analysis is going to be repeated and there are some iteration processes, it is advisable not to use to fine mesh as this takes a longer computation time. The requirement for a fine mesh is not deemed applicable to a large structural global model. If there is a requirement to study local effects such as peak stress or stress raisers at discontinuities, welds or transition etc., a local detailed model is preferred with a finer mesh.
ANSYS has the capability of doing sub modelling from a global local model to a local detail model with a finer mesh. Figure 6-12 shows the element description e from the ANSYS documentation. SHELL181 is suitable for analysing thin to moderately-thick shell structures. It is a four-node element with six degrees of freedom at each node: translations in the x, y, and z directions, and rotations about the x, y, and z-axes. (If the membrane option is used, the element has translational degrees of freedom only). The degenerate triangular option should only be used as filler elements in mesh generation. Through wall stress gradients has to be accounted for D/t<10 in accordance with thin wall theory. The ratio is 168.3/18.3=11 which is at the limit and hence the element is deemed suitable.
SHELL181 is well-suited for linear, large rotation, and/or large strain nonlinear applications. Change in shell thickness is accounted for in nonlinear analyses. In the element domain, both full and reduced integration schemes are supported. SHELL181 accounts for follower (load stiffness) effects of distributed pressures.

SHELL181 Input Data

The following figure shows the geometry, node locations, and the element coordinate system for this element. The element is defined by shell section information and by four nodes (I, J, K, and L).



x_o = Element x-axis if ESYS is not provided.

x = Element x-axis if ESYS is provided.

Figure 6-12 ANSYS SHELL181 Element

6.4 Material Properties

The analysis of the spool is a linear material model analysis. In order to compensate for the submerged flooded weight due to buoyancy forces, the density factor for the steel has been reduced by 15% (see Appendix 3 for calculations). In accordance with ASME B31.8 there is no temperature de-rating for steel pipe for temperatures up to and including 121°C Ref. /15/ Table 841.1.8-1. The physical properties at elevated temperature are extracted from tables given in at ASME BPVC Ref. /16/. The following values are used, see Table 6-3.

Table 6-3 Analysis Material Properties

Material type	X65
Yield strength (20 °C)	450
Tensile strength (20 °C)	531
Density	6672 kg/ m^3
Young's modulus	2.0×10^5 N/mm^2
Linear expansion	16×10^{-6}/°C

6.5 Spool Loads

The load cases for the spool are described in Table 6-4. Due to the amount of work involved in assessing and checking all load cases as described in chapter5.8, the analysis cases are limited to load cases 2, 3, and 4 in this chapter.

Figure 6-13 shows the imposed rotations and deflections which was produced by the DOE (Design Of Experiment) function within the ANSYS Design Explorer environment based upon the statistical distribution of the input parameters for all of the 9 configurations as described in chapter 5.6. ANSYS Design Explorer then creates a set of possible combinations of the imposed rotations and deflections based on type of experiment. Here the default value is used, which is called “Central composite design”. The result of these set of combinations is shown in Table A2-1 . This is then done for the 9 spool configurations. The combinations and the result are then used as basis for statistical calculations provided by the ANSYS “six sigma tool”. The results of this is a probability distribution for stresses, loads and reaction forces which then can be evaluated and further assessed in the optimisation tool.

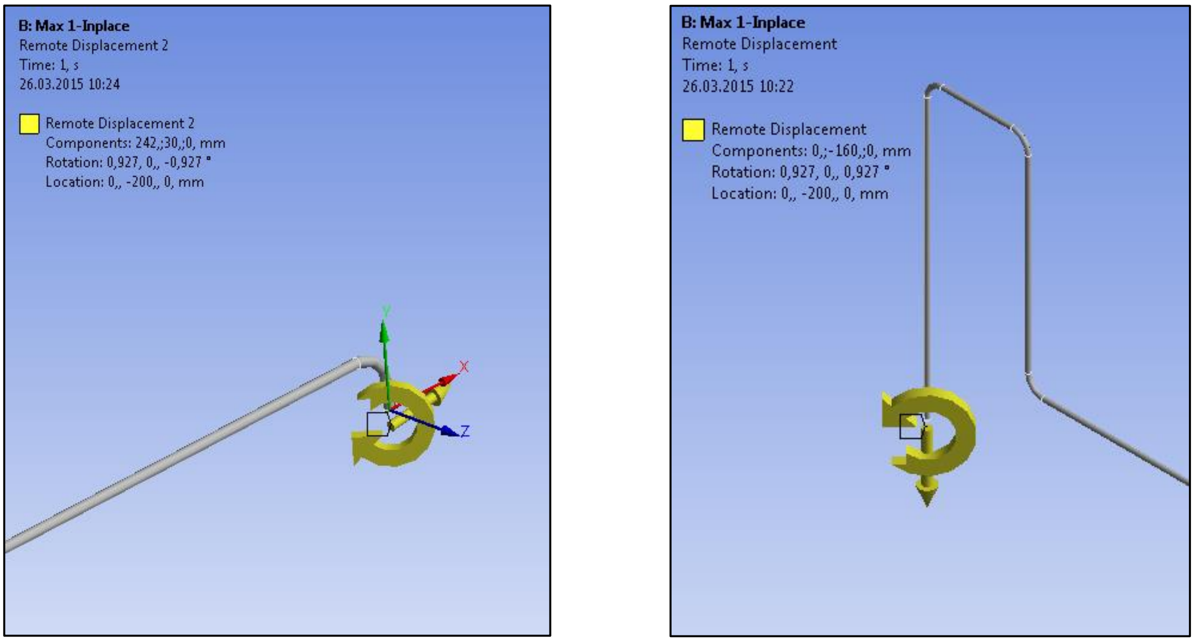


Figure 6-13 Spool End Constraints and Boundary Conditions

Table 6-4 Load description for jumper spool

Load case	Load Description	Description
1	Connection	The clamp connectors are closed. This will force the pipe hub ends into position and close the two hubs together and seal. Hence many possible combinations of angular and linear tolerances from fabrication, metrology and height difference from settlements can occur. Drag loads from current is applied
2	Subsea System Pressure Test	The subsea pressure test is simulated by applying the test pressure and the pipeline expansion. According to the subsea test pressure defined in Sec. A847.2 in ASME B31.8, the installed pipeline system shall be hydrostatically tested to at least 1.25 times the maximum allowable operating pressure. A value of 1.25 x DP is applied. Drag loads from current is applied. BOP is inactive and no deflections from XT is applied
3	Operation	The operational pressure, temperature and pipeline expansion is simulated. The settlements, XT-deflections and the linear fabrication and metrology tolerances are applied as deflections at the jumper end nodes. Drag loads from current is applied
4	Seal replacement	Temperature and pressure is removed, and then the seal is replaced by un-clamping one of the hubs end. The jumper end is stroked 500mm in vertical direction. Drag loads from current is applied
5	Slugging	Slug loads can occur from production wells. This is not considered applicable for water injection spools.
6	VIV	VIV effect and fatigue assessment is checked in accordance with Ref./25/

The operational loadcase for the spool analysis is shown in Figure 6-14.

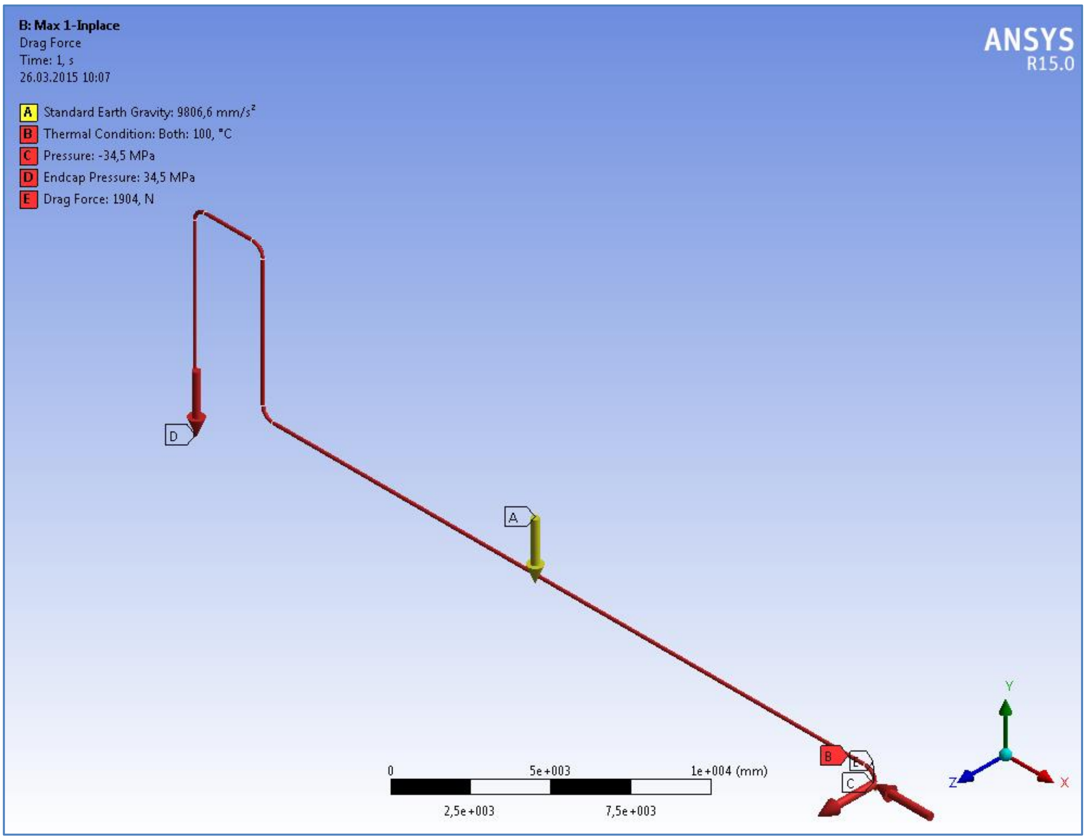


Figure 6-14 Spool Loads

7. ANALYSIS RESULTS

7.1 ANSYS Design Explorer Result Operational loads

The stress level from the ANSYS Six Sigma analysis is presented in the table below. As seen from the table column marked red the max/min search from the response surface algorithm calculates max stresses above the allowable stress limit for all configurations. However the probability of reaching theses stresses is very low. The highest probability of exceedance of the allowable stress limit as set forth in the code is the "Max 1" and "Max 2" configuration with a probability of exceedance of $4{\cdot}10^{-7}$,this equals 1 event out of 2.5 million (high Sigma level),events where the stress can rise at that level or higher with the imposed deformations at the given design pressure and temperature.
More interesting is the 10^{-4} probability which is marked green and yellow here the 2 configurations is slightly above code limit of 405MPa. The mean stress for all configurations is well below the allowable limit and the standard deviation is also in a narrow range within the acceptable code limit of 405 MPa for the Pipe.

Table 7-1 Spool configurations versus von Mises stress and probability level

Spool Type		Mean stress [MPa]	Std. Deviation [MPa]	Calculated Max Stress [MPa]	Probability	Sigma level	10^{-4} probability Stress [MPa]	Sigma level
Max	1	321	23	451	$4{\cdot}10^{-7}$	4.9	415	3.7
	2	324	23	453	$3{\cdot}10^{-7}$	5.2	414	3.7
	3	325	19	456	$2{\cdot}10^{-9}$	5.9	402	3.7
Nom	1	265	24	417	$6{\cdot}10^{-8}$	5.3	367	3.7
	2	267	23	410	$4{\cdot}10^{-8}$	5.4	362	3.7
	3	263	22	461	$8{\cdot}10^{-14}$	7.3	364	3.7
Min	1	227	25	427	$7{\cdot}10^{-10}$	6.1	334	3.7
	2	227	25	440	$1{\cdot}10^{-10}$	6.4	334	3.7
	3	226	24	433	$1{\cdot}10^{-10}$	6.3	335	3.7

The 10^{-4} probability level is equal to a load or event with a return period of 1/10000 years. The probability that the given load will not be exceeded during n- years is $(1\text{-}1/R_p)^n$, Where R_p is the return period. Therefore the probability that the design load is to be exceeded at least once during n-years is:

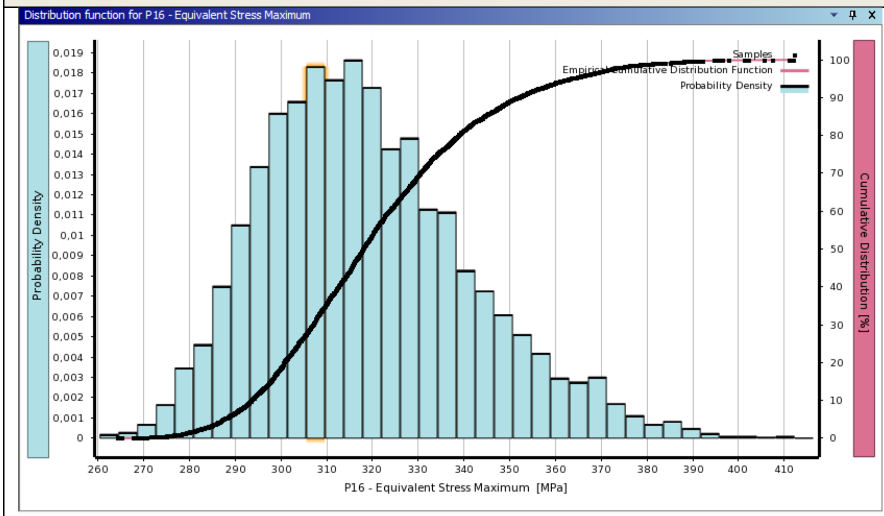
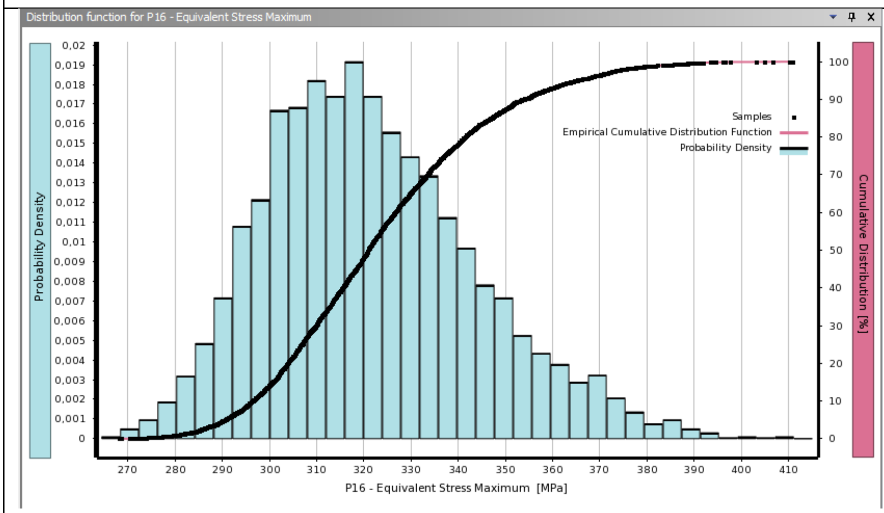
$$P_n = 1 - (1 - 1/R_p)^n \tag{7.1}$$

So given a design life of 25 years this equals:

$$P_n = 1 - \left(1 - \frac{1}{10000}\right)^{25} = 0.0025 = 0.25\% \tag{7.2}$$

The frequency distribution and the probability density function for the resulting stress levels for all spool configurations is shown in Table 7-2 to Table 7-4.The results are calculated by the “six sigma” tool in ANSYS design explorer. The software calculates cumulative distributions and probability density functions. One can observe that the stress distributions are almost identical and is positive skewed to the right. This means that the mass of the distribution is concentrated on the left indicating that the probability of high stress levels is low. Table 7-2 shows that the probability of reaching stresses greater than 390 MPa is less than 0.1% for the Max1 configuration. This indicates that the safety level for the spool design is higher than required by governing codes.

Table 7-2 von Mises stress distribution “Max” configuration

von Mises stress distribution “Max” configuration	
	**Max 1:** L_{spool}=30m Leg 1=*7267* mm Leg 2=3000 mm Leg 3=6510 mm Leg 4=27000 mm Leg 5=757 mm
	**Max 2:** Leg 1=*8117* mm Leg 2=3000 mm Leg 3=6510 mm Leg 4=27000 mm Leg 5=757 mm

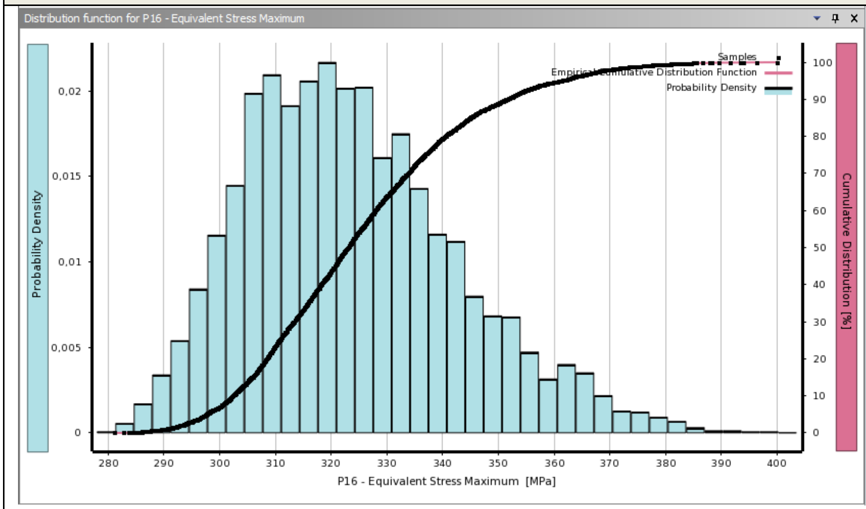
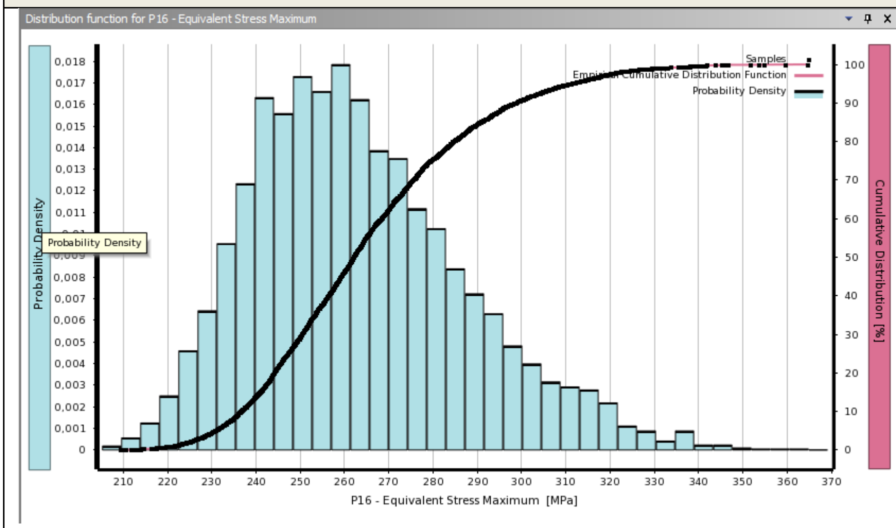
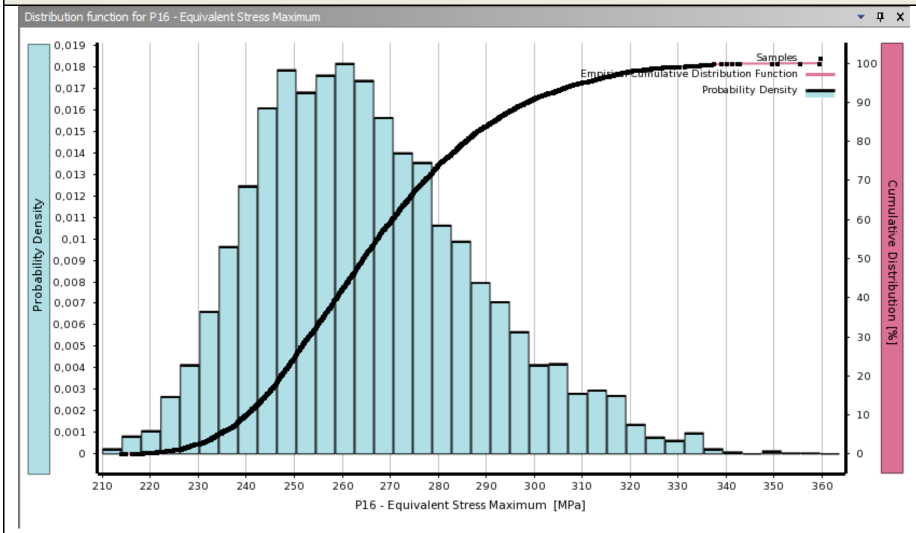
von Mises stress distribution “Max” configuration	
	**Max 3:** Leg 1=*8767* mm Leg 2=3000 mm Leg 3=6510 mm Leg 4=27000 mm Leg 5=757 mm

Table 7-3 von Mises stress distribution “Nom” configuration

von Mises stress distribution “Nom” configuration	
	**Nom 1:** L_{spool}=24m Leg 1=*7267* mm Leg 2=3000 mm Leg 3=6510 mm Leg 4=21000 mm Leg 5=757 mm

<table>
<tr><th colspan="2">von Mises stress distribution “Nom” configuration</th></tr>
<tr><td>


</td><td>

Nom2:

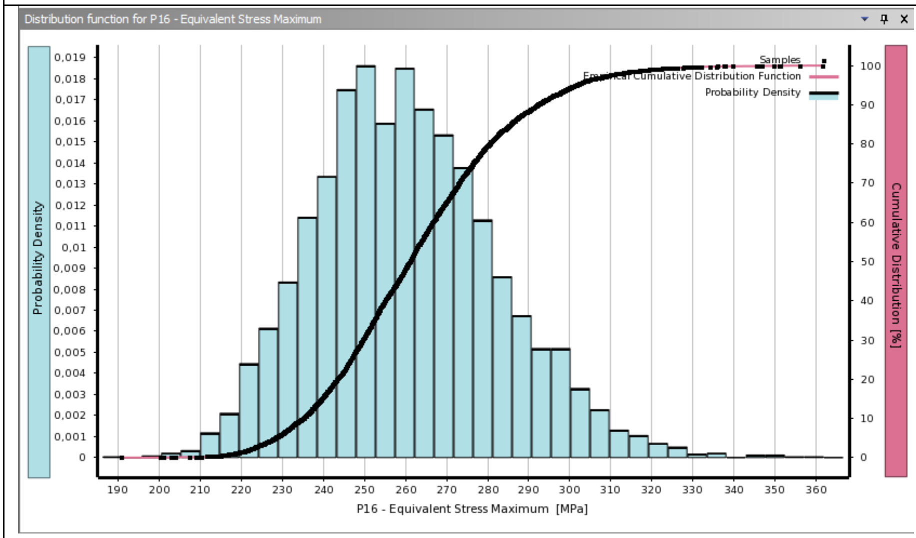
Leg 1=*8117* mm

Leg 2=3000 mm

Leg 3=6510 mm

Leg 4=21000 mm

Leg 5=757 mm

</td></tr>
<tr><td>


</td><td>

Nom3:

Leg 1=*8767* mm

Leg 2=3000 mm

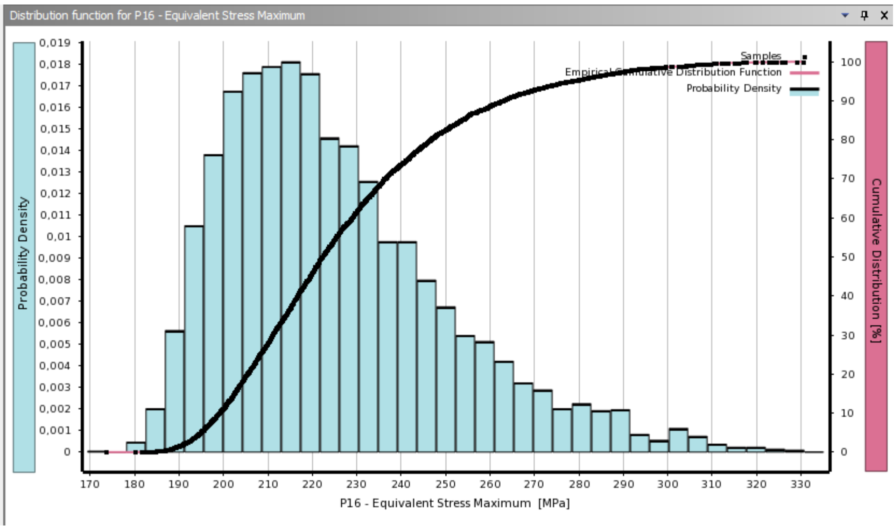
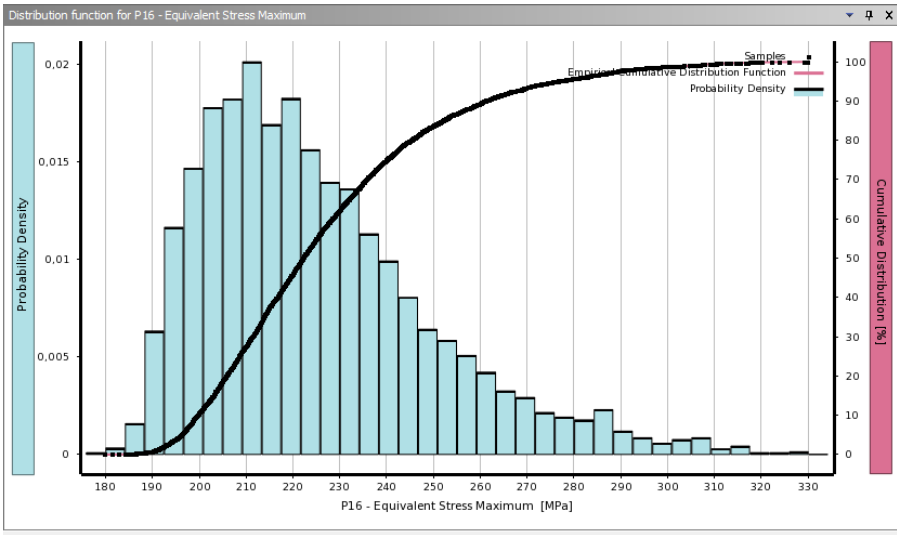
Leg 3=6510 mm

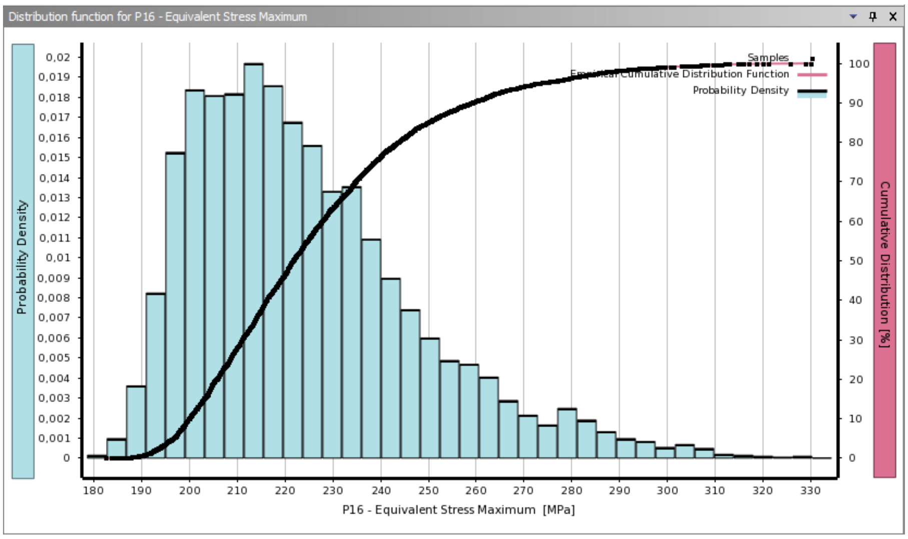
Leg 4=21000 mm

Leg 5=757 mm

</td></tr>
</table>

Table 7-4 von Mises stress distribution "Min" configuration

<table>
<tr><th colspan="2">von Mises stress distribution "Min" configuration</th></tr>
<tr><td>
</td><td>Min 1:
L_{spool}=18m
Leg 1=7267 mm
Leg 2=3000 mm
Leg 3=6510 mm
Leg 4=15000 mm
Leg 5=757 mm</td></tr>
<tr><td>
</td><td>Min 2:
Leg 1=8117 mm
Leg 2=3000 mm
Leg 3=6510 mm
Leg 4=15000 mm
Leg 5=757 mm</td></tr>
</table>

<table>
<tr><th colspan="2">von Mises stress distribution “Min” configuration</th></tr>
<tr><td>


</td><td>
Min 3:

Leg 1=8767 mm

Leg 2=3000 mm

Leg 3=6510 mm

Leg 4=15000 mm

Leg 5=757 mm
</td></tr>
</table>

7.2 Reaction Forces and Bending Moments

The maximum and minimum values for the reaction forces can be extracted by use of the max/min tool. The values are shown in Figure 7-1 , Figure 7-2 and Figure 7-3 , the configurations are numbered from 1 to 9 representing 9 configurations (Max 1 to 3, Nom 1 to 3 and Min 1 to 3). Values for bending moments and reaction forces can also be extracted by use of the “Six Sigma” probability table. Here the values can be extracted from the Percentile-Quantile table for a specific parameter see Table 7-5 and Table 7-6 4-21 for Max1 and Min3 configuration. The reaction forces are of interest for the installation process and to the loads on the clamp connectors.

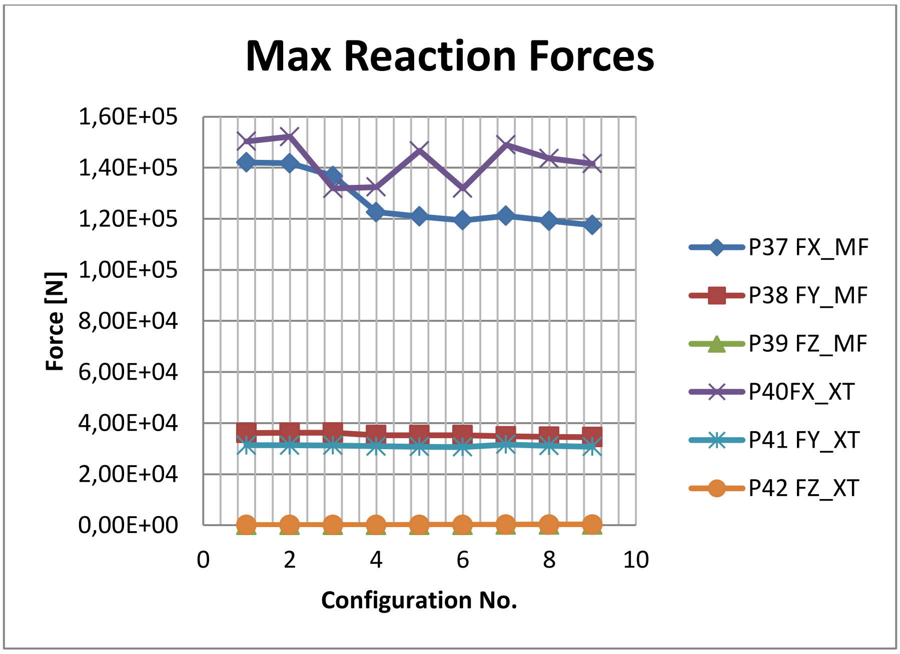


Figure 7-1 Max Reaction Forces (Abs values)

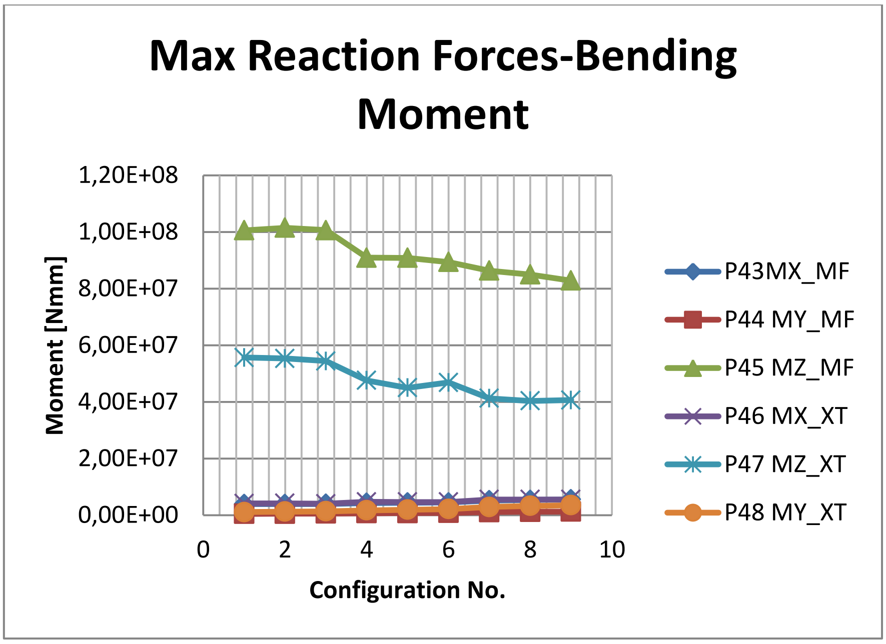


Figure 7-2 Max Reaction Moments (Abs. values)

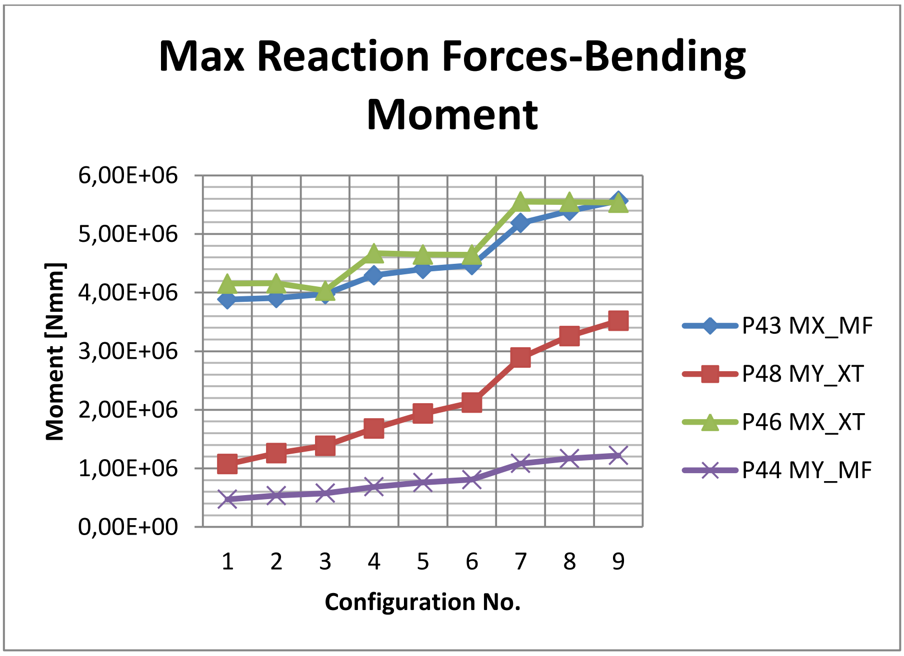


Figure 7-3 Max Reaction Moments MY and MX (Abs. values)

As seen from Figure 7-2 and Figure 7-3 , the governing bending moments is the MZ bending moments which naturally increases with the length of the spool, (configuration Max). The other bending moments increases when the spool becomes shorter; this is due to and increased overall structural stiffness for the spool. The mean reaction force and moments are shown in Figure 7-4and Figure 7-5.

Table 7-5 Reaction Forces “Max 1”configuration 10^{-4} probability

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]

35,9	-93	0,04	3,3	-0,4	87,8
X-tree Side					
FY [kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
31,0	143	-0,04	-3,8	0,9	-51,0

Table 7-6 Reaction Forces "Min 3"configuration 10^{-4} probability

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
33,9	91,8	0,023	5,0	1,1	68,6
X-tree Side					
FY [kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
30,0	-94,0	-0,023	4,5	3,0	11,0

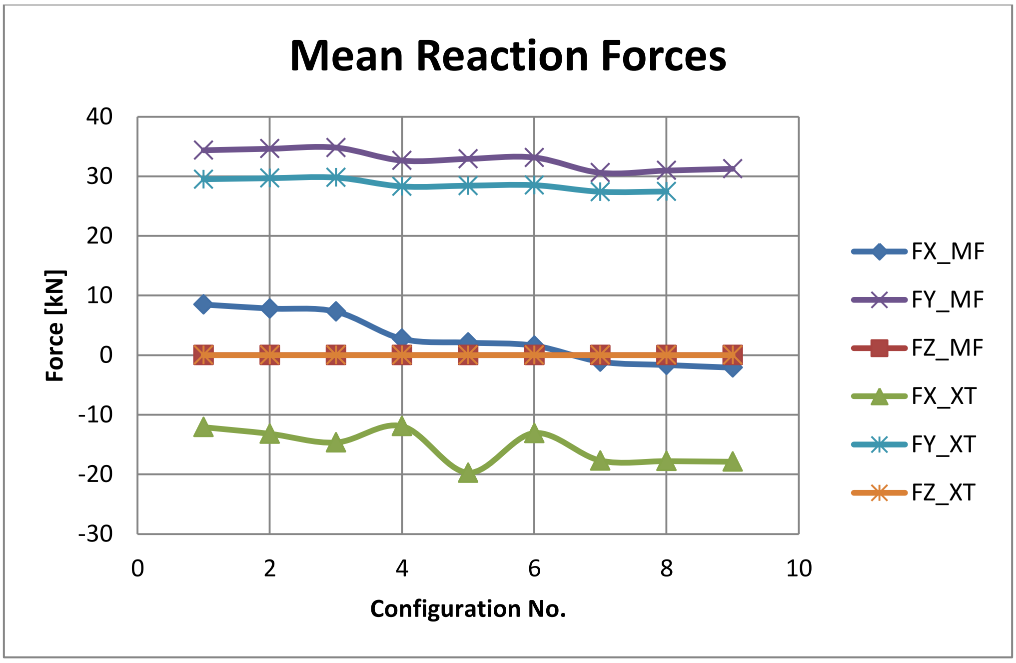


Figure 7-4 Mean reaction forces

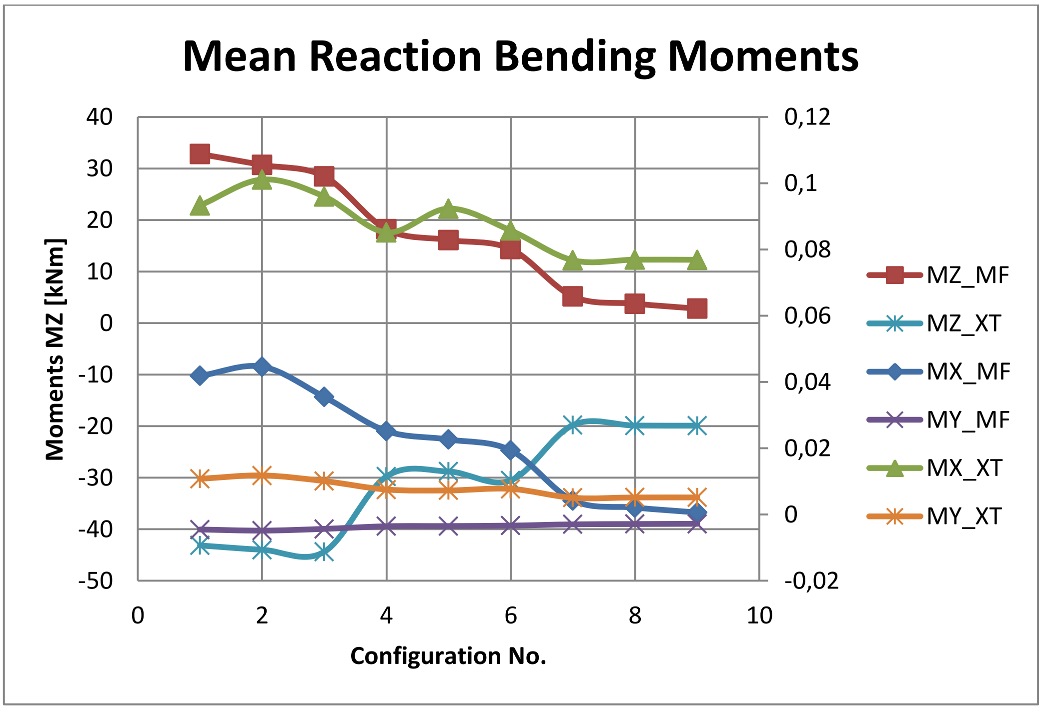


Figure 7-5 Mean reaction bending moment

7.3 Stress Response and Sensitivity

The sensitivity to stress increase imposed by the deformations is shown by the response plots in Figure 7-7 to Figure 7-10 for the “Max 1” configuration. One can observe how the response of stress is of a non-linear character towards these imposed deflections and rotations. Hence a small increase of these values gives a large stress increase.

It is important to remember that a response surface is a curve fit based on a set of data. If not enough design points were solved or if the response surface algorithm is not appropriate then the response surface will not be accurate. One can refine the response surface type to create a more accurate response surface based upon more design points or type of curve fit algorithm. The technique used here is called the Kriging algorithm see Figure 7-6. This is a multidimensional interpolation combining a polynomial model similar to the one of the standard response surface, which provides a “global” model of the design space, plus local deviations determined so that the Kriging model interpolates the DOE points. Output=f(inputs) + Z(inputs) ,where f is a second order polynomial (which dictates the “global” behavior of the model) and Z a perturbation term (which dictates the “local” behavior of the model)

Since Kriging fits the response surface through all design points the “Goodness of fit metrics” will always be good

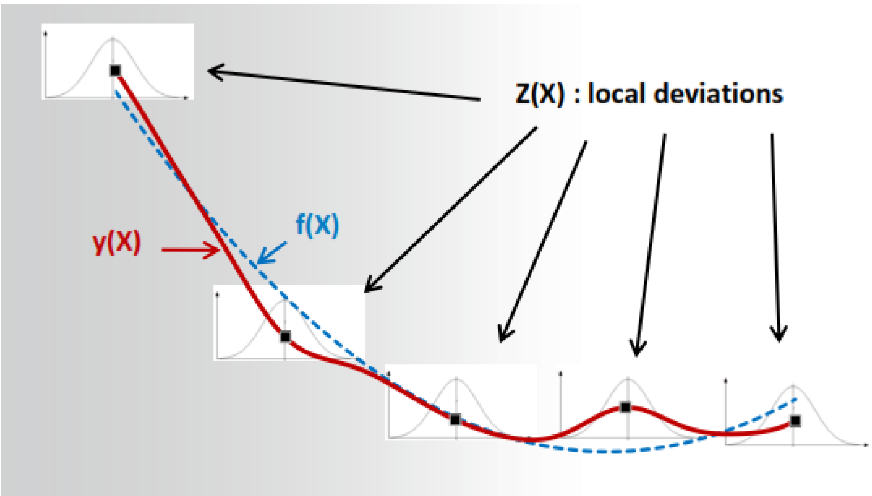


Figure 7-6 Kriging Algorithm curve fit (Source ANSYS lectures)

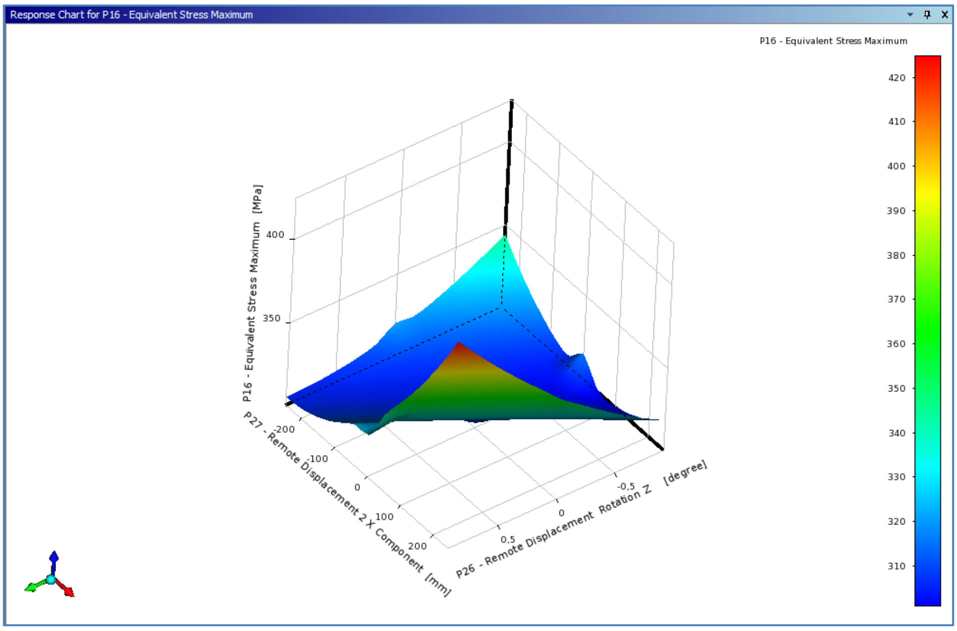


Figure 7-7 Stress Response "Max1" Configuration Dx_{Tree} versus $Rz_{Manifold}$

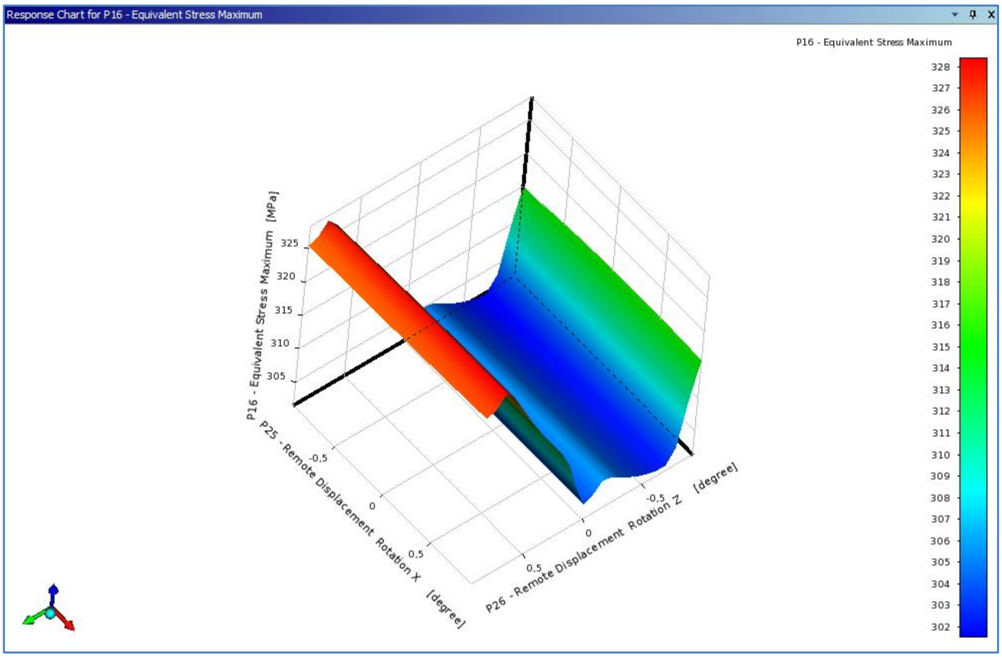


Figure 7-8 Stress response "Max 1" configuration Rotations Rx and Rz Manifold end

Figure 7-7 and Figure 7-8 shows the equivalent stress response for the imposed rotations and deformations at the spool ends. Figure 7-7 shows a non-linear response between positive displacements and rotations but less sensitive to negative displacements and positive rotations Figure 7-8 has a parabolic non-linear shape for two axis rotation.

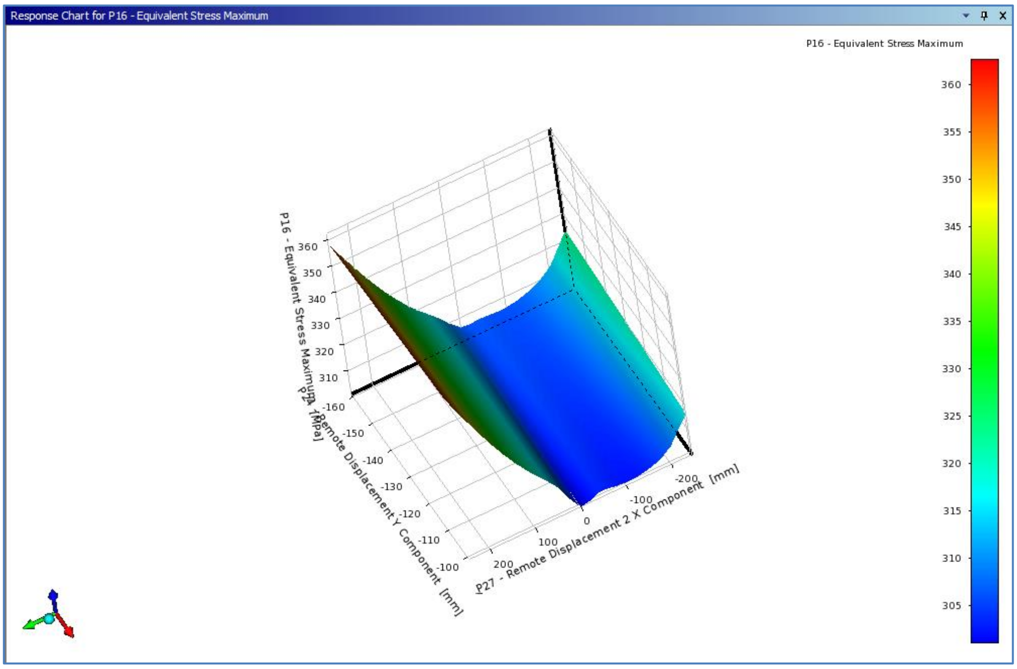


Figure 7-9 Stress response "Max 1" configuration Displacements Dx_{Tree} and $Dy_{Manifold}$

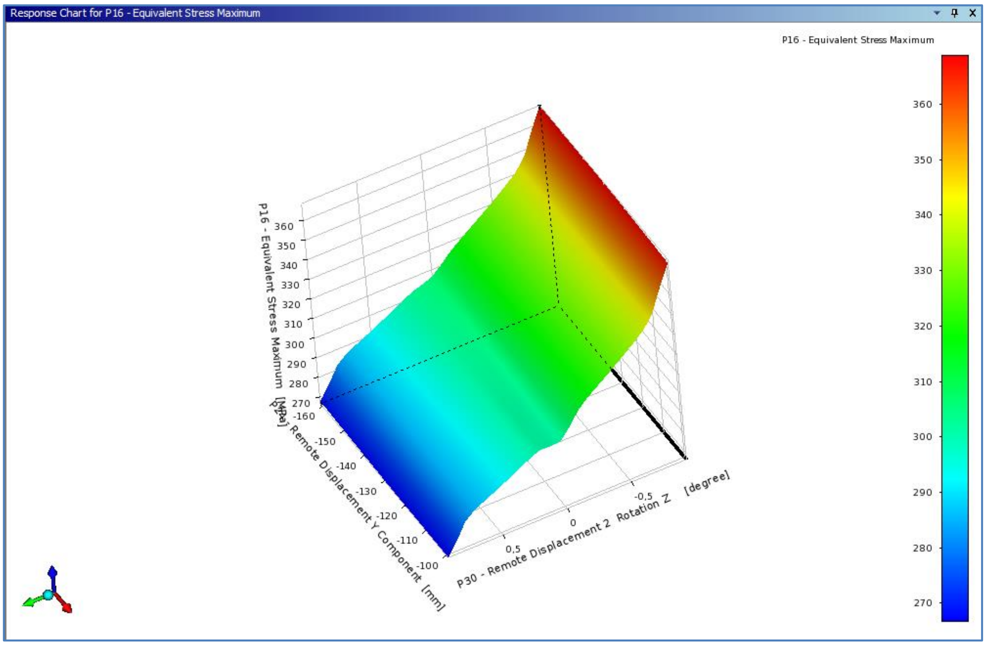


Figure 7-10 Stress response "Max 1" configuration Displacements Rz_{Tree} and $Dy_{Manifold}$

Figure 7-9 shows a non-linear response between the vertical and horizontal displacements at each spool end. Figure 7-10 shows an almost linear response between the rotation among Z- axis at XT end and the vertical displacement at the manifold end. The Software is provided with a tool for measuring the goodness of the curve fit as seen in Figure 7-11 , most of the parameters lies within a straight line which is a measure of good fit.

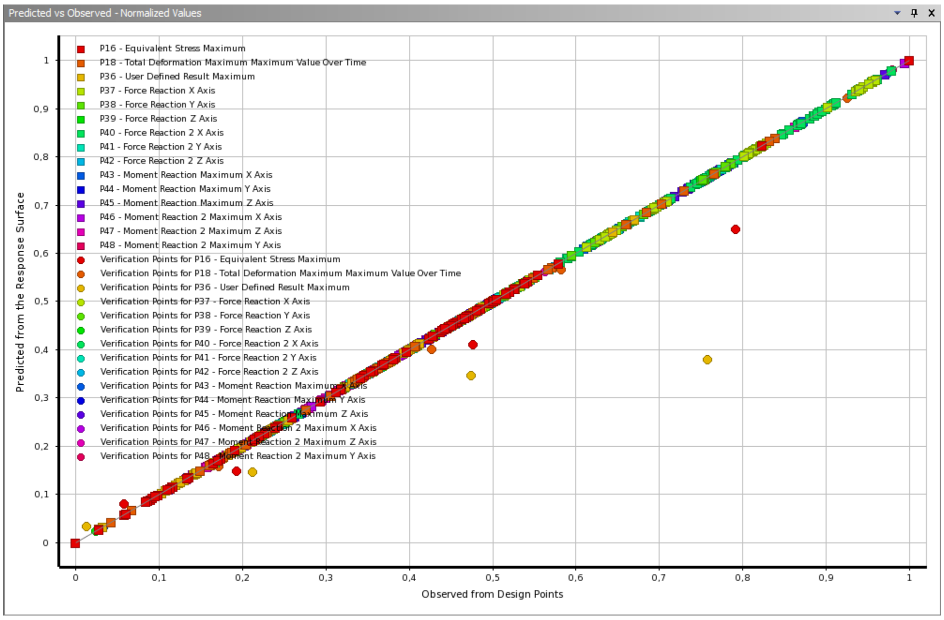


Figure 7-11 Goodness of fit for response algorithm

The sensitivity between the input variation parameter is shown Figure 7-12 . It can be seen that the imposed rotations contributes to high stress level.

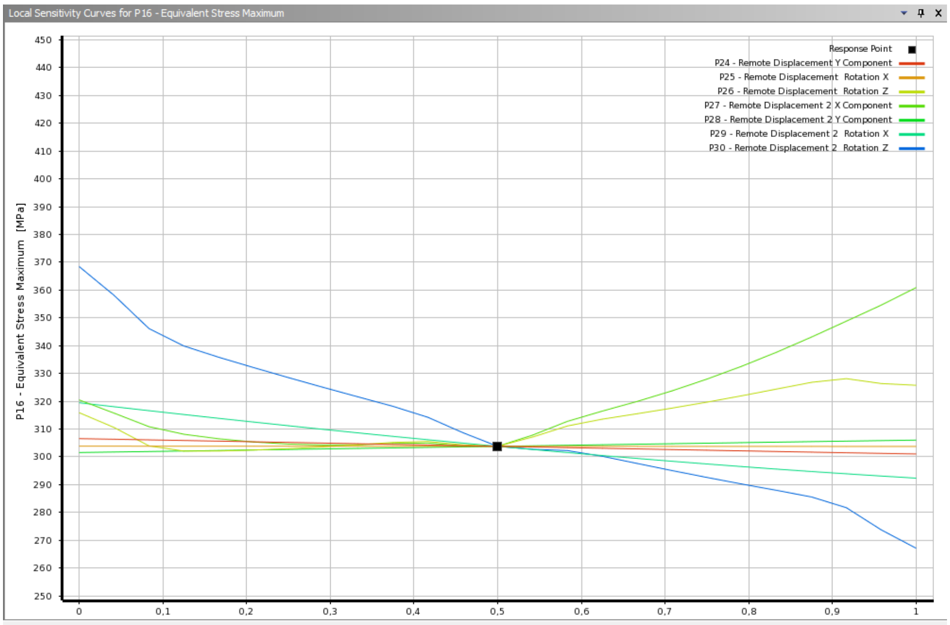


Figure 7-12 von Mises Stress Sensitivity "Max 1" Configuration

The results from the sensitivity analysis show that the spool design is sensitive to variation in displacements and rotations. This sensitivity must be taken into consideration when assessing the total safety levels for spool design. Especially angular deviation will require more attention during fabrication, methodology and quality control for the spools. If a Tie-in connection with large angular deviations outside the specification range for the tolerances is made up, then the strains in the spools is likely to reach levels beyond the yield point of the material and a redistribution of the stresses in the spool is likely to occur which in most cases is considered to be OK. But this may have an effect on the limits for number of start up and shut down cycles (hot and cold) for the spool as accumulation of plastic strains can occur. This must also be taken into considerations.

7.4 Optimal Spool Configuration

Table 7-7 shows a set of worst case rotations and displacements equivalent for 10^{-4} probability stress level that has been checked with the response surface optimization tool. A value for stresses between 380MPa and the design limit of 405 MPa, as given by the code was set as the target range. (A tolerance range for the stress target was used in order to allow for computation for more possible geometrical candidates). The software was also given a constraint to calculate candidates for geometrical shapes that would minimizing the bending moments for the 30m long spool. The analysis shows that in order to be within the stress limit for the 30meter long spool the geometry has to change to some of the candidates as suggested by the software algorithm.

Table 7-7 Imposed spool end deformations (10^{-4} - Extremes)

Type	Spool length [m]	Dy_{MF} [mm]	Rx_{MF} [Deg]	RZ_{MF} [Deg]	Dx_{XT} [mm]	Dy_{XT} [mm]	Rx_{XT} [Deg]	Rz_{XT} [Deg]
Max 1	30	-160	0.9	0.9	248	-30	0.9	0.9

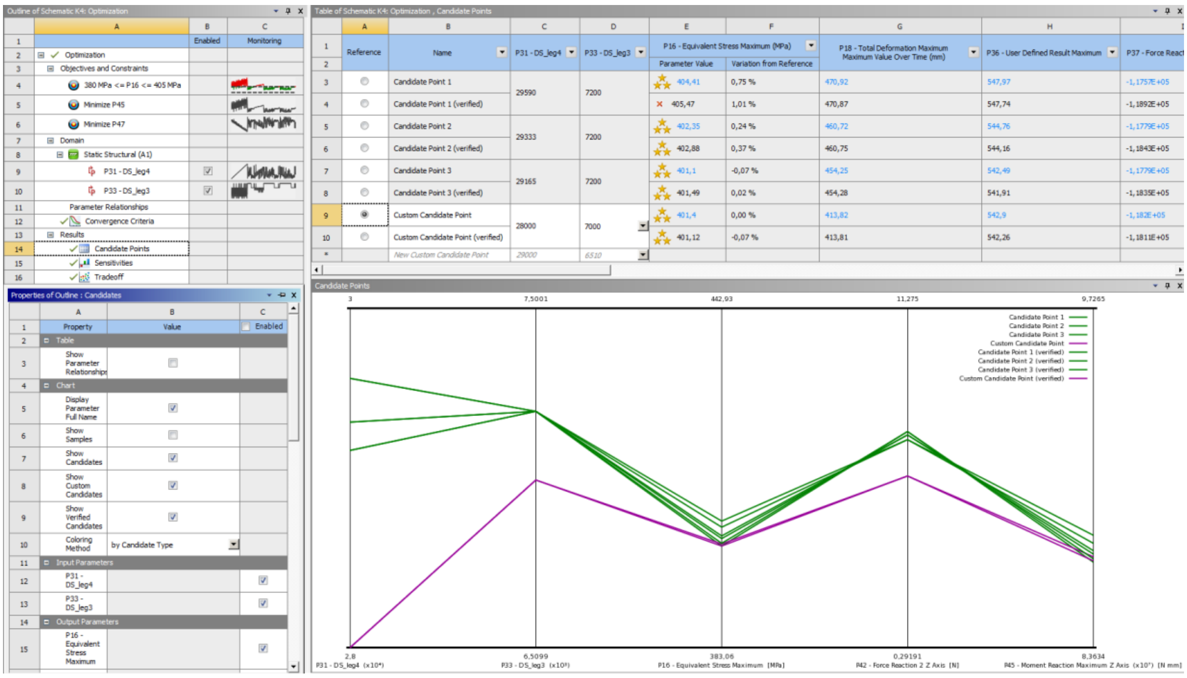


Figure 7-13 ANSYS Optimisation Results and candidates for “max 1” configuration

Figure 7-13 Show’s the possible candidates as calculated and verified by the optimisation tool. The output is supplied with a samples chart, where each sample is displayed as a group of line curves where each point is the value of one input or output parameter. This is useful to find custom candidates. The optimisation method used is called MOGA or (Multi Objective Genetic Algorithm), which is an iterative multi objective algorithm. Benefits of method are:

- Helps identify global and local minima
- Provides several candidates in different regions
- Accurate solution
- Can handle multiple goals

The drawback of this method is that it might concentrate on a single region in the design space. Based upon the suggested candidates it seems that the custom candidate point is a good choice with a stress level of 401 MPa. The optimal dimensions for "Max 1" are given in Table 7-8

Table 7-8 Optimal Geometry for 30m long spool

Type	Spool length [m]	Leg 1 [mm]	Leg 2 [mm]	Leg 3 [mm]	Leg 4 [mm]	Leg 5 [mm]	Height Difference [mm]
Max 1	30	**7227**	**2000**	**7000**	**28000**	757	**-530**

7.5 Analysis Results Operational

Based upon the results from ANSYS Design Explorer analysis the "Max 1" configuration is checked as this configuration has the highest stress level. The spool is checked with max corroded wall thickness (3mm) and the imposed displacement for 10^{-4} probability stress level. Maximum drag force from current is applied.
The mesh used here is denser as shown in Figure 7-14. Note that ANSYS displays the shell as a thick solid wall. The user has the option to turn of this graphical feature and only show shell elements as thin elements. The end lid is solid elements.

The ANSYS stress plot shown in Figure 7-15 reports a high stress value of 501 MPa. By a close Inspection of the results shown in Figure 7-17 the high stress is caused by a peak value located at the contact area between the flat end lid and pipe wall of the spool at the Manifold end (MF). The end lid does not exist and is used as FEA modelling technique in order to transfer endcap forces to the boundary condition of the spool. In reality this pipe end is welded to a "Hub". Hence the peak stress between the end lid and pipe wall is ignored. The deformation plot in Figure 7-16 shows that the spool deflects or sags at the middle. This is mainly caused by the spool self-weight and is contributing to high stresses and moments in the spool.

The membrane and bending stresses in the shell in the pipe is extracted at a distance away from the structural discontinuity. Guidelines on how to extract stresses away from discontinuity and how to classify stresses, so called stress linearization in a FEA model can be found in the ASME BPVC Ref /16/ Chapters 5. Design by Analysis. A case study where this technique is discussed and analysed can be found in the INAC conference paper /18/.
This technique is very useful for solid elements and through wall stresses.
According to this code a stress is classified as local if the local stress of 1.1S does not extend in the meridian direction with a distance greater than $\sqrt{Rt}$, Ref /16/ Section 5.2.2.2. Here *S* is the allowable stress for the material at design temperature and *R* is the mean radius, *t* is the wall thickness.

Distance away from structural discontinuity can be assessed by use of equation (7.3) /18/.

$$d \leq 2 \cdot \sqrt{R \cdot t} \tag{7.3}$$

This becomes:

$$d \leq 2 \cdot \sqrt{(0.5 \cdot (168.3 - 15.3)) \cdot 15.3} = 68mm \tag{7.4}$$

Region of local stress then becomes $\sqrt{Rt} = 34mm$
The average values from the shell element Top/Bottom, across the section at a distance *d* from discontinuity is used for comparison towards the code stress limits see Figure 7-20.

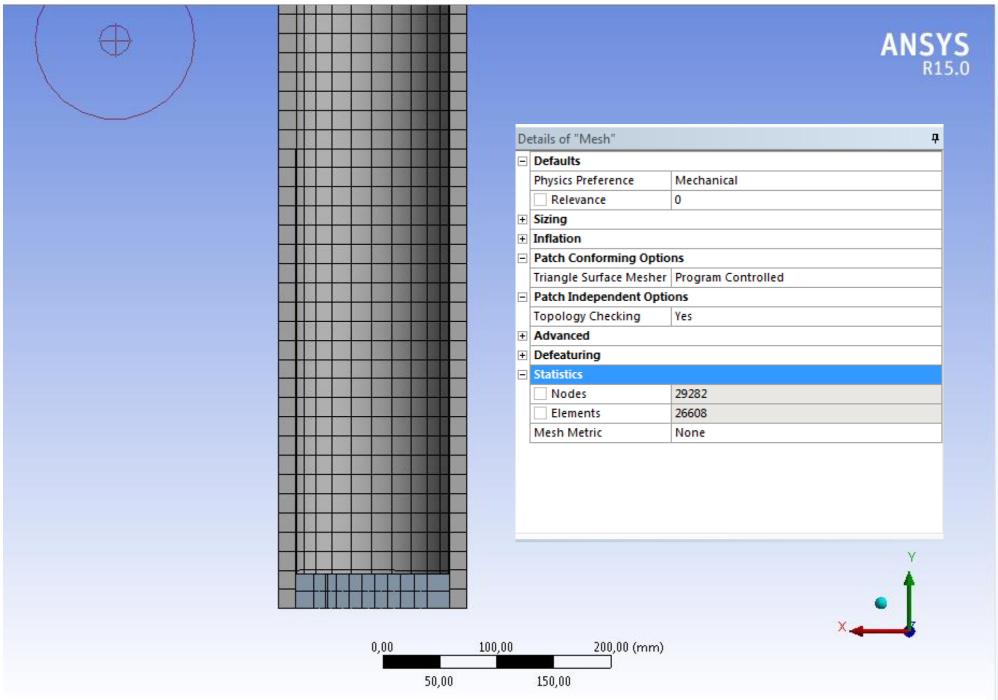


Figure 7-14 Details of Mesh FE model

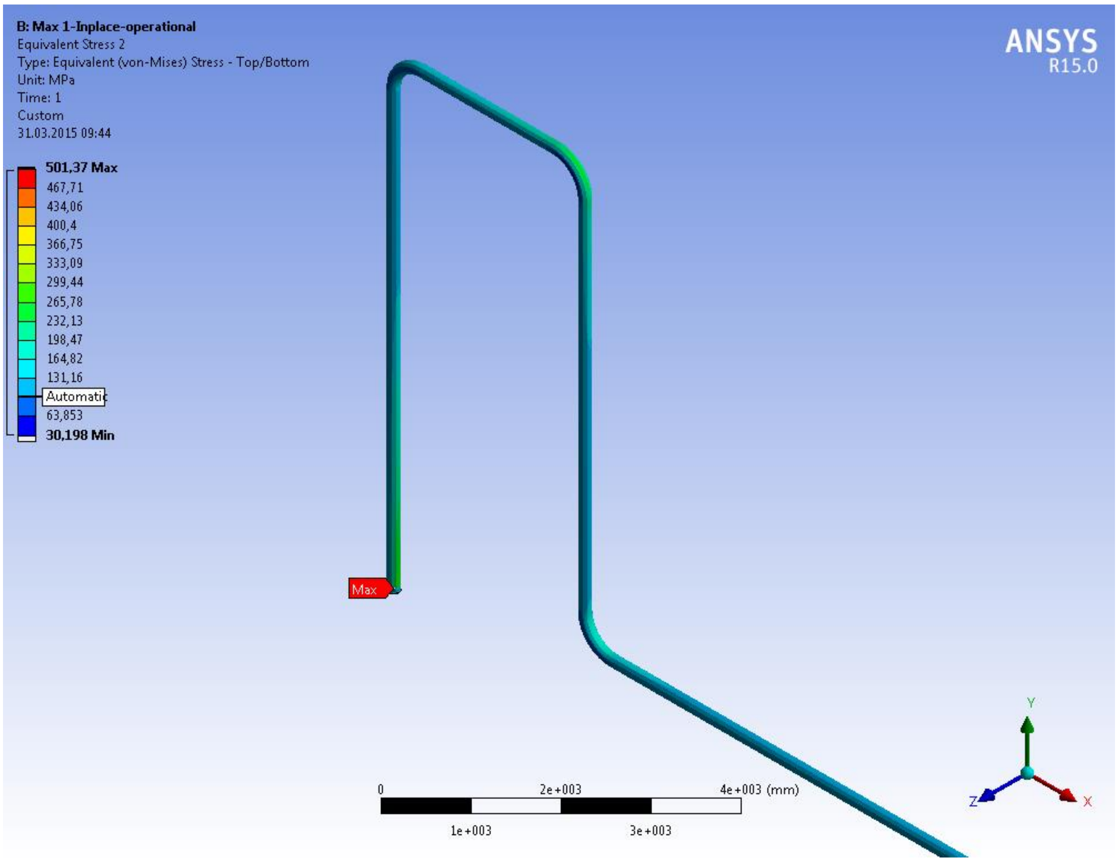


Figure 7-15 Max von Mises stress-operational

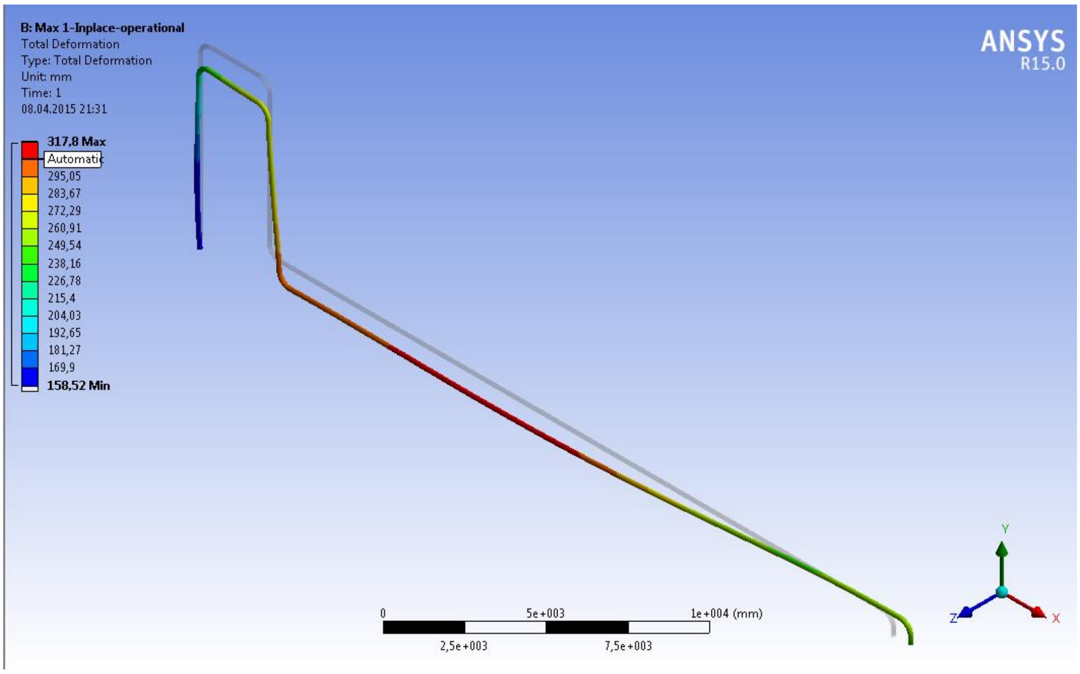


Figure 7-16 Total deformation spool

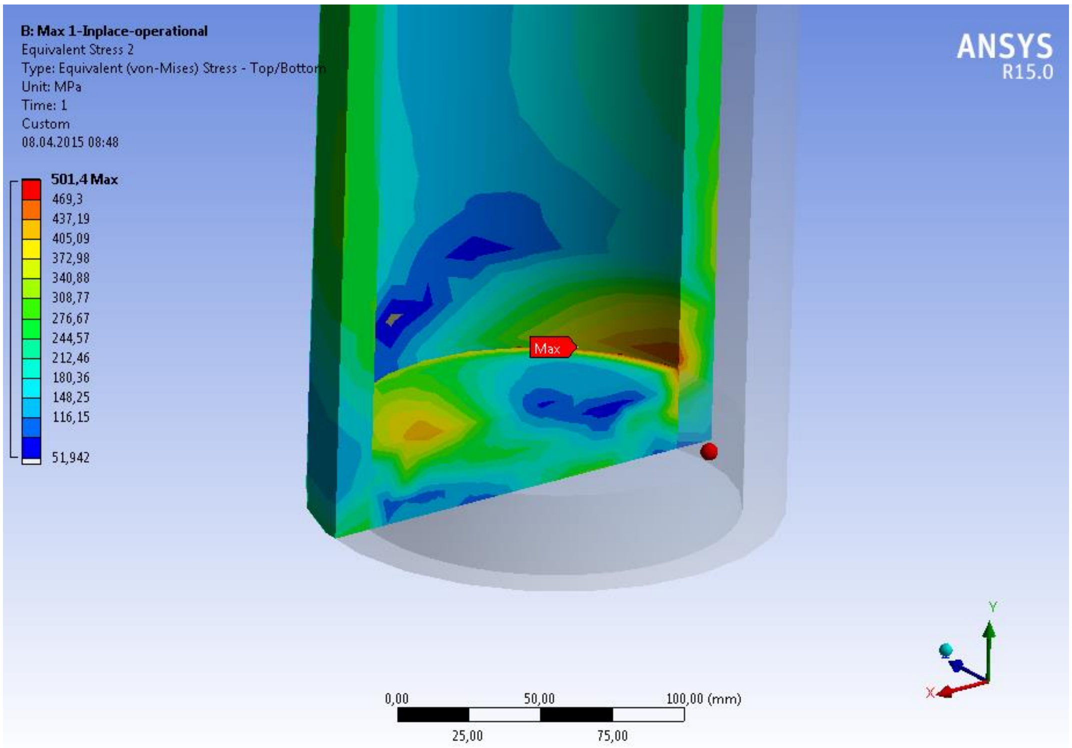


Figure 7-17 location of max peak stress operational at MF end

The stress utilisations (UF) towards the stress limits as described in chapter 5.9 is tabulated in Table 7-9. The ASME B31.8 code check is using formulas with bending moments, stress intensification factors (SIF) and then calculates the stresses based upon area and section modulus of the pipe. For comparison towards this limit the stresses tabulated in Table 7-9 is interpolated at cross sections by use of ANSYS surface tool.

Table 7-9 Max Spool Stresses -operational

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF	Ref.
Manifold end	313	405	0.77	364	360	**1.01**	Figure 7-19 Figure 7-20
Bend between leg2 /leg3	315	405	0.78	300	360	0.83	Figure 7-22 Figure 7-23

Table 7-10 Max Spool Reaction Forces -Operational

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
18.3	33.7	-0.11	-3.8	1.6	64
X-tree Side					
1.2	-38	0.8	-0,18	-4.6	-34

The code hoop stress criteria, is given by an analytical formula (5.1) and (5.2) , and is calculated in Appendix 3. Stress plots of longitudinal and von Mises stress are shown in Figure 7-18 to Figure 7-23.

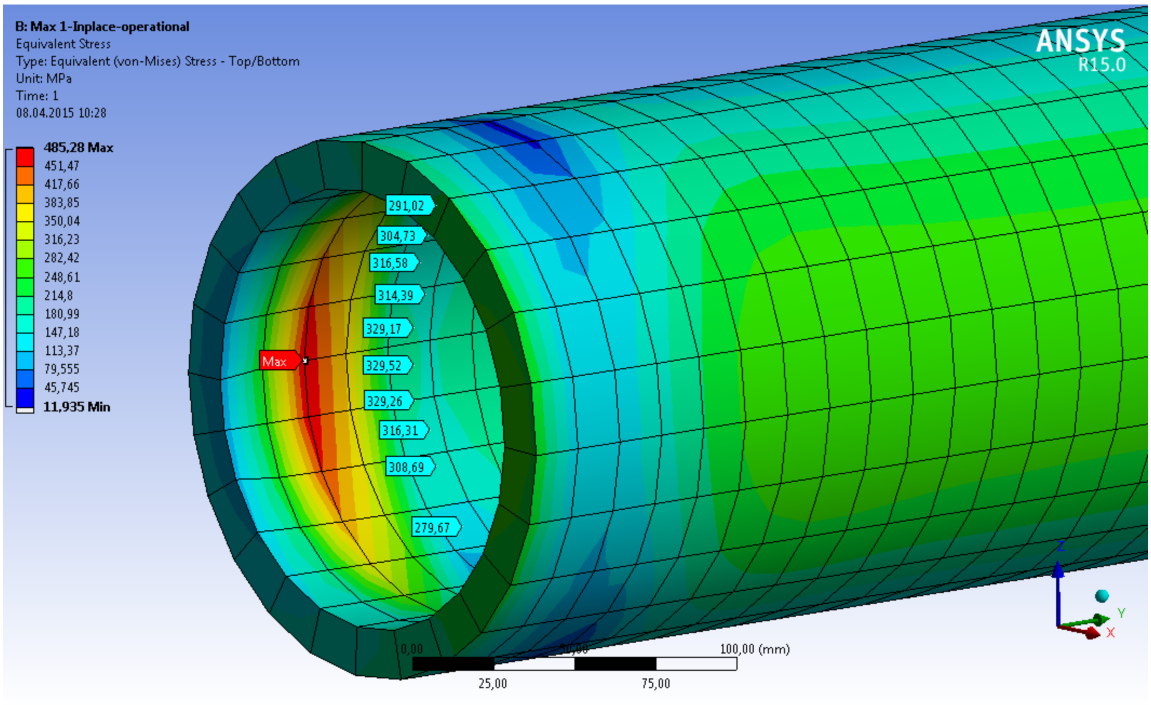


Figure 7-18 Max von Mises stress operational at MF end

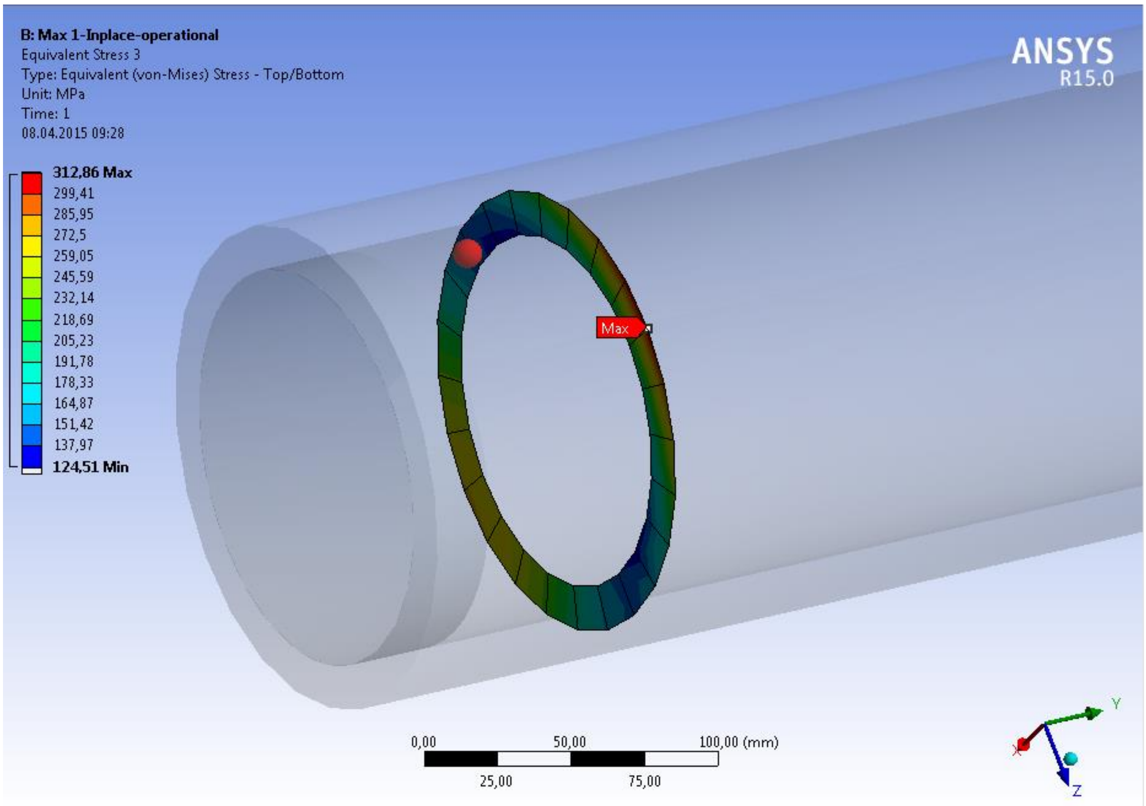


Figure 7-19 Cross sectional von Mises stress operational at MF end

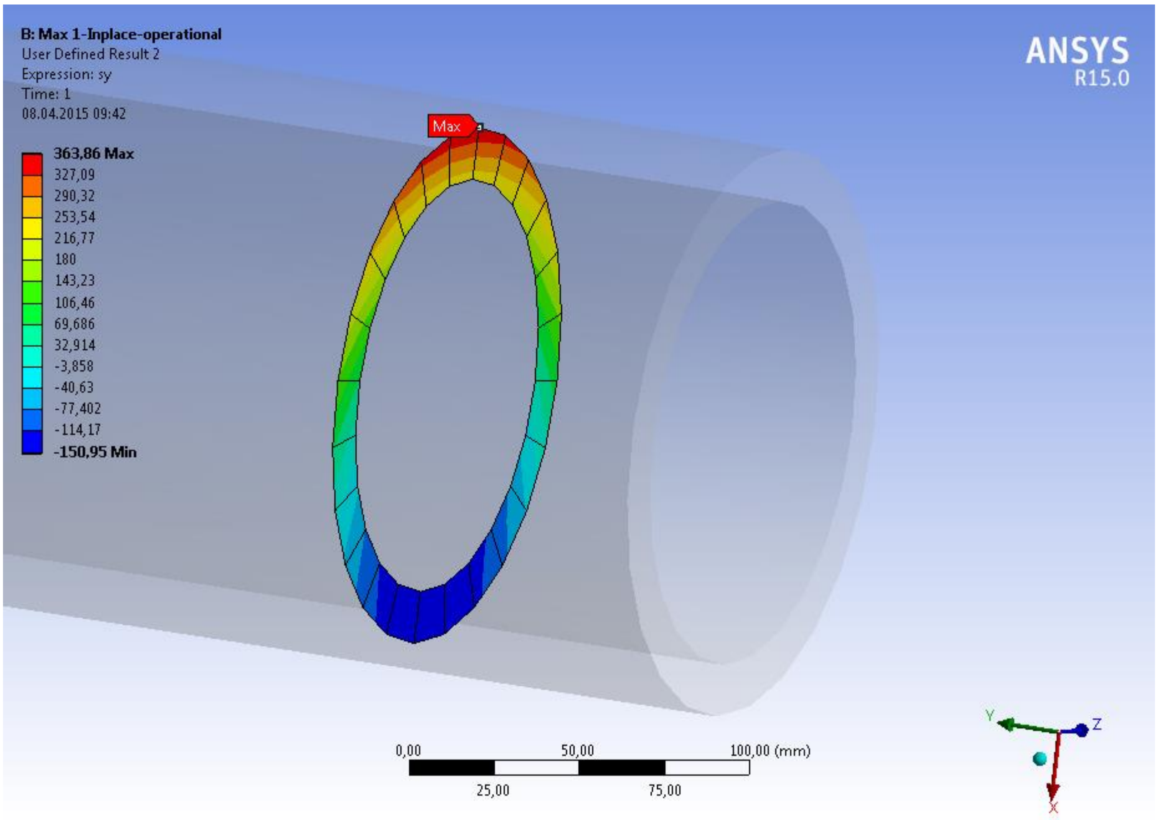


Figure 7-20 Cross sectional max longitudinal stress in pipe operational at MF-end

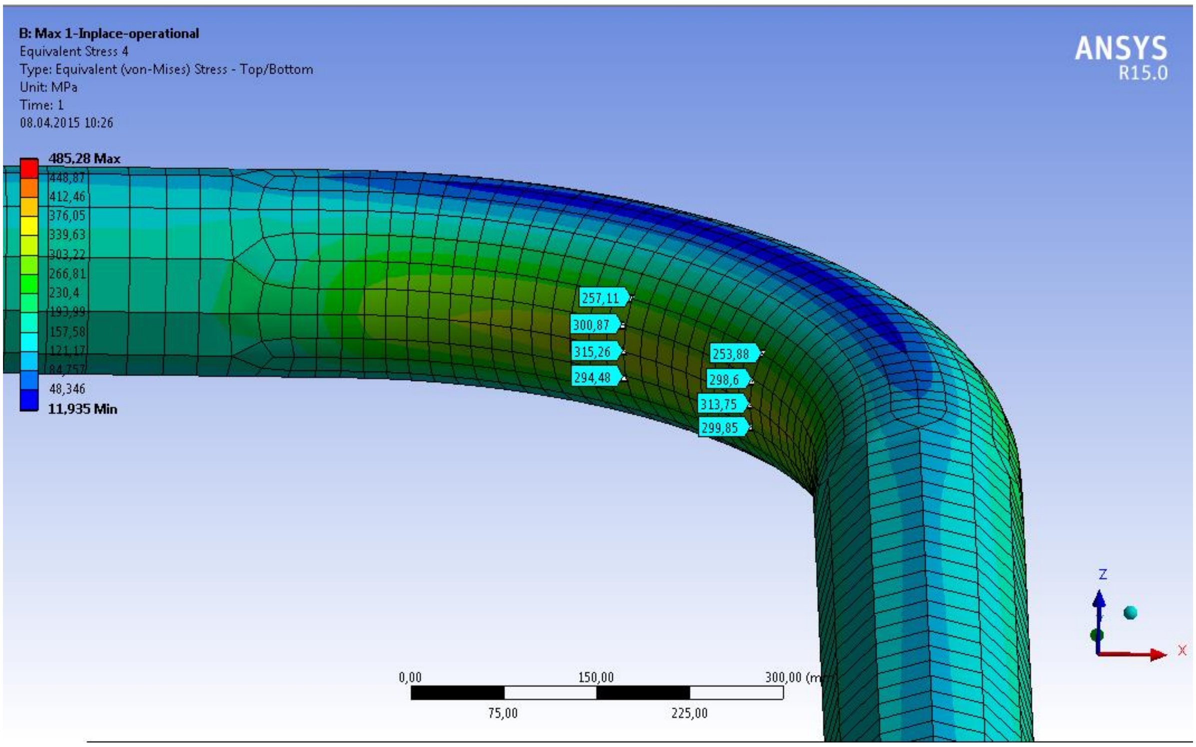


Figure 7-21 Von Mises stress operational at intrados of bend

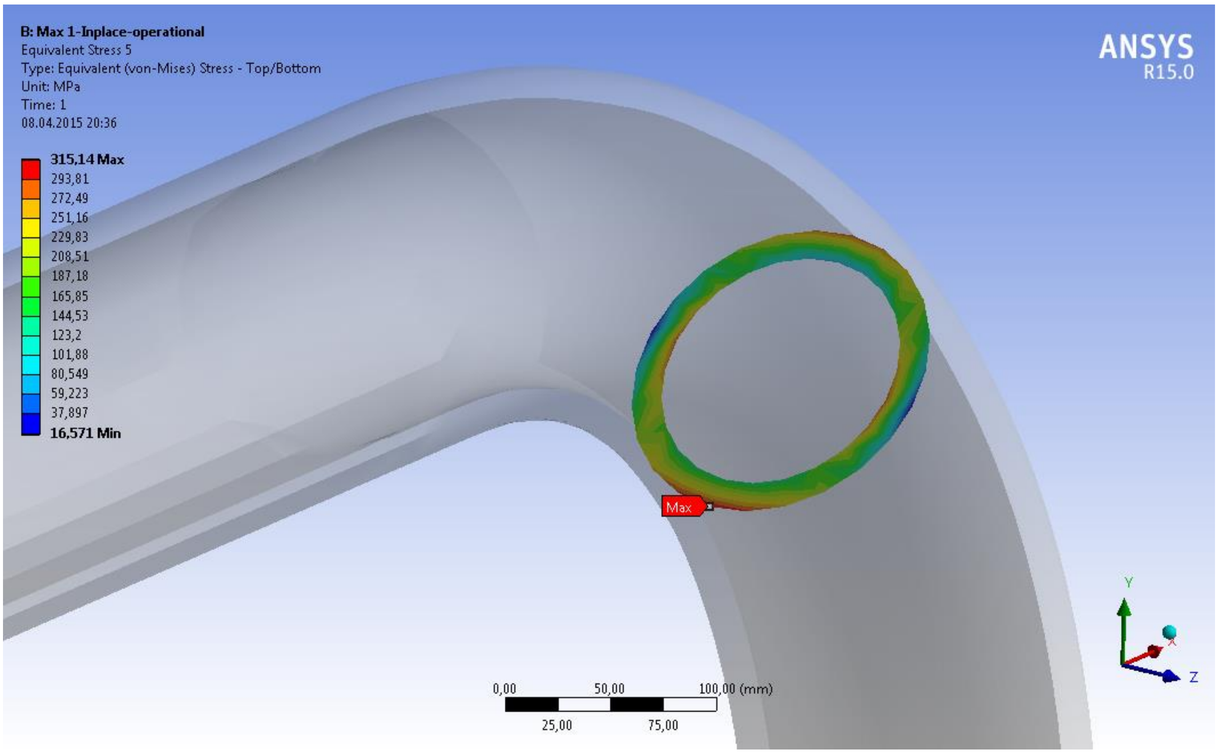


Figure 7-22 Cross sectional von Mises stress at bend

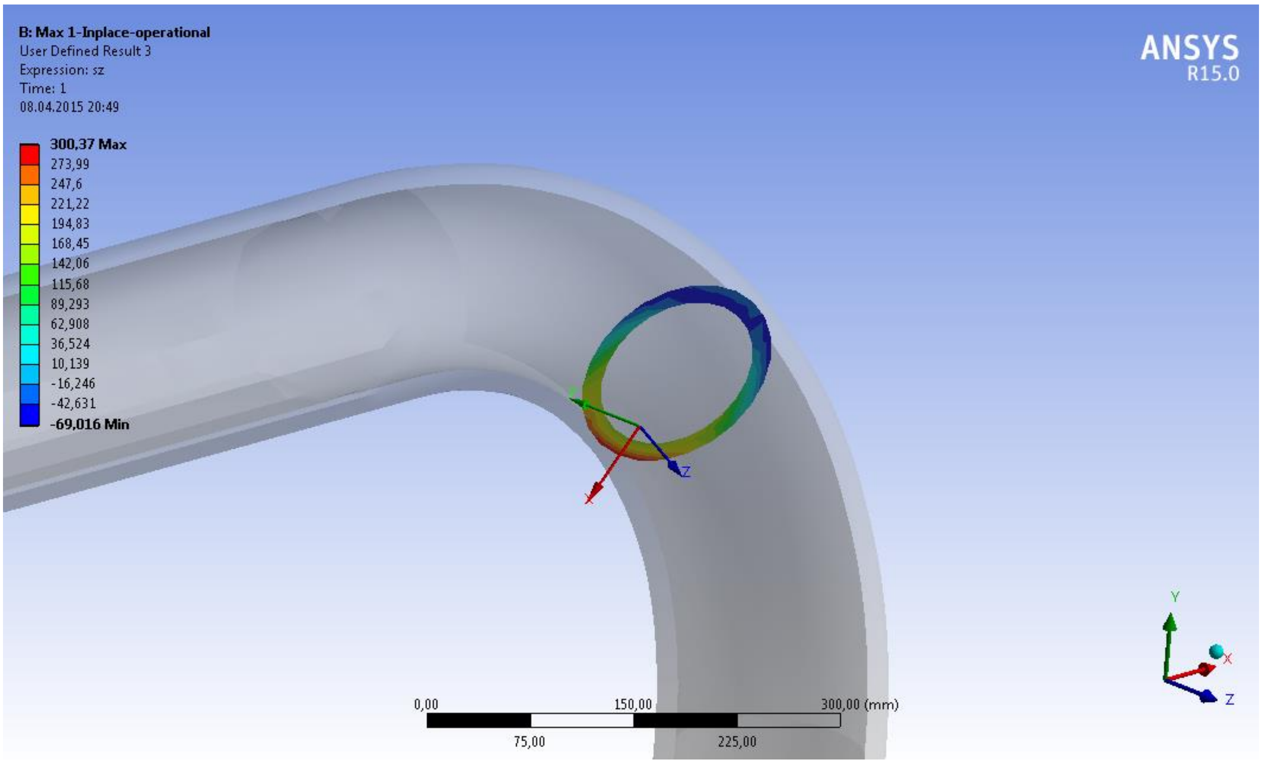


Figure 7-23 Cross sectional longitudinal stress at bend

7.6 Analysis Result FAT and Offshore Hydro testing

The result from the hydro testing of the spool on land and subsea is shown in Table 7-11. As seen from the stress plots the location of max stress is located at the XT end of the spool for the test load case.

Table 7-11 Max Spool Stresses -FAT/Subsea test

Location	Combined stress S_C (von Mises) [MPa]	Stress limit [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit [MPa]	UF	Ref.
XT-end Subsea Test	306	432	0.70	302	432	0.70	Figure 7-26 Figure 7-27
XT-end FAT	378	432	0.86	373	432	0.86	Figure 7-30 Figure 7-31

Table 7-12 Reaction forces Subsea Test

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
20.2	-6.0	-1.1	-3.9	1.5	-47.9
X-tree Side					
FY [kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
14.8	6.0	-0.8	-0.2	-4.7	-51.2

Table 7-13 Reaction forces FAT Test

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
27.7	-6.2	0	0.01	0.0	59.5
X-tree Side					
FY [kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
19.7	6.2	0	0.01	0.0	-63.1

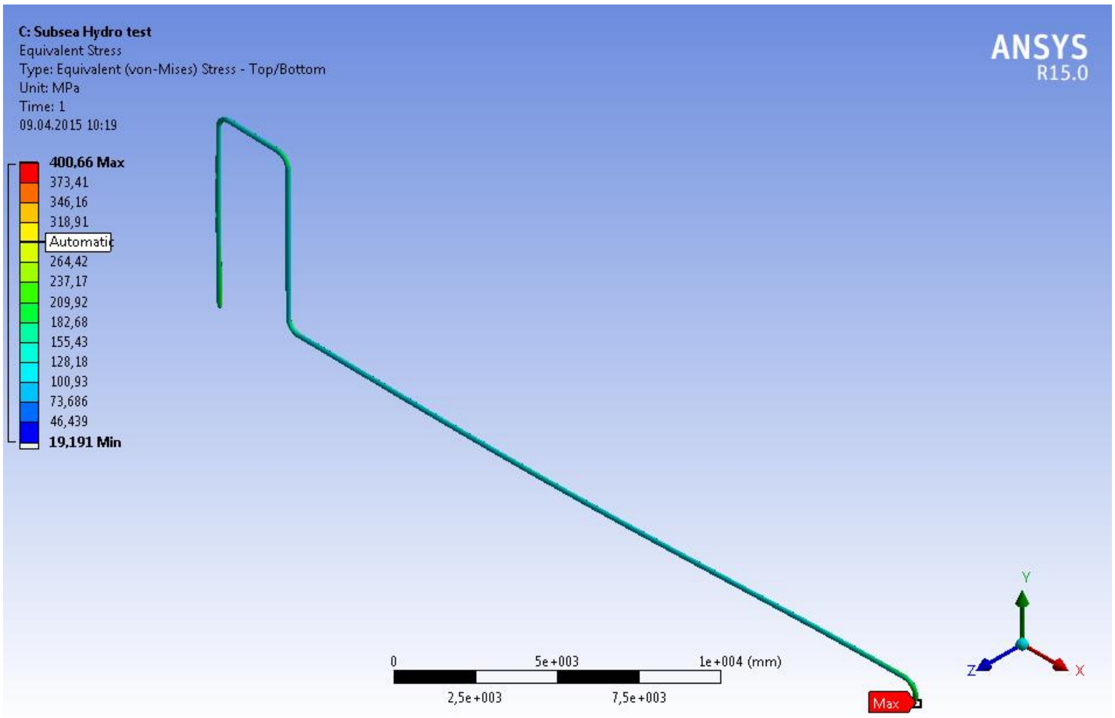


Figure 7-24 Max von Mises stress - Subsea Hydro Test

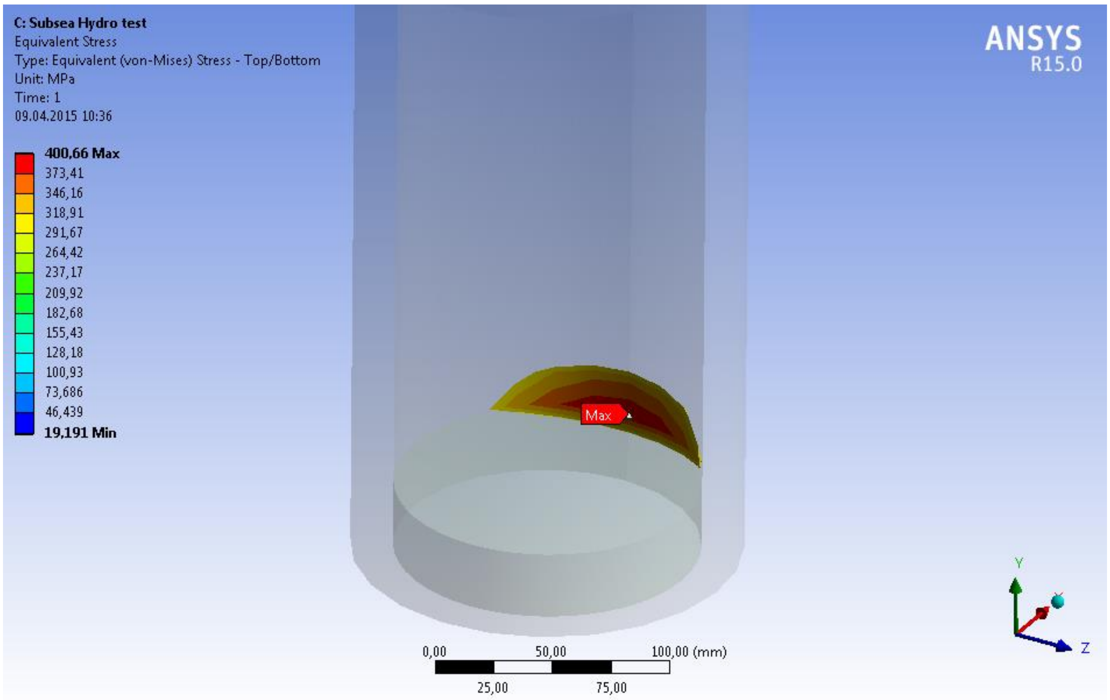


Figure 7-25 Max stress location- Subsea Hydro test

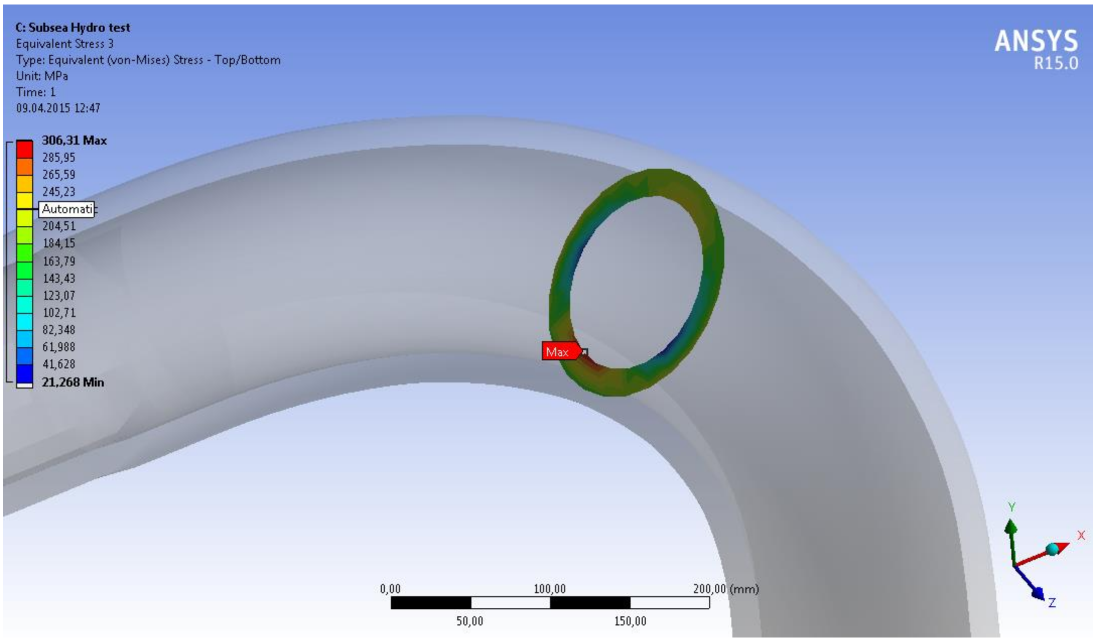


Figure 7-26 Cross sectional von Mises stress-Subsea Hydro Test at bend XT- end

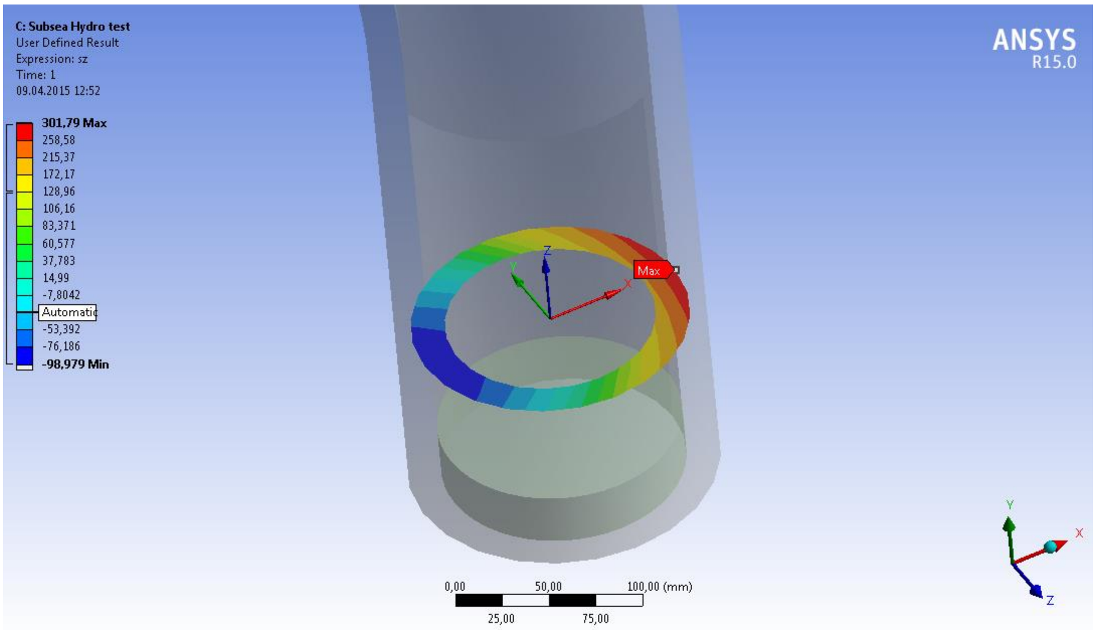


Figure 7-27 Cross sectional longitudinal stress-Subsea Hydro Test at XT- end

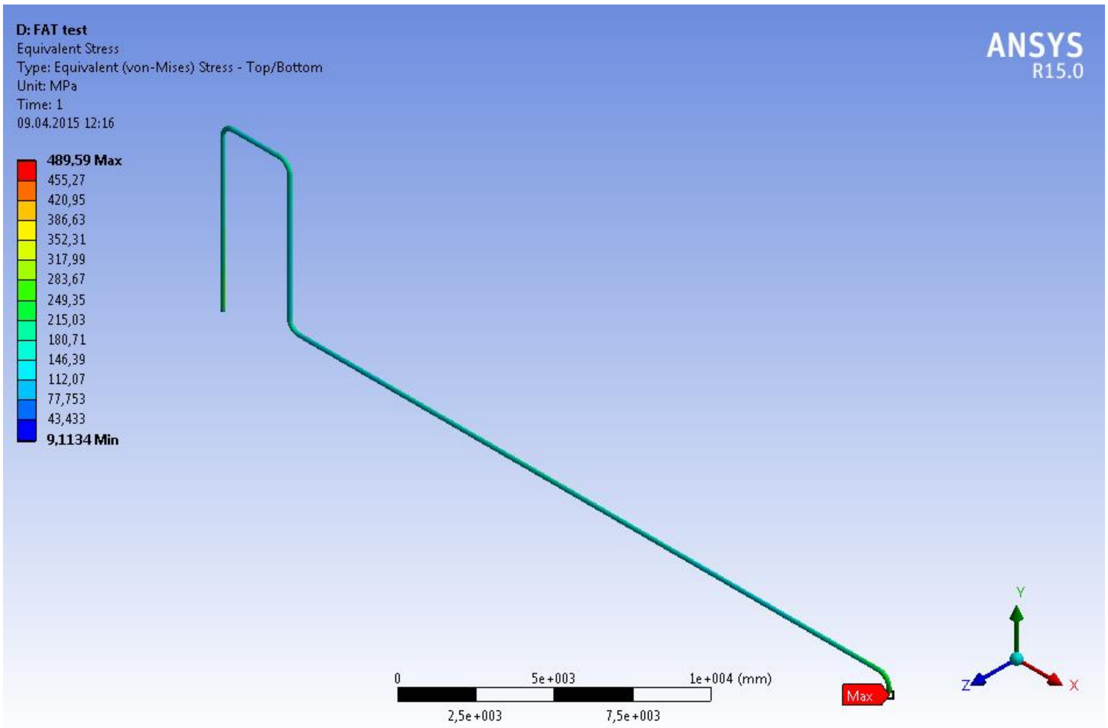


Figure 7-28 Max von Mises stress - FAT

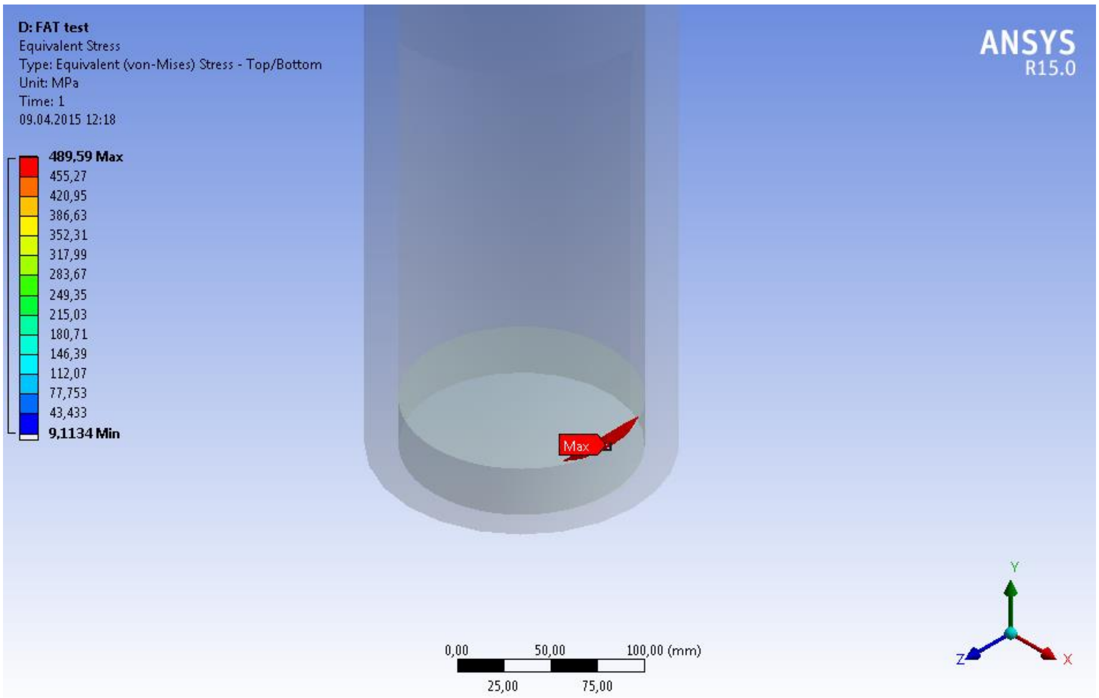


Figure 7-29 Max stress location- FAT

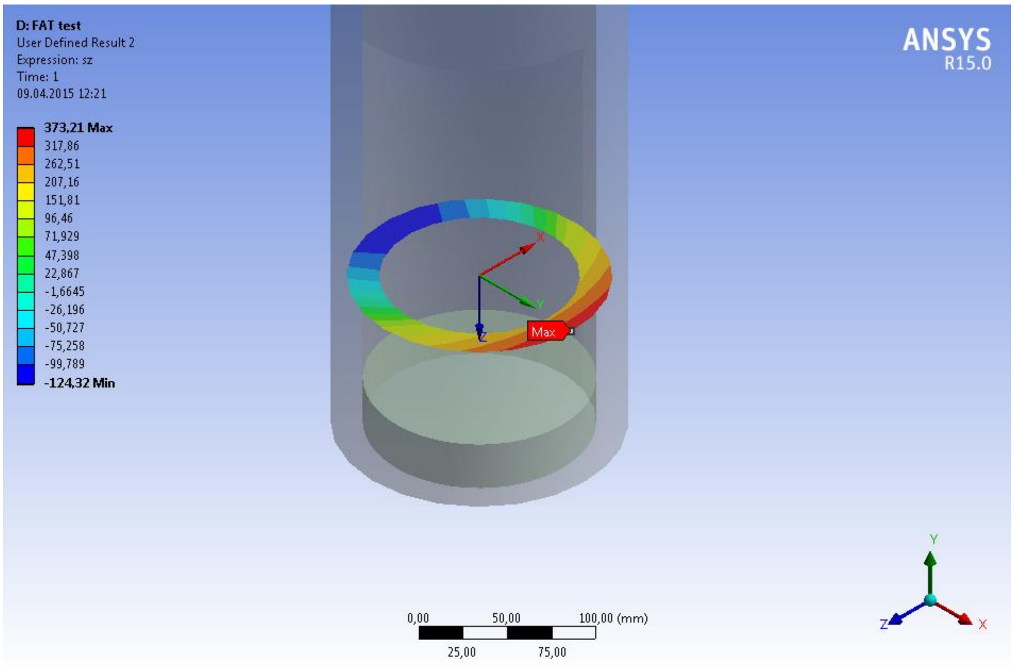


Figure 7-30 Cross sectional longitudinal stress -FAT at XT- end

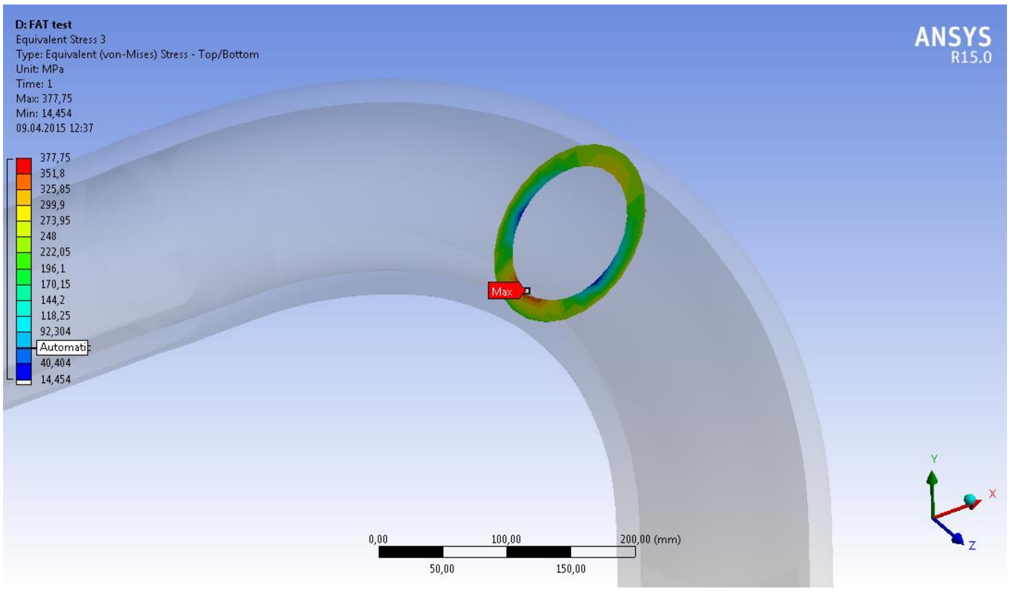


Figure 7-31 Cross sectional von Mises stress -FAT at bend XT- end

7.7 Analysis Result Seal Replacement

The stress results from the seal replacement load case are shown in Table 7-14. The load step applied is:

1. Stroke up of Manifold end and closing of connector
2. Stroke up of Christmas-tree end and closing of connector.

Imposed deformations and tolerances are applied when simulating the seal replacement cycle. Stress as a function of time during seal replacement is shown the figure below.

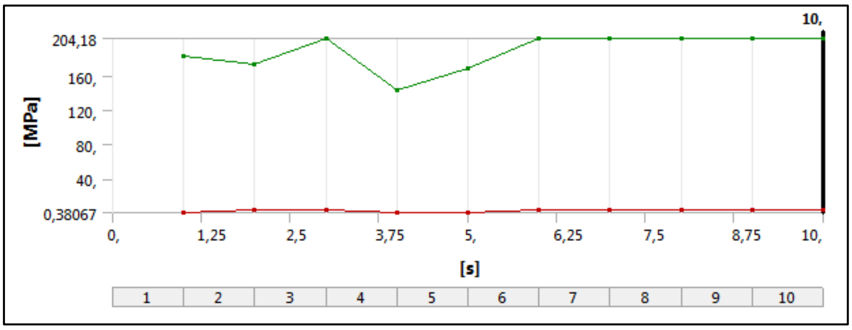


Figure 7-32 Stress in spool- seal replacement

Table 7-14 Max Spool Stresses -Seal Replacement

Location	Combined stress S_C (von Mises) [MPa]	Stress limit [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit [MPa]	UF	Ref.
MF-end	204	432	0.47	-203	432	0.47	Figure 7-36 Figure 7-37

ANSYS Stress and deformation plots for the load sequence are shown in Figure 7-33 to Figure 7-37 on the next pages. All stresses and deformations are moderate and hence the spool has enough flexibility for the service load case.

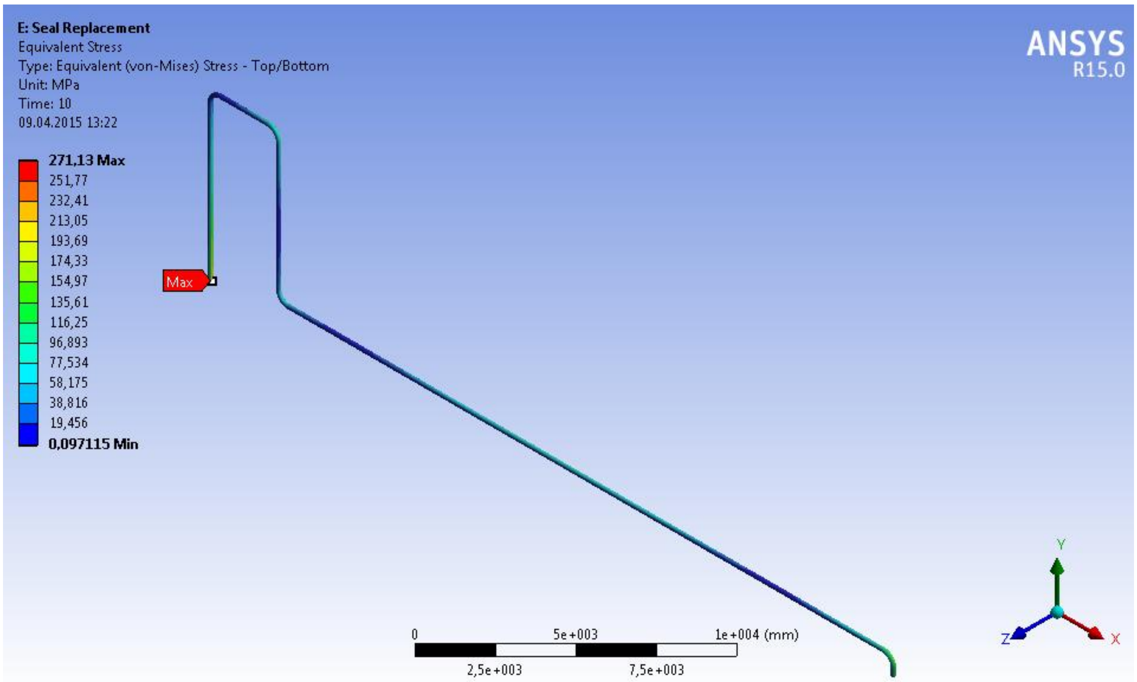


Figure 7-33 Max stress at Seal Replacement sequence

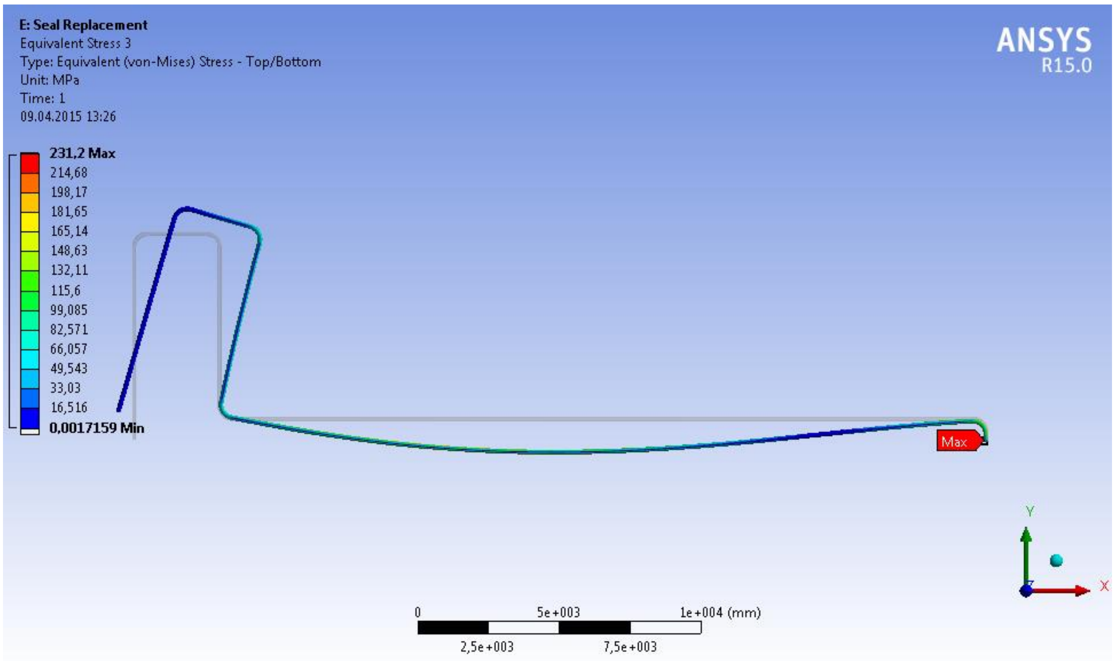


Figure 7-34 Max stress Seal Replacement stroking of MF end

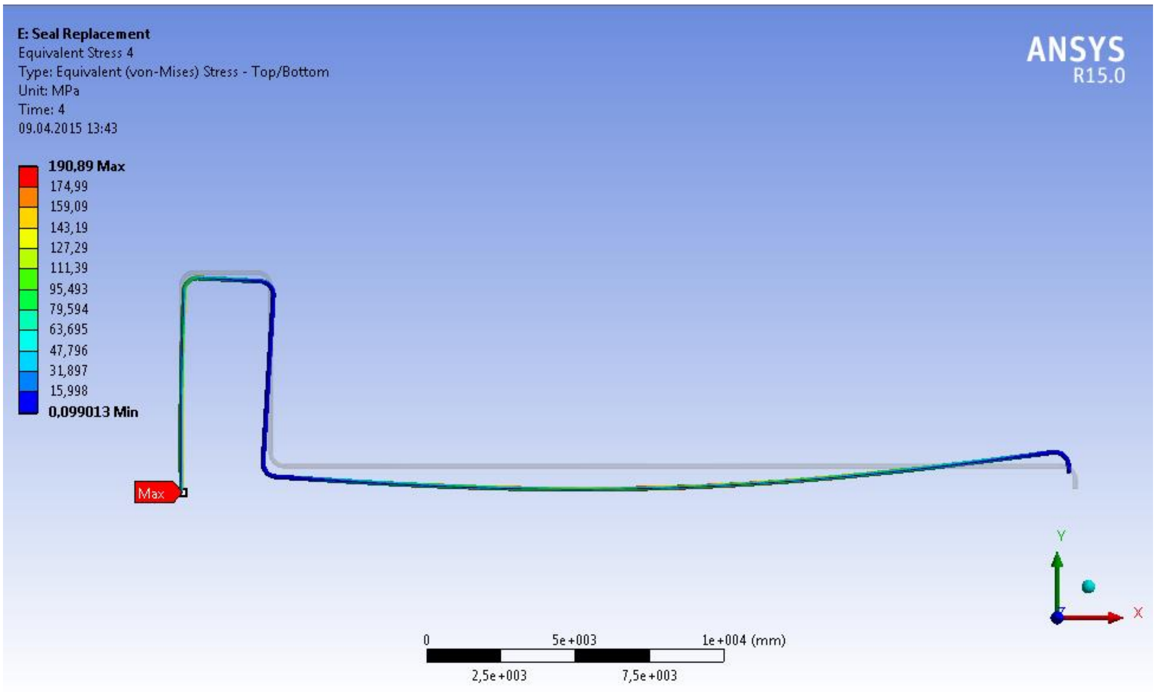


Figure 7-35 Max stress Seal Replacement stroking of XT end

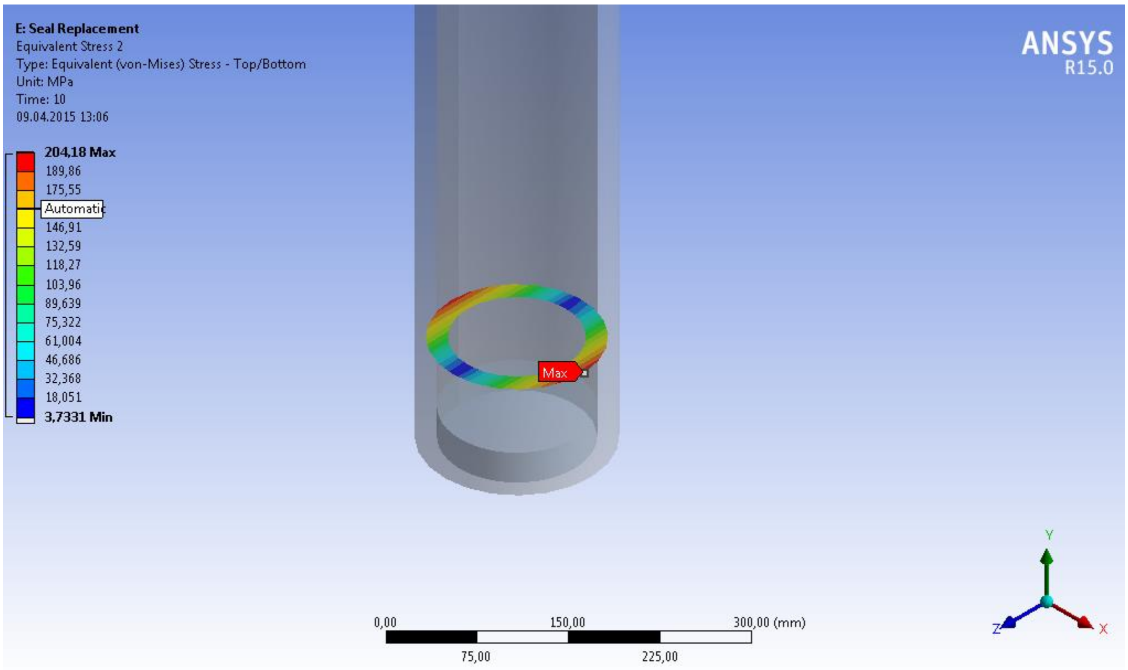


Figure 7-36 Max von Mises stress-Seal Replacement MF end

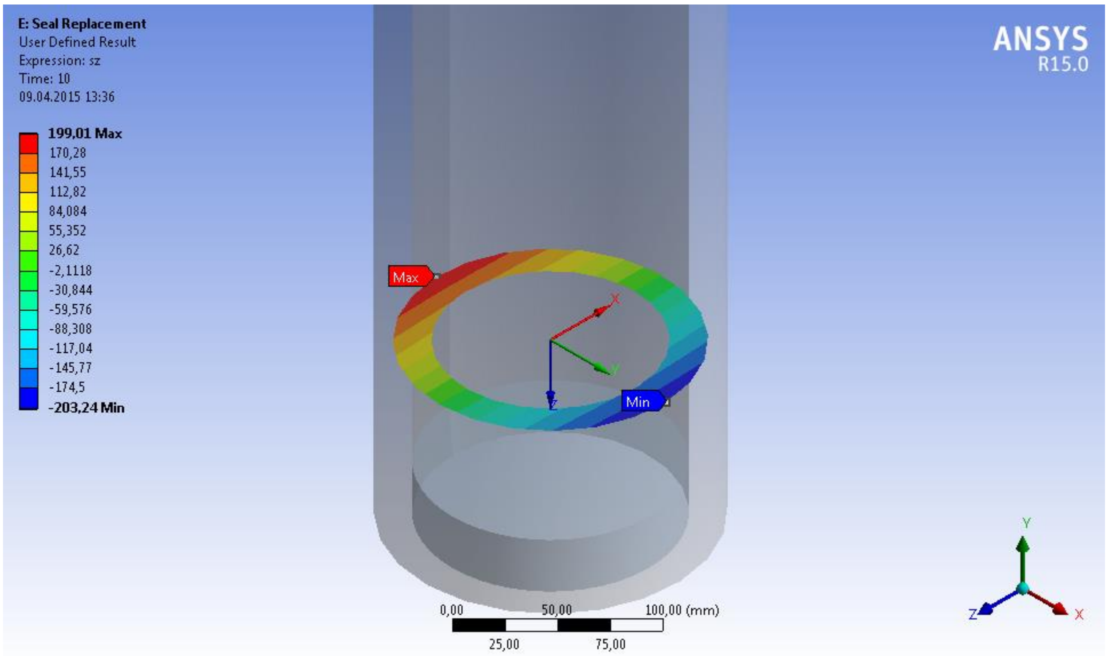


Figure 7-37 Max longitudinal stress-Seal Replacement MF end

7.8 Summary

The usage of the ANSYS design explorer tool shows that the spool is highly sensitive to variation in imposed end rotations and displacements. However the statistical analysis shows the probability of reaching these high stress levels above the allowable code limit is very low, ($1/10^4$ events). The analysis also shows an alternative geometry for the max 1 configuration which will give lesser stress and forces at the connector ends Ref. Table 7-8.

The analysis of the spool also shows a highly utilised spool for the Max 1 configuration with a max **UF=1.01** toward the allowable stress limit of 405MPa, located at the manifold end for the operational load cases. The spool has been analysed with nominal wall thickness and shell elements. The other load cases such as hydro testing and seal replacement shows moderate stress levels with a max UF= 0.86 for the XT tree end.

8. VERIFICATION AND COMPARISON OF RESULTS

In order to check and compare the shell element results from the screening and optimisation process from ANSYS Design Explorer, three different models have been applied for the "Max1" configuration.

- Pipe Element model using ANSYS PIPE289 and ELBOW290 elements
- Solid Element model
- Autopipe Piping Software

8.1 ANSYS Pipe Beam Element Model

The classical way of checking spools was historically made with ANSYS beam Element type PIPE16 and PIPE18. These elements are no longer advised to use as they did not capture the thick wall effects and did not account for cross sectional distortion or non-linear material properties. PIPE18 had the option of using SIF factors and flexibility factors as given in Appendix E of Ref /15/. The elements are today replaced with PIPE288 (Two nodes), PIPE289 (three nodes) and ELBOW290. The following element description is from the ANSYS documentation.

- PIPE289 is a quadratic three-node pipe element in 3-D. The element has six degrees of freedom at each node (the translations in the x, y, and z directions and rotations about the x, y, and z directions). The element is well-suited for linear, large rotation, and/or large strain nonlinear applications. PIPE289 is based on Timoshenko beam theory, a first-order shear-deformation theory. Transverse-shear strain is constant through the cross-section; that is, cross-sections remain plane and undistorted after deformation. The element can be used for slender or stout pipes. Due to the limitations of first-order shear-deformation theory, only moderately "thick" pipes can be analyzed.

- ELBOW290 element is suitable for analyzing pipe structures with initially circular cross-sections and thin to moderately thick pipe walls. The element accounts for cross-section distortion, which can be commonly observed in curved pipe structures under loading. ELBOW290 is a quadratic (three-node) pipe element in 3-D. The element has six degrees of freedom at each node (the translations in the x, y, and z directions and rotations about the x, y, and z directions). The element is well-suited for linear, large rotation, and/or large strain nonlinear applications. Change in pipe thickness is accounted for in geometrically nonlinear analyses. The element accounts for follower (load stiffness) effects of distributed pressures. ELBOW290 can be used in layered applications for modeling laminated composite pipes. The accuracy in modeling composite pipes is governed by the first-order shear-deformation theory (generally referred to as Mindlin-Reissner shell theory).

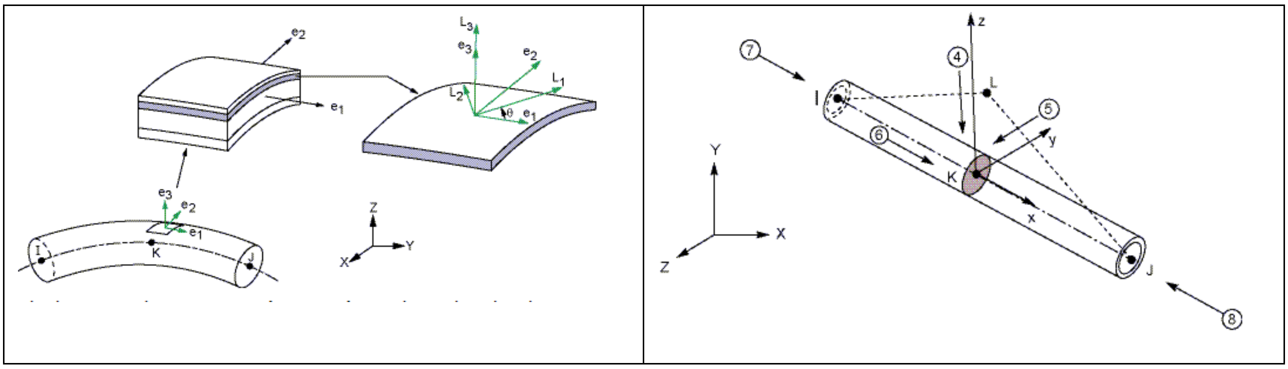


Figure 8-1 ANSYS ELBOW290 and PIPE289 Elements

PIPE 288 and PIPE 289 cannot model cross sectional distortion or collapse in the non-linear analysis. Element ELBOW290 is well-suited for linear and nonlinear applications with large deflection and/or large strain. Various plasticity models, including J2-plasticy, more advanced plasticity models and thermal-plastic-creep models are allowed. Furthermore, this element allows to be used for modelling multi layered composite pipes Ref /21/ .
SIF effects at joints and weld transition is not included in these elements and this has to be accounted and evaluated for in the analysis where required. The flexibility related to bending stiffness effects can be accounted for by use of the SFLEX command for the PIPE288 and PIPE289 Elements.

The Pipe element model is constructed in ANSYS design modeller by use of line bodies and then meshed with beam elements Ref Figure 8-3. The beam elements is given pipe properties as for the shell element model with OD=168.3mm and corroded wall thickness Wt=15.3mm.
ANSYS APDL Commands is used for mesh refinement such as SECDATA, here number of cells along the circumference is determined default is 8 cells. Here we have used 16 cells for better accuracy and a meshing space of 100mm between each element Ref Figure 8-4. The pipe model is then transferred into ANSYS Classical GUI where stress components can be plotted and listed.

The pipe element function in ANSYS workbench and the default elements has the capability to model through wall gradient temperatures. Here the assumption is a uniform design temperature as for the shell models. In reality the fluid would take some time to heat up the pipe and hence one could argue that this effect should be accounted for, especially for thick pipes with large temperature gradients through the wall. Here the operational temperature is max 34^{o} C for Water injection and will most likely settle to the ambient temperature subsea.

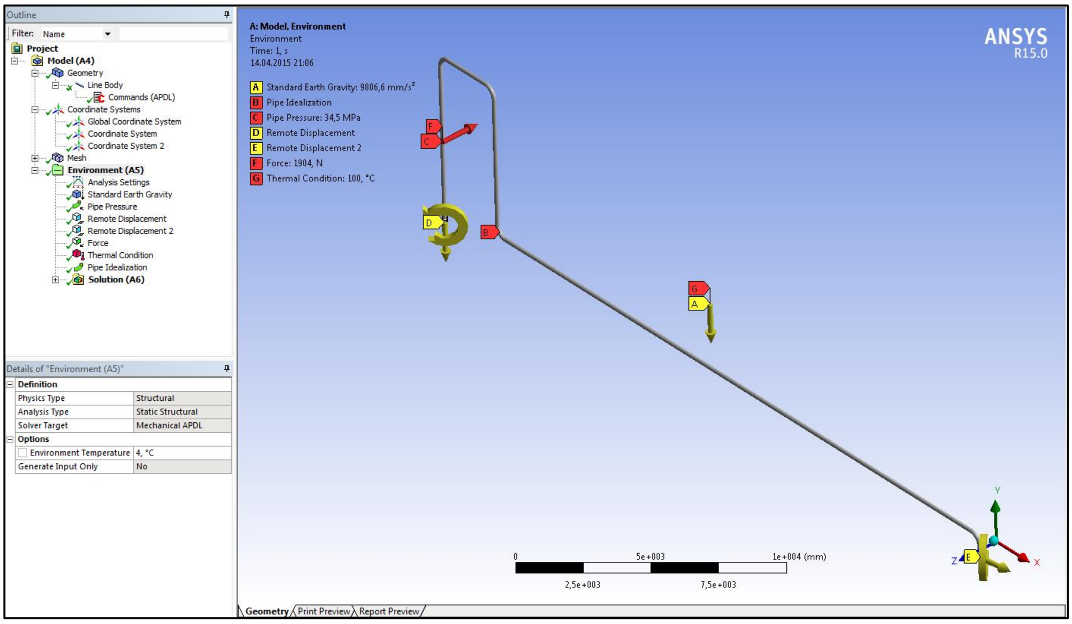


Figure 8-2 ANSYS WorkBench Pipe Model-Loads

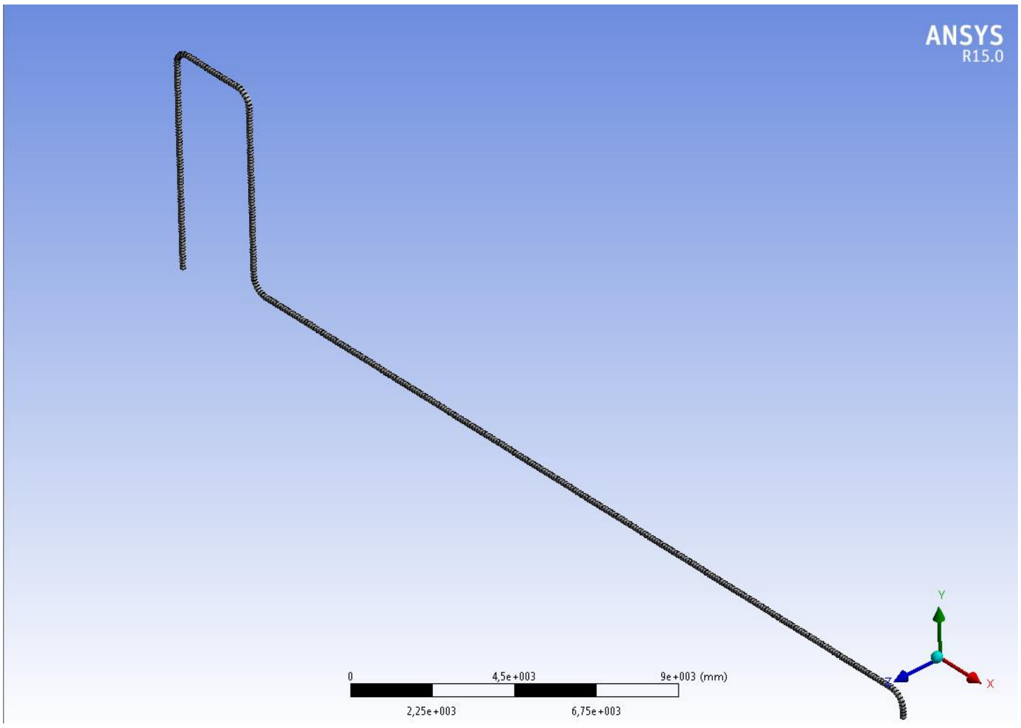


Figure 8-3 ANSYS Workbench Mesh

Table 8-1 list the results from the pipe element model. The analysis shows an over utilisation towards the code limit of 107 %. The pipe element model reports the max von Mises stresses at the bend in the XT-end. Max utilisation is towards the longitudinal stress criteria located at the bend between leg2 and 3.

Table 8-1 Max spool stress Pipe-Model

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF	Ref.
XT-end	400	405	0.99	302	360	0.84	Figure 8-8 Figure 8-9
Bend between leg2 /leg3	352	405	0.87	385	360	**1.07**	Figure 8-10 Figure 8-11

Table 8-2 Reaction forces beam model

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
13,9	-11,1	1,1	4,1	-1,7	66,9
X-tree Side					
FY [kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
7,2	11,1	0,8	0,28	4,5	-49

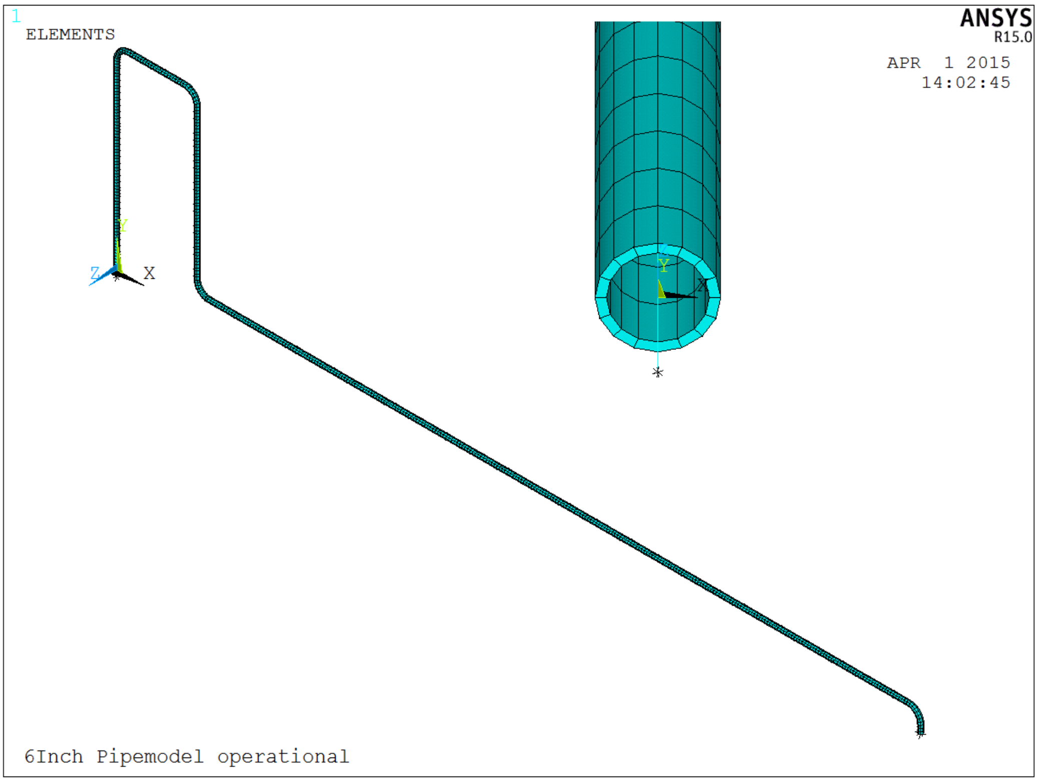


Figure 8-4 ANSYS Pipe Element model

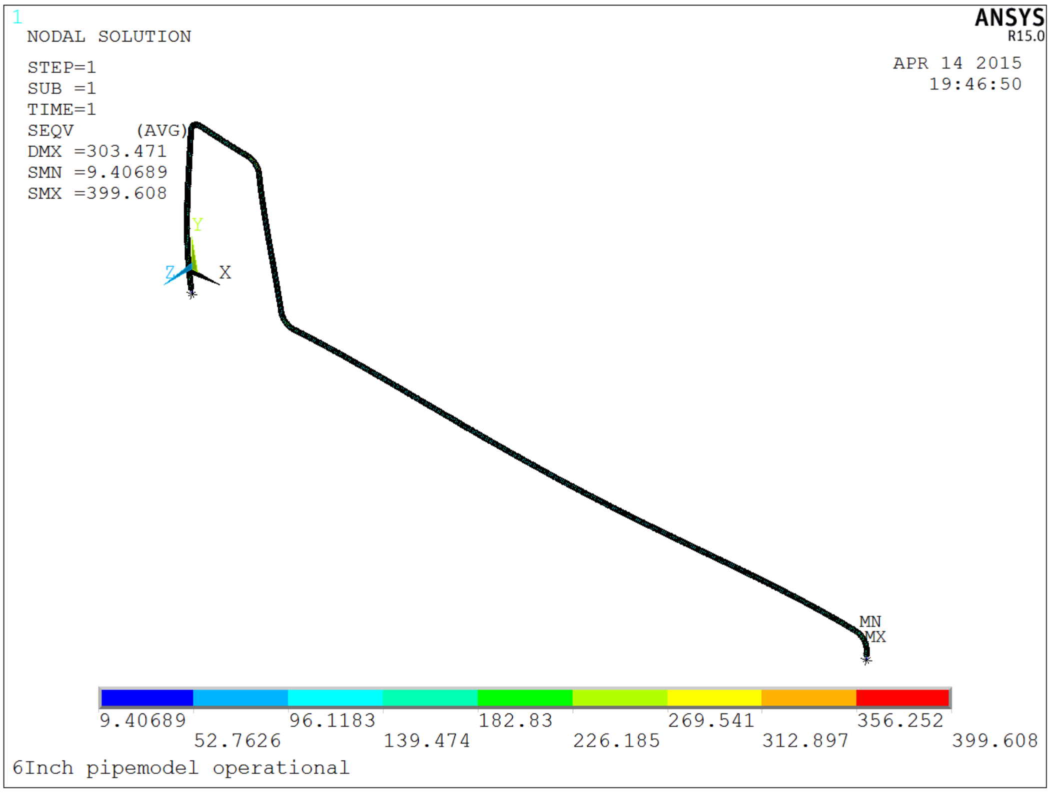


Figure 8-5 Max von Mises stress Pipe Element model

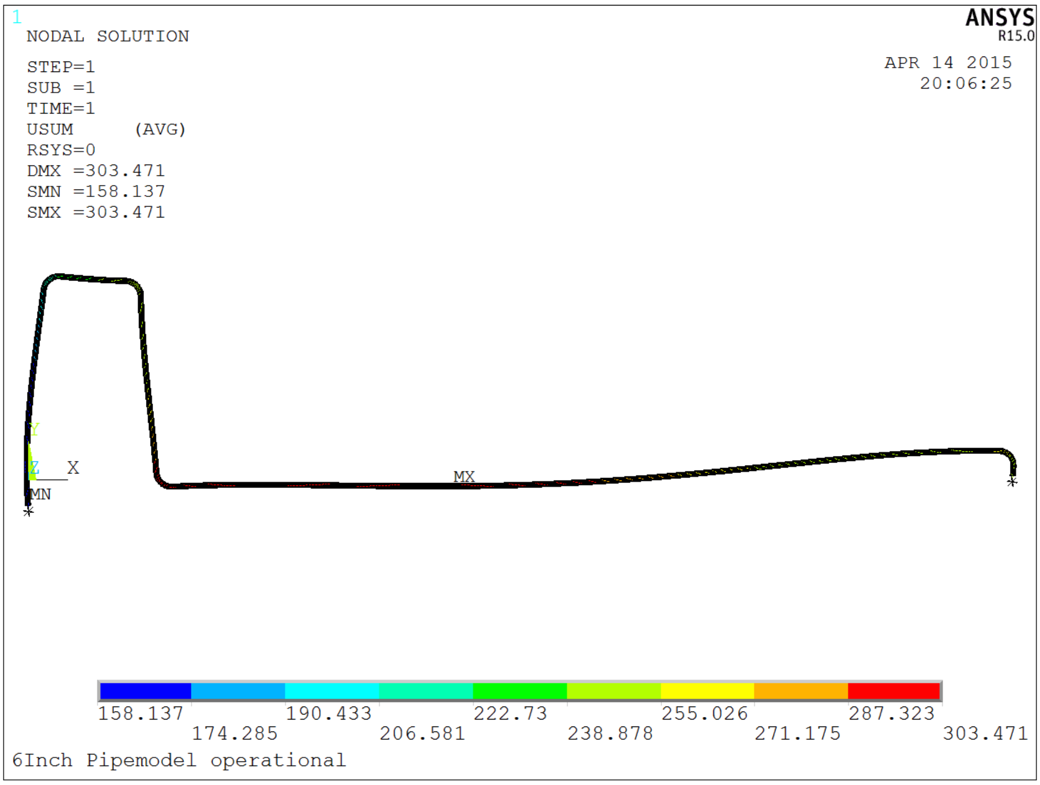


Figure 8-6 Max displacement Pipe Element model

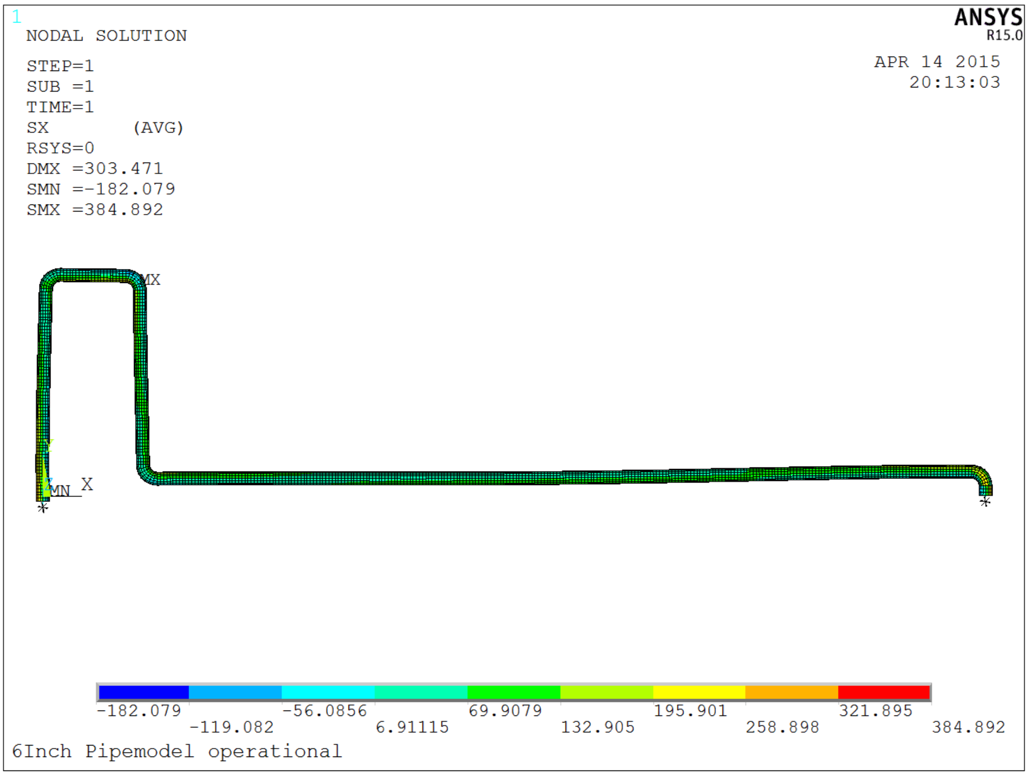


Figure 8-7 Max longitudinal stress Pipe Element model

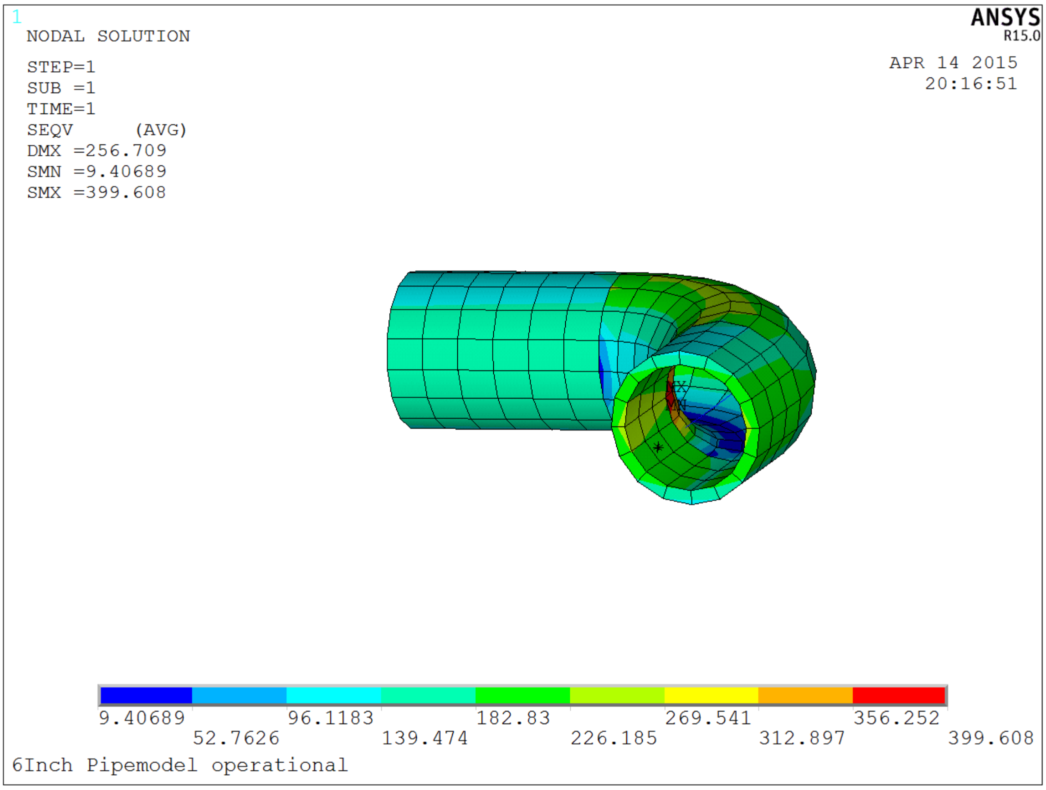


Figure 8-8 Max von Mises Stress Pipe Element model-XT-End

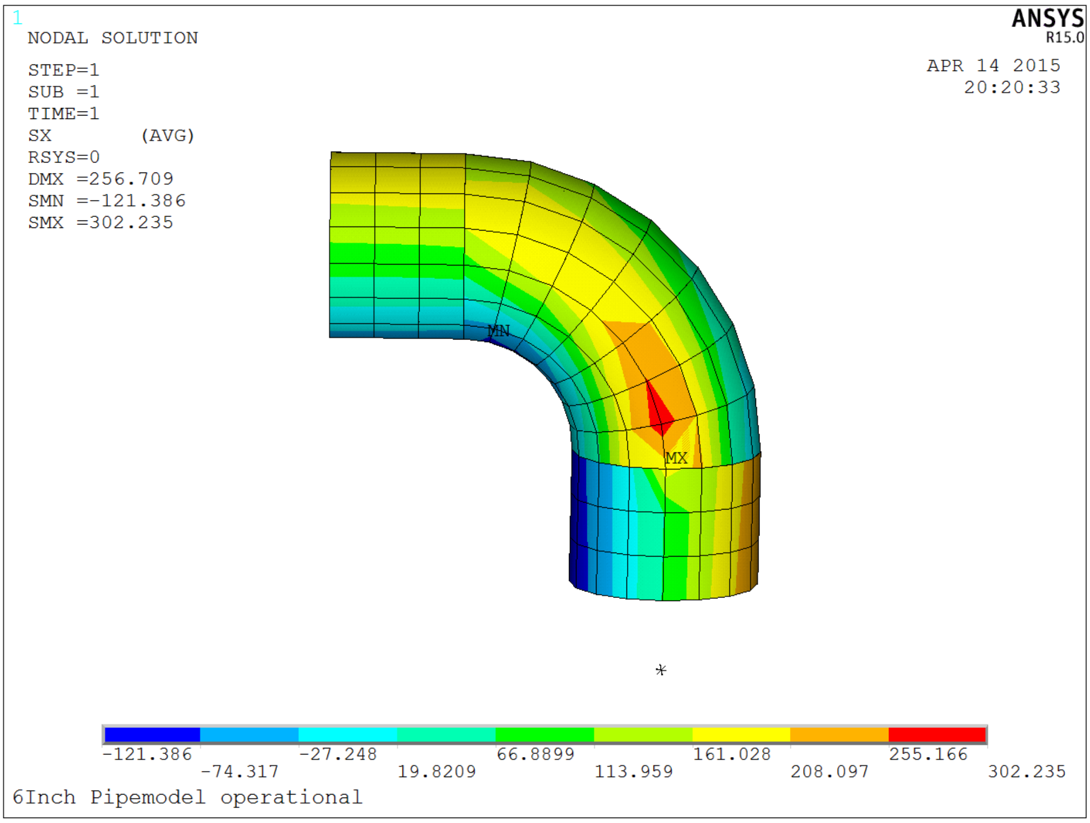


Figure 8-9 Max longitudinal stress Pipe Element model-XT End

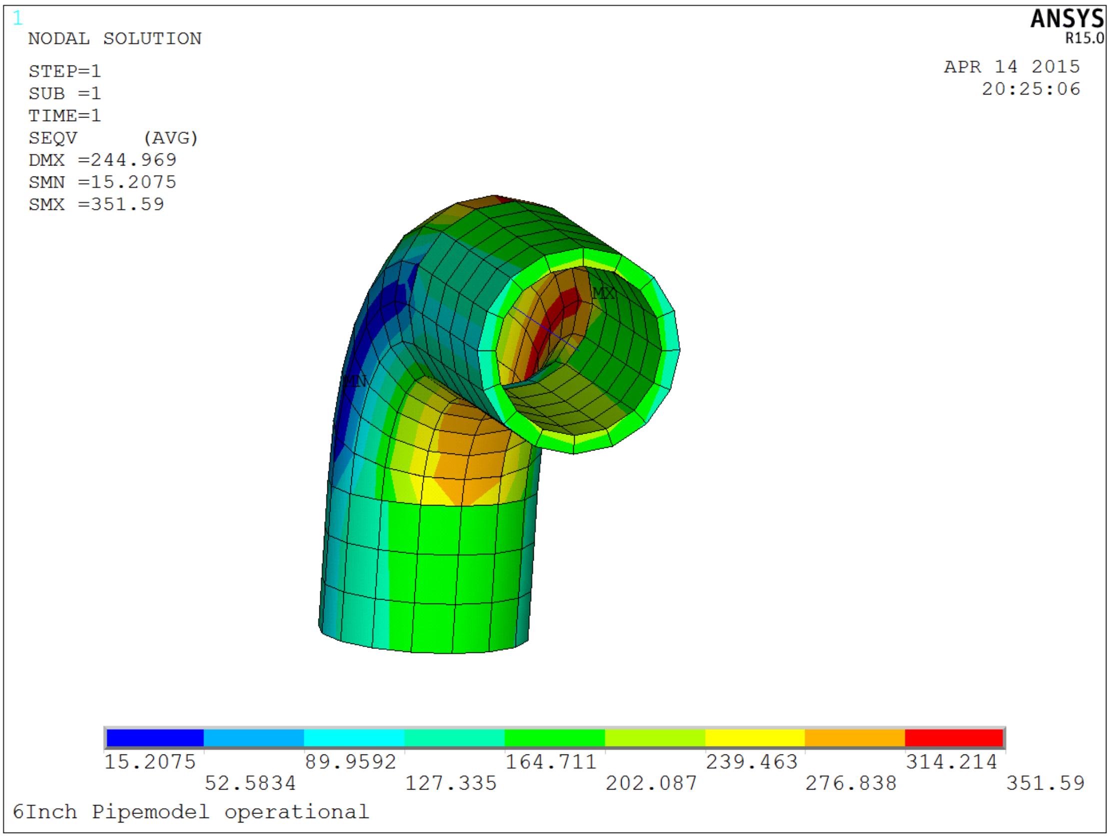


Figure 8-10 Max von Mises stress Pipe Element model at bend

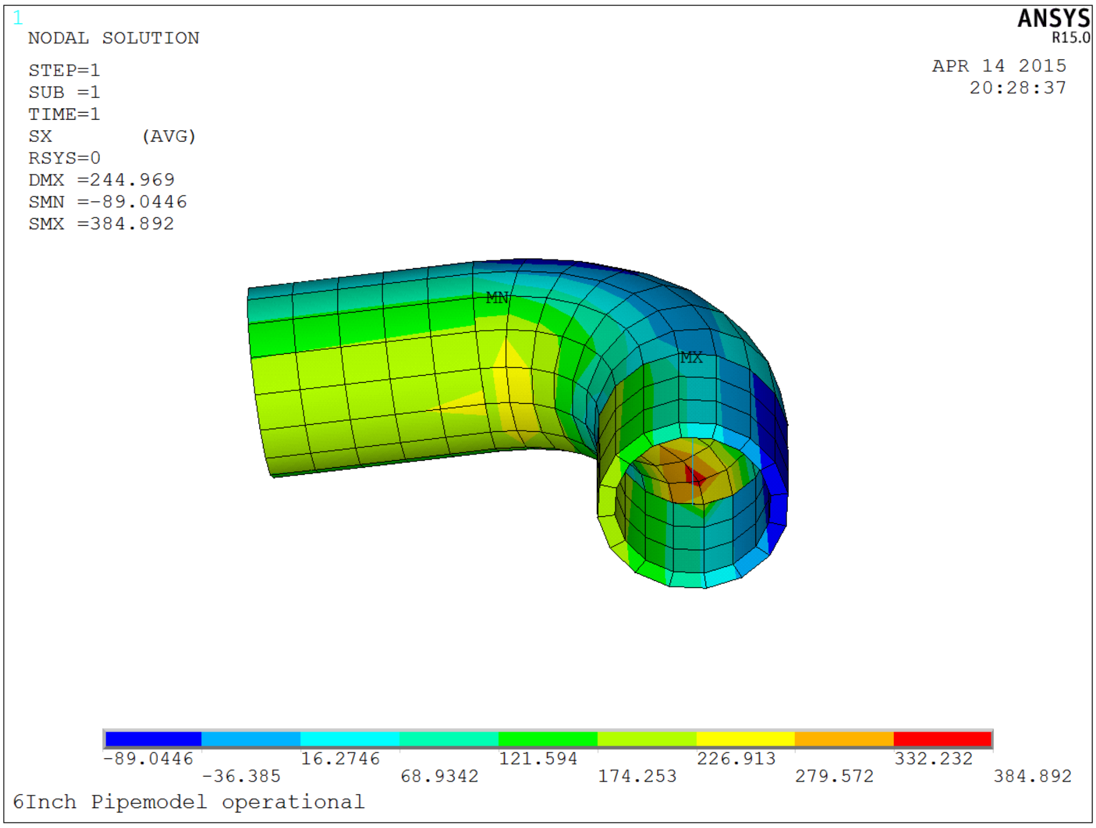


Figure 8-11 Max Longitudinal Stress Pipe Model at Bend

8.2 ANSYS Solid Element Model

A solid model has been developed for the "Max1" Configuration. The model input and boundary condition is equal to the shell model Ref 7.5. The advanced mesh generator in ANSYS workbench has a function called Sweep. For meshing through wall thickness with several elements this method is very feasible for pipes. There is 3 hex meshing or sweeping approaches in workbench. (For an in depth study to this technique, refer to the ANSYS help documentation).

- Standard Sweep method

  -When creating a hex mesh, a source face is meshed and then extruded to the target face. This method of meshing complements the free mesher. If a body's topology is recognized as sweepable, the body can be meshed very efficiently with hexahedral and wedge elements using this technique. The number of nodes and elements for a swept body is usually much smaller than ones meshed with the free mesher. In addition, the time to create these elements is much smaller.

- Thin Sweep method

  -Good at handling multiple sources and targets for thin parts

- Multizone

-Provides free decomposition approach: Attempts to slice up the model without having to do this manually to the geometry.
-Supports multi-source and multi target approach

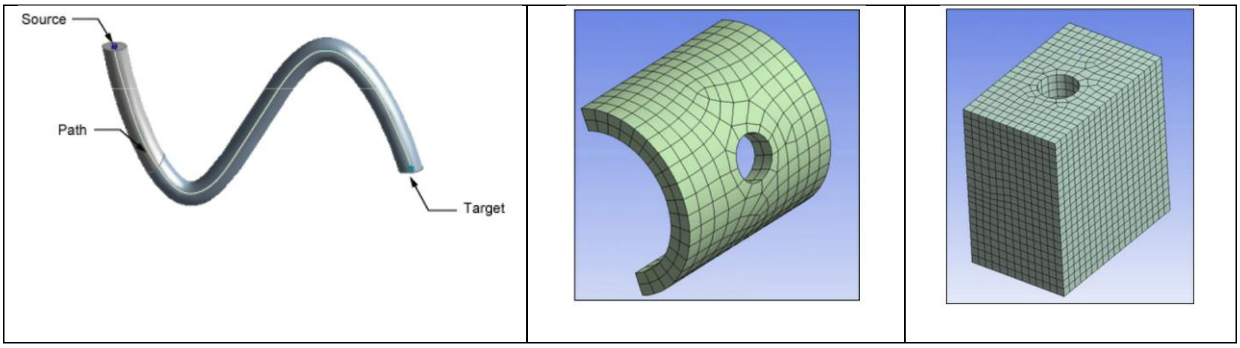


Figure 8-12 ANSYS Sweep Meshing-Examples

The solid mesh for the spool is shown Figure 8-13 and the applied loading is shown in Figure 8-14

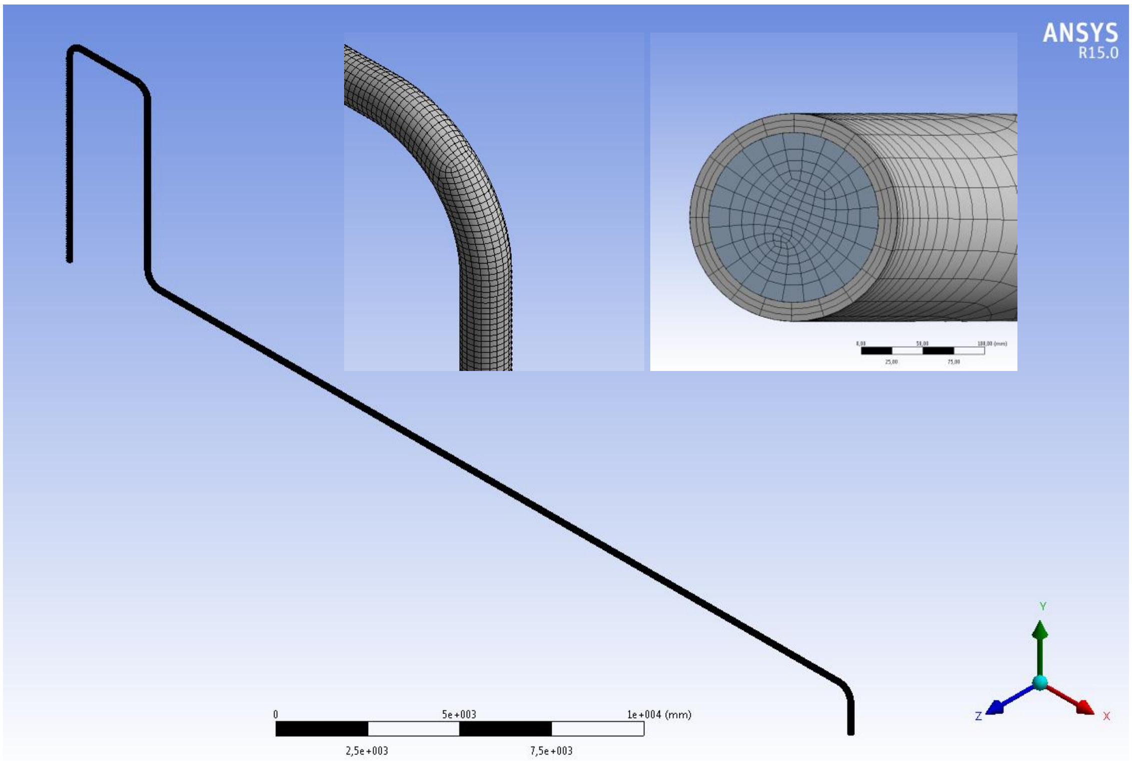


Figure 8-13 Solid Mesh of Spool solid model

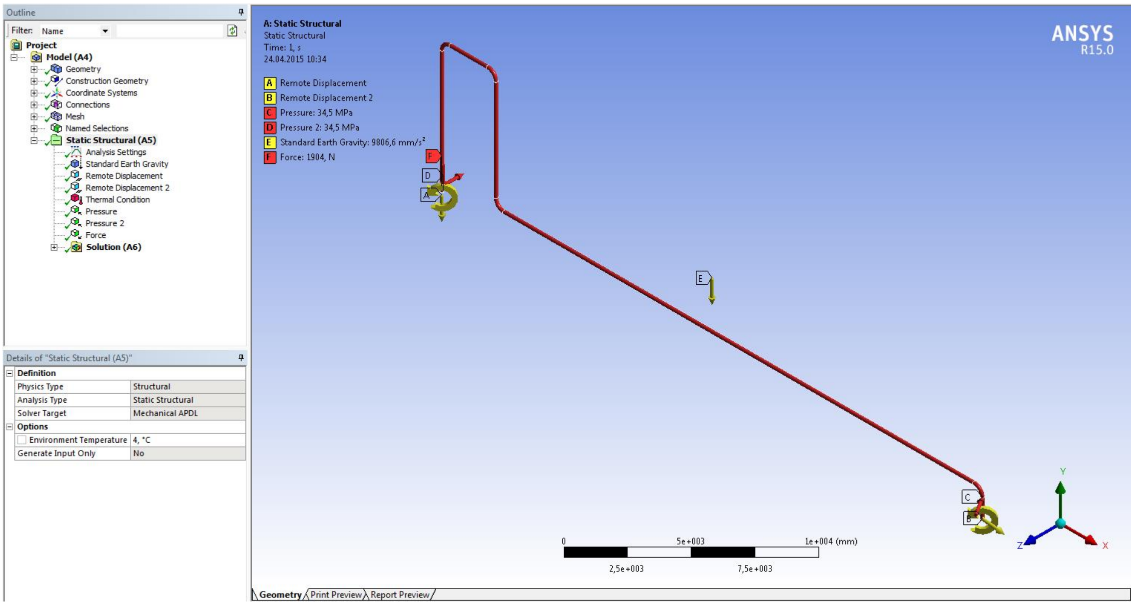


Figure 8-14 ANSYS Workbench solid model-loads

Table 8-3 list the results from the solid element model. The results are extracted in a distance *d* away from boundary conditions and structural discontinuity.

Table 8-3 Max spool stress solid-Model

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF	Ref.
MF end	337	405	0.83	384	360	**1.07**	Figure 8-20 Figure 8-21
Bend between leg2 /leg3	390	405	0.87	298	360	0.82	Figure 8-24 Figure 8-25
Bend XT-end	310	405	0.77	147	360	0.40	

Table 8-4 Reaction forces solid model

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
14.7	11.5	1.7	4.2	-1.7	73
X-tree Side					
FY [kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
-11.5	-0.81	-6.8	37	0.7	-4.2

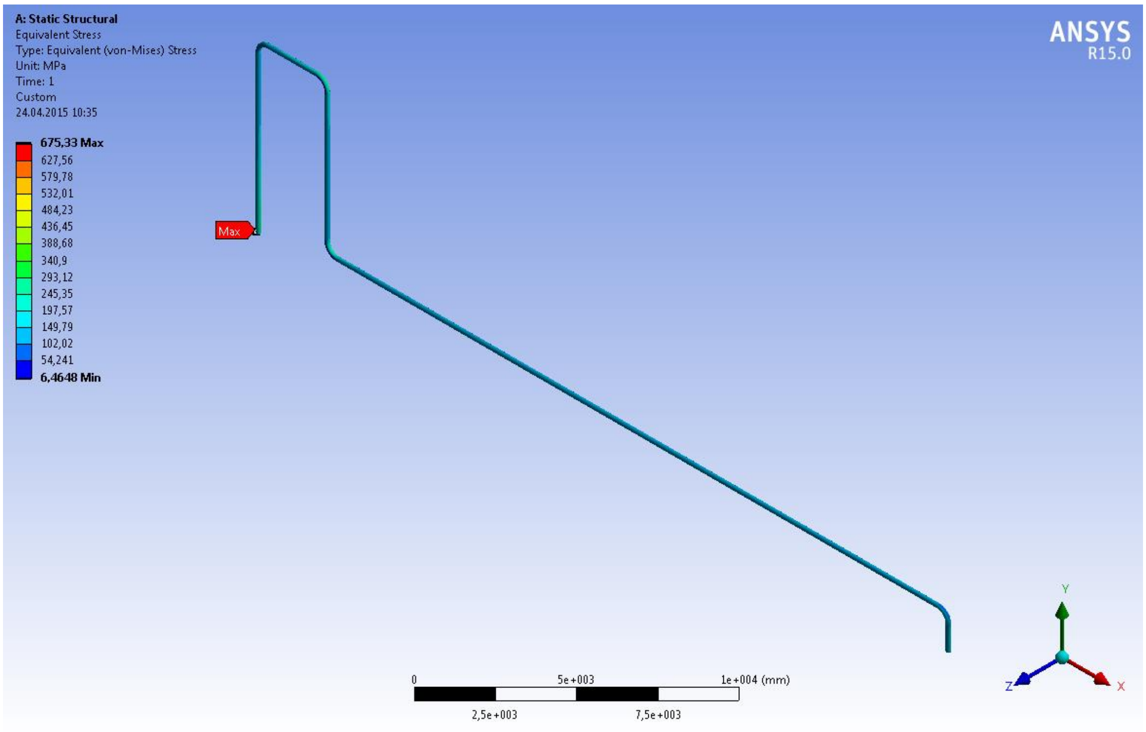


Figure 8-15 Max von Mises stress solid model

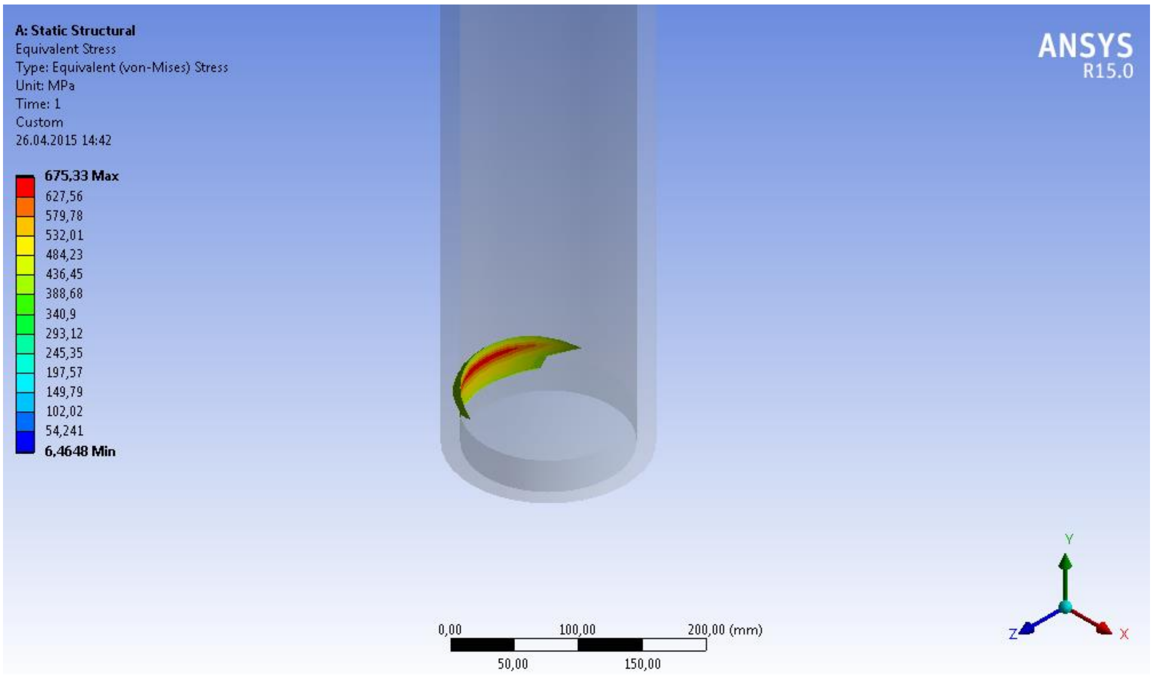


Figure 8-16 Area of max stress higher than 405 MPa

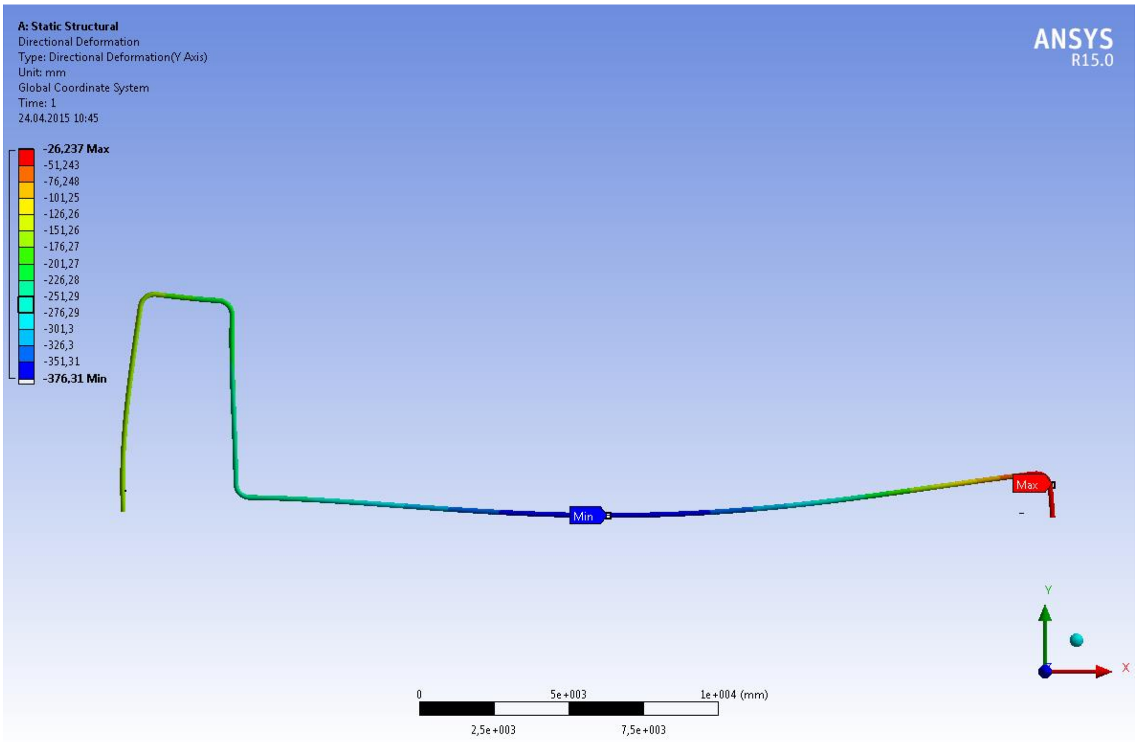


Figure 8-17 Max displacement solid model

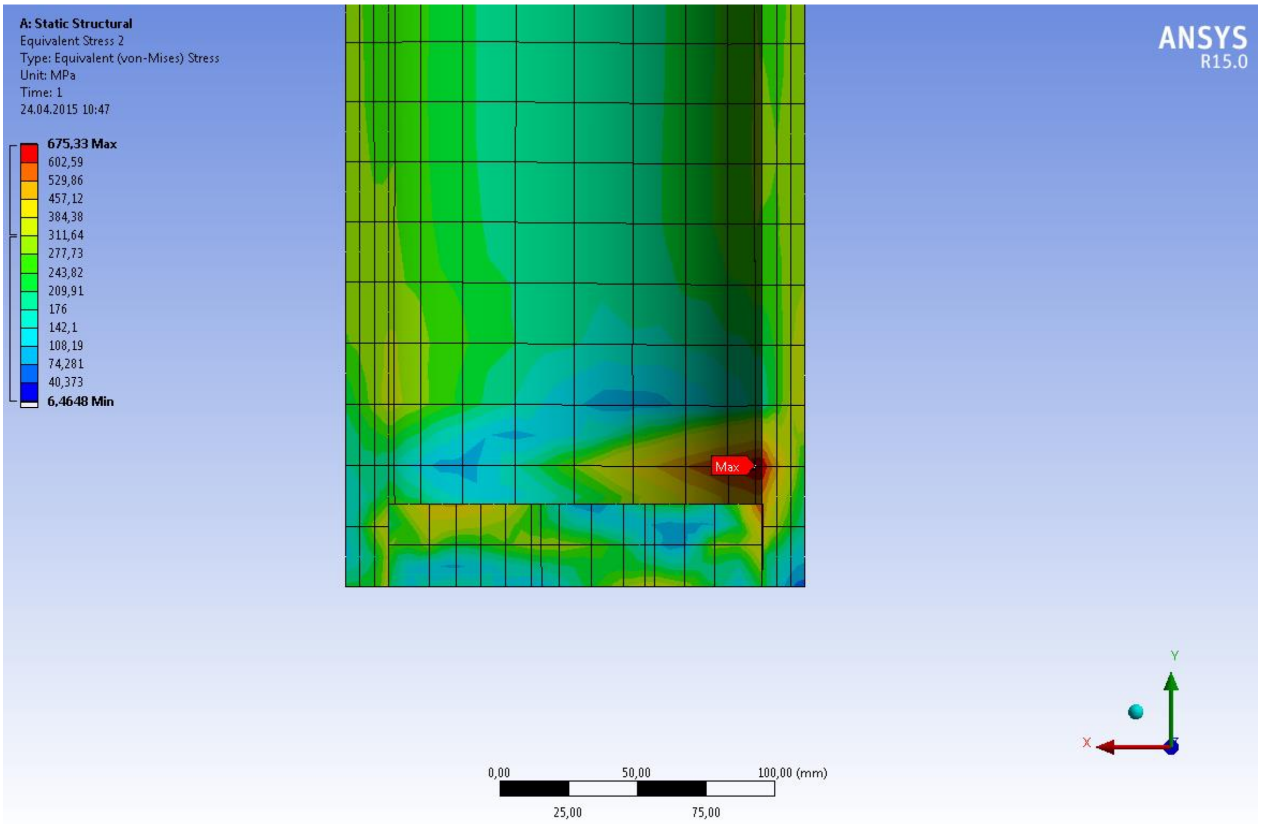


Figure 8-18 Detail max von Mises stress solid model at MF end

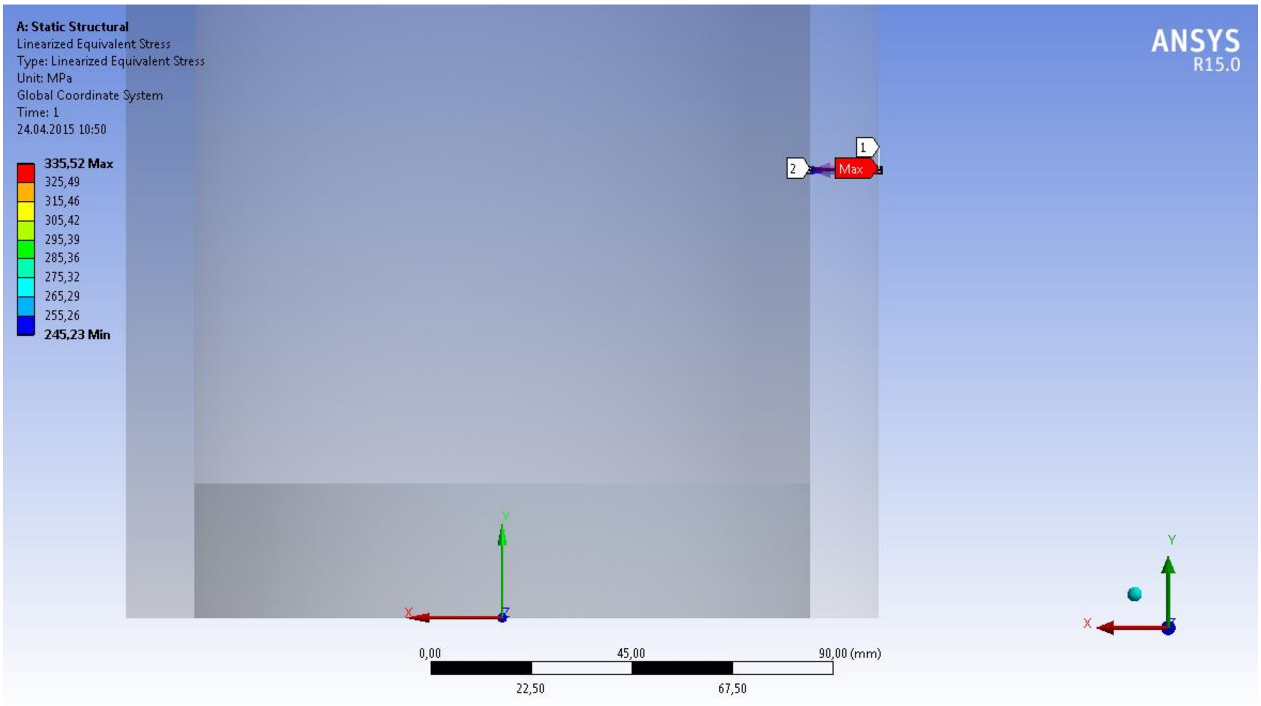


Figure 8-19 Linearized von Mises stress through pipe wall solid model

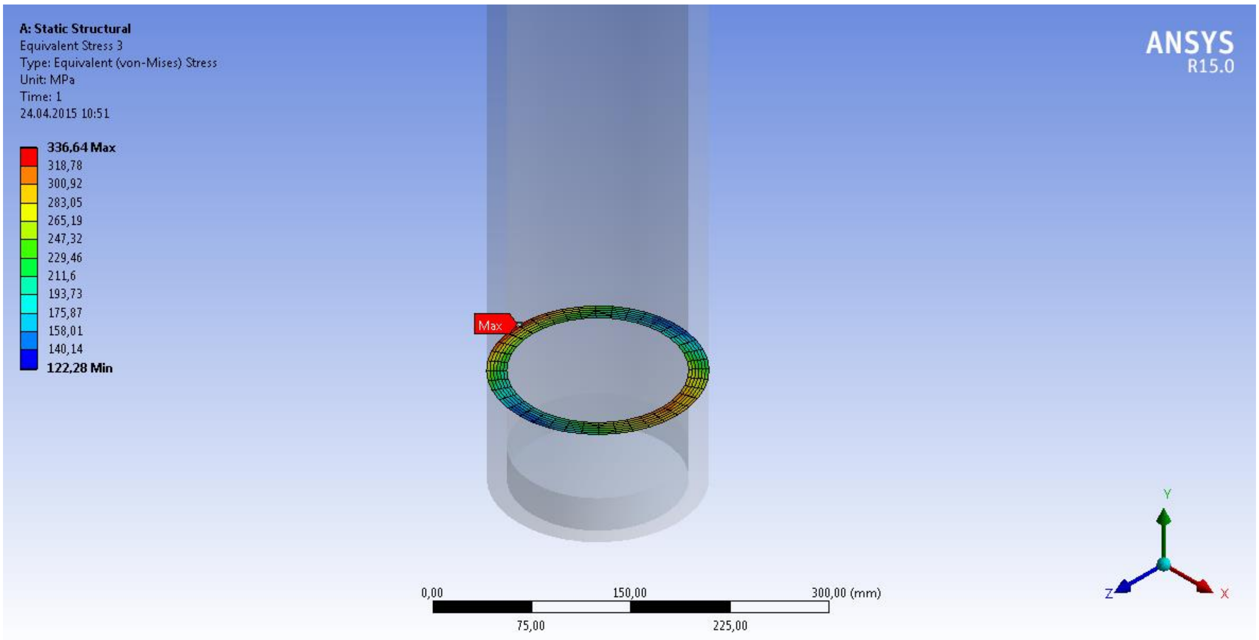


Figure 8-20 Cross sectional von Mises stress solid model at MF end

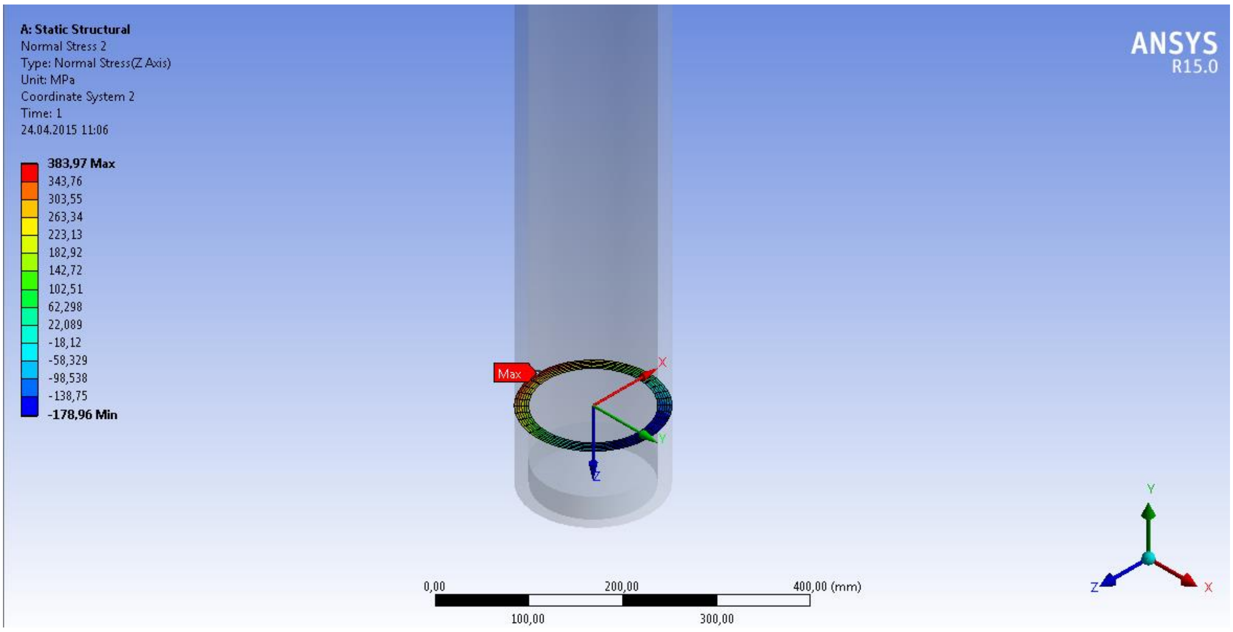


Figure 8-21 Cross sectional longitudinal stress solid model at MF end

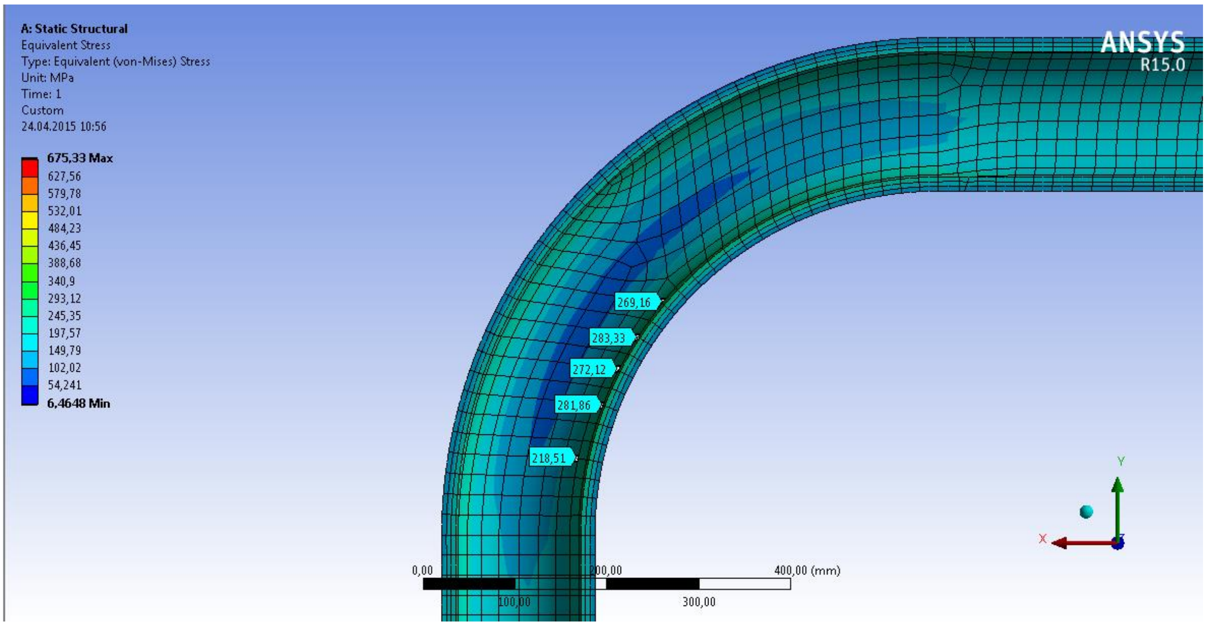


Figure 8-22 von Mises stress bend solid model -XT end

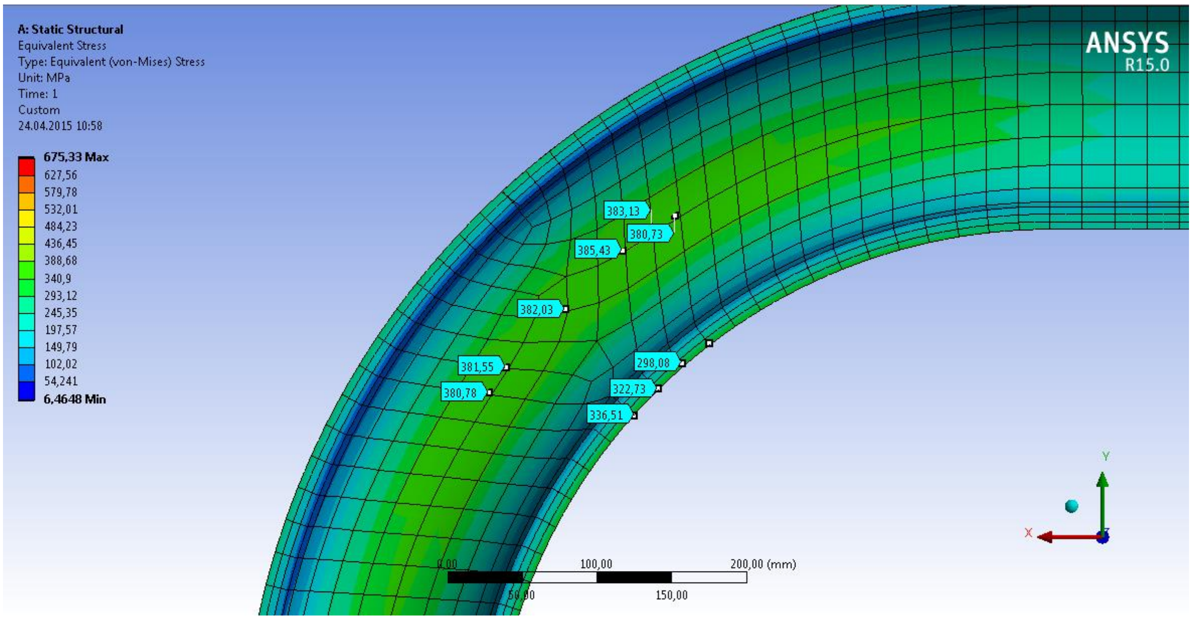


Figure 8-23 Max von Mises stress bend solid model between leg 2 and 3

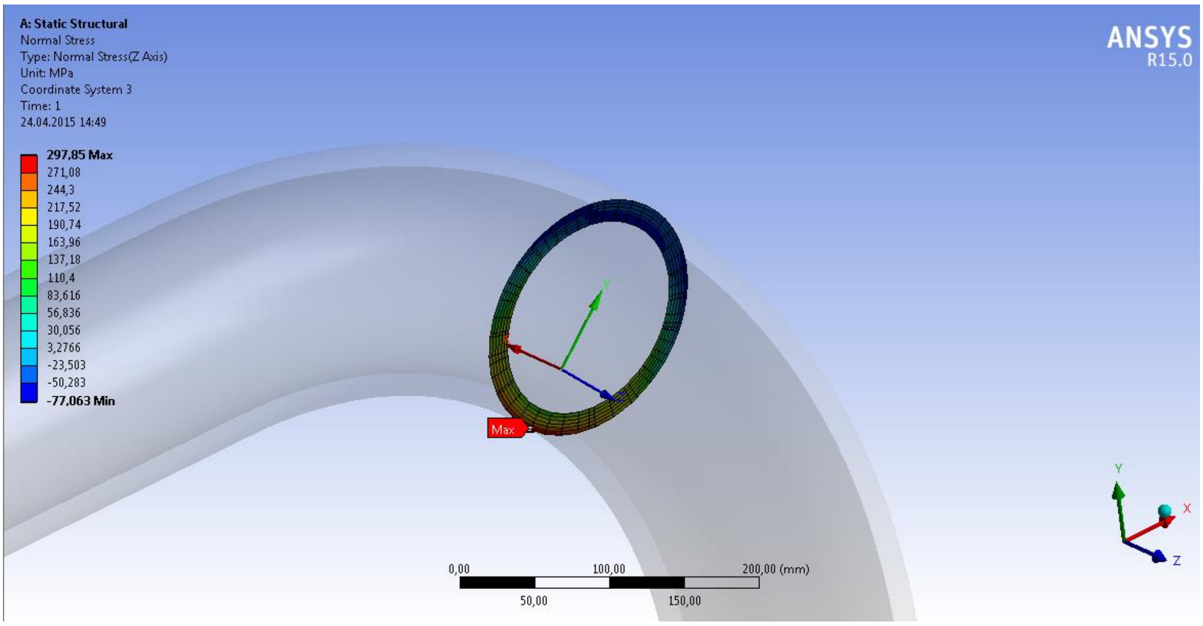


Figure 8-24 Cross sectional longitudinal stress solid model bend between leg 2 and 3

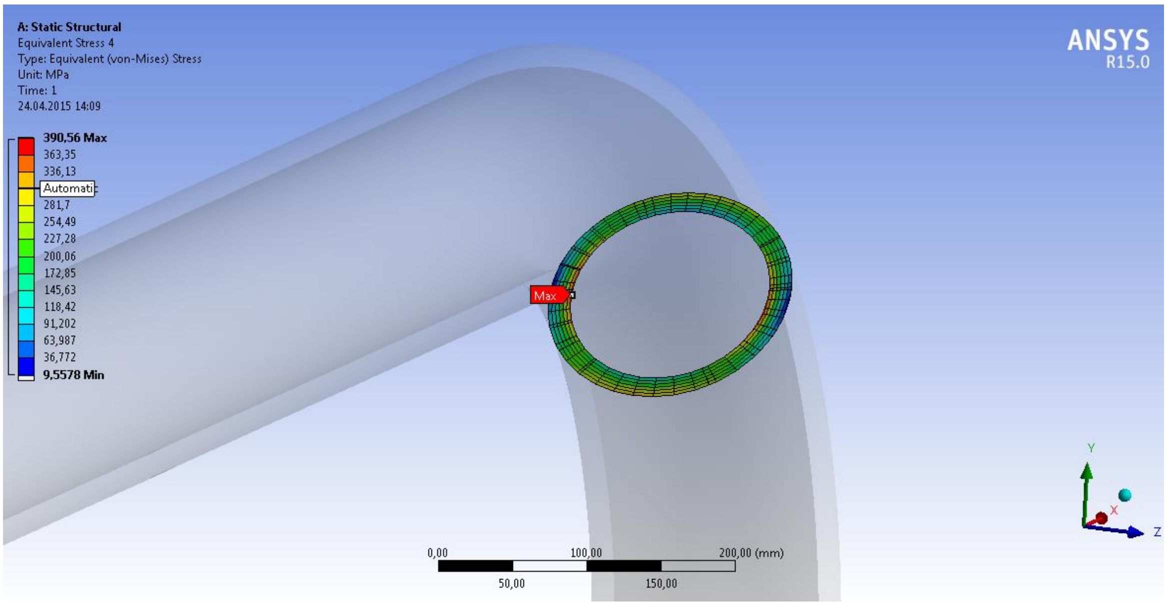


Figure 8-25 Cross sectional von Mises stress solid model bend between leg 2 and 3

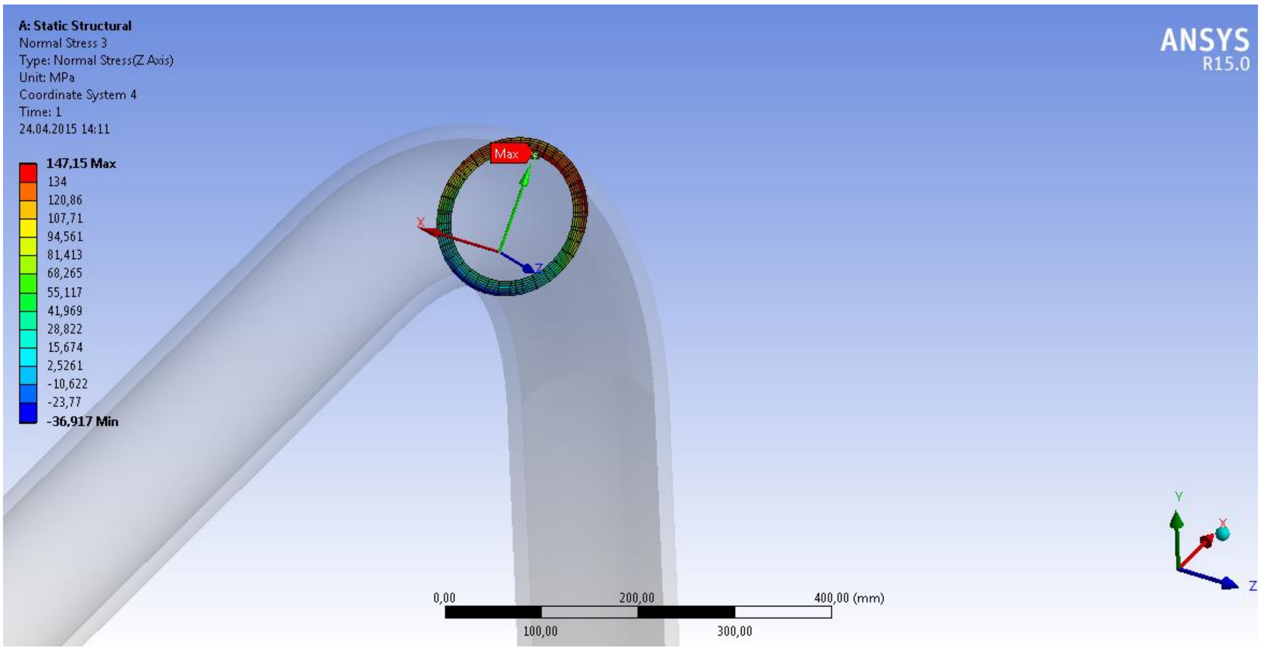


Figure 8-26 Cross sectional longitudinal stress solid model bend XT end

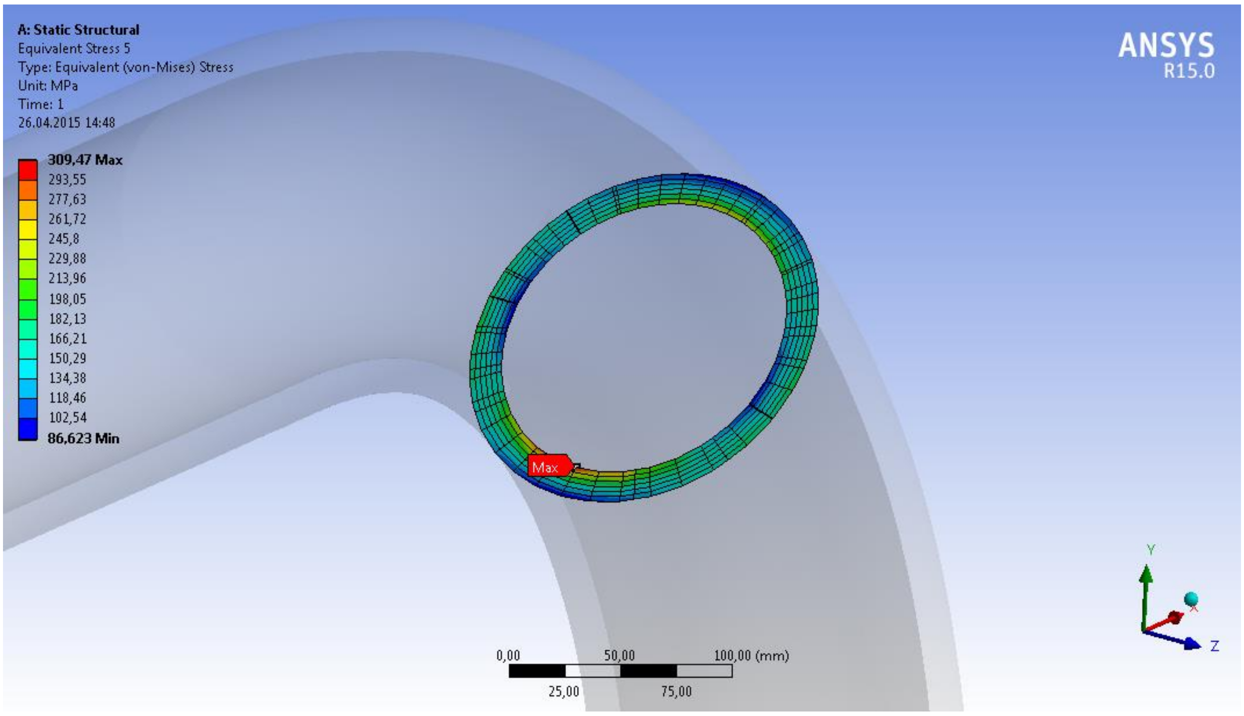


Figure 8-27 Cross sectional von Mises stress solid model bend XT end

8.3 Bentley AutoPIPE Model

One of the analysis software on the market for piping design and process industry is the Bentley AutoPIPE software. This software is tailor made for checking against code and fabrication of piping. This software is natural choice for piping engineers to use when modelling large complex piping system. It is very fast to use and has many features for pipe stress analysis and functions for generating stress isometrics. The three versions are:

- Standard
- AutoPIPE Plus
- AutoPIPE Nuclear

The following description is from the AutoPIPE software documentation:
Bentley AutoPIPE is a stand-alone computer aided engineering (CAE) program for calculation of piping stresses, flange analysis, pipe support design, and equipment nozzle loading analysis under static and dynamic loading conditions. In addition to 24 piping codes, AutoPIPE incorporates ASME, British Standard, API, NEMA, ANSI, ASCE, AISC, UBC, and WRC guidelines and design limits to provide a comprehensive analysis of the entire system. AutoPIPE is available for Windows XP/Vista and can be licensed across networks. The software uses a two node beam element for calculation of forces, displacements and stresses.

There are three versions of AutoPIPE: Standard, Plus, and Nuclear. The Plus version offers several advanced analysis capabilities not available in the Standard version.The Nuclear version offers all of the features of the Plus version with the addition of all the features for ASME class 1 design. A KHK2 Add-On option is also available for the Plus or Nuclear version that allows use of the Japanese KHK Level 2 piping code in addition to all the features of the Plus or Nuclear version. AutoPIPE is a proven, well established program which has been commercially available since 1986. AutoPIPE's rigorous quality assurance practices have withstood numerous on-site audits, making AutoPIPE one of the few PC based piping programs approved for use in nuclear safety applications.

Since the spool is in elastic behaviour and there are no "large" non-linear effects for load and material, the software is very versatile for piping design. The ASME codes which are one of many codes incorporated into these softwares utilises that the goal is to ensure a safe design Ref /22/. In effect the code says that if we calculated the stresses using a *linear analysis* and if we design these piping systems so that these stresses are less than the allowable stresses, then the design has adequate factors of safety for both plastic collapse and fatigue. The calculated stresses are not real they are based upon nominal values which are used in the design

Piping systems can behave non-linearly because of yield and creep of the pipe, and because of non-linear behaviour of the supports. We do not need to account for pipe non-linearity in a structural analysis because the design procedures ensure that the pipe shakes down to elastic behaviour after one or at most a few thermal cycles. However to calculate the stresses accurately we may need to account for support nonlinearity.

For horizontal subsea spools the design utilises large non-linear displacements when stroking is applied and non-linear contact between spool and seabed hence here a non-liner analysis model is mostly used and real stresses and forces are reported and compared towards LRFD codes such as DNV-OS-F101. This feature is not available in AutoPIPE. The recommendation from Bentley is to check the feasibility of the software by using the following rule of thumb:
"As a check that AutoPIPE can handle large deformations, check that maximum slope angle in radians of the deformed pipe≈sin(slope angle), then the solution should be OK. For example 0.025 radians over a large span of 200m then the solution should be OK"

The software version which is used is the standard version. This version does not come with code check according to *ASME B31.8 2010 chapter VIII Offshore Gas Transmission Pipelines*. But the standard version does the code check in accordance with *ASME B31.8 2010 Onshore Pipelines*. This is the same code only different sections within the same code.
The difference is the load combinations and criteria for calculation of longitudinal stresses and allowable stresses. *Chapter VIII* specifies other loadings typical for offshore pipelines compared to onshore pipelines. The code operates with the terminology *restrained piping* and *unrestrained piping* this dictates how one should calculate stress and what limits to compare against.
The following table list the code load combinations for restrained piping according to the code. For the spool checking this becomes:

Table 8-5 AutoPIPE ASME B31.8 code Load combinations

Load combination	Description
GR[1]+ Max P(1)	Sustained loading ,gravity + max pressure P=345 bar Imposed rotations and deflections at anchor points
Ambient to T(1)	Temperature expansion at ambient temperature=4°C T1=ambient gravity not included
Ambient to T(2)	Temperature expansion from ambient to T2=100 °C-Temperature gravity not included
Max P(1)	Max Pressure P=345bar
GR+T(1)+P(1)	Gravity + Temperature (1)+Pressure (1)
GR+T(2)+P(2)	Gravity + Temperature (2)+Pressure (2)

In order to be able to compare the results from the ANSYS model, the documentation of the software and the understanding of how the program calculates stresses must be done. The software has many options which must be checked and understood before running analysis. The software is designed in such a way that it reflects the process from design to fabrication, control, testing and operating. The limit for longitudinal stresses and the setting for using the octahedral von Mises criteria for yield instead of using the Tresca criteria are changed from default values by the software. Stress Intensification factors (SIF) at transitions and welds is default by programme and is set equal to SIF=1.0. Here the software has the option to input user SIF and also to allow the software to calculate SIF by entering values at transitions such as weld eccentricity.

The stresses are calculated based upon the following formulas from the code (Equation 833.3a):

$$|S_L| = kS_yT \tag{8.1}$$

Here k is default to *0.75* for unrestrained piping but changed to *0.80* in accordance with *chapter VIII* of the code.

S_L = Maximum longitudinal stress, psi (positive tensile or negative compressive) = combined axial and bending stress as $\sigma_a + \sigma_b$, or $\sigma_a - \sigma_b$ whichever results in the larger stress value.

Bending stress:

$$\sigma_b = \frac{\sqrt{(0.75i_iM_i)^2 + (0.75i_oM_o)^2 + M_t^2}}{Z_{nom}} \tag{8.2}$$

The SIF factors (i)are reduced with a factor of 0.75 according to code for sustained loads, this is not the case for the formula in *Section VIII* of the ASME B31.8 code Ref. equation (5.6) where there are no such reductions. The SIF factors are code SIF factors and are used for fatigue calculations based upon thermal expansions and are based upon tests towards commercial girth welded pipes Ref. Section 3 of Ref. /35/. These factors are not the same as geometrical stress concentration factors (SCF or theoretical SIF) and should not be used with fatigue curves from other codes. The usage of SIF factors can vary between the ASME codes.
The section modulus Z_{nom} is based upon the nominal wall thickness value. This is changed to the corroded wall thickness Wt=15.3mm as analysed in the ANSYS model. For Hoop stress the nominal wall thickness is used by default.

[1] Buoyancy force from submerged spool of 100N/m and Drag force of 46N/m is applied in the GR-Load case.

It should be noted that the AutoPIPE program calculates the code combination stresses with default section modulus (Z_{red}) based upon the reduced wall thickness which is equal to:

$$t_{red} = t_{nominal} - t_{corrosion} - t_{mill} \tag{8.3}$$

The mill tolerance is usually 12.5% of the wall thickness.

The mill tolerance is set to zero to achieve correct comparison

The axial stress is calculated as:

$$\sigma_a = \frac{F_{axl}}{A} = \frac{PD_o}{4t} = 0.5S_h \tag{8.4}$$

The axial stress includes the pressure term from endcap:

Where

M_i= In plane bending moment
M_o= out of plane bending moment
M_t= Torsional moment
Z_{nom}= Section modulus, nominal wall thickness
D_o= Outer diameter
A= Section area of pipe
F_{axl}=Axial force including pressure term
$i_{o,i}$=Stress intensification factor (SIF) out of plane or in plane
t= Nominal wall thickness
P= Pressure
S_h=Hoop stress

8.4 Bentley AutoPIPE Results

The stress results from the analysis is shown in Figure 8-28 the utilisations are in accordance with the code check towards the stress limits as described in chapters 5.9.3. The results are listed in Table 8-6

Table 8-6 AutoPIPE ASME B31.8 Code stress utilisations Corroded condition

Load combination	Code Stress [MPa]	Allowable [MPa]	Location	Utilisation UF
Sustained loading	435	405	MF end-(A00)	**1.07**
Ambient to T(1)	NA	198	NA	NA
Ambient to T(2)	17	3	Bend between leg 2 and 3	**5.0**
Max P(1)	159	324	XT-end –(A05)	0.49
GR+T(1)+P(1)	404	405	MF end-(A00)	1.00
GR+T(1)+P(1) Longitudinal stress S_L	470	360	MF end-(A00)	**1.14**
GR+T(2)+P(2)	389	405	MF end-(A00)	0.96
GR+T(2)+P(2) Longitudinal stress	393	360	MF-end	**1.09**

As seen from Table 8-6 the software reports utilisations above 1. This means that the spool does not meet the code requirements in corroded condition. The thermal expansion cases ambient to T(1) or T(2) are related to *fatigue code limits based upon a minimum of 7000 thermal cycles* , and is given for unrestrained pipelines. The thermal expansion criteria are given by the following equation (Equation 833.8).

$$S_A = f[1.25(S_c + S_h) - S_L] \tag{8.5}$$

Where:

- f= $6N^{-0.2} \leq 1.0$, SN fatigue curve
- S_c= $0.33S_uT$ at the minimum temperature
- S_h= $0.33S_uT$ at the max temperature
- S_L= Longitudinal stress
- S_u= Ultimate tensile strength
- T= Temperature derating factor
- N= equivalent number of cycles during the expected service life of the piping system (N_{min}= 7000)

The formula is very dependent upon the longitudinal stress and since the longitudinal stress ratio is over utilised there is basically no fatigue capacity left for thermal expansion stress cycles. The minimum value for the f factor is 1.0 which equals to 7000 thermal cycles and cannot be lower according to the code. Pressure and temperature fluctuations for subsea spools are very depend upon the production stream medium i.e multiphase well stream, produced gas, Injection water or oil. However one major difference is the amount of start-up and shut downs cycles. Normally the spools are connected to a subsea production pipeline or a well and experience less start up and shut downs (N≤1000 cycles) during its lifetime compared to other piping systems. So this means that equation (8.5) cannot be used for cycles less than 7000. Equation (8.5) is not given in Section VIII of the ASME B31.8 code for offshore pipelines. ASME B31.8 section A842.2.5 refers to API RP 1111 for fatigue evaluation.
From project experience the DNV-RP-C203 Ref. /14/ for low cycle / high cycle fatigue and VIV assessment's (Miner summation) is often used.

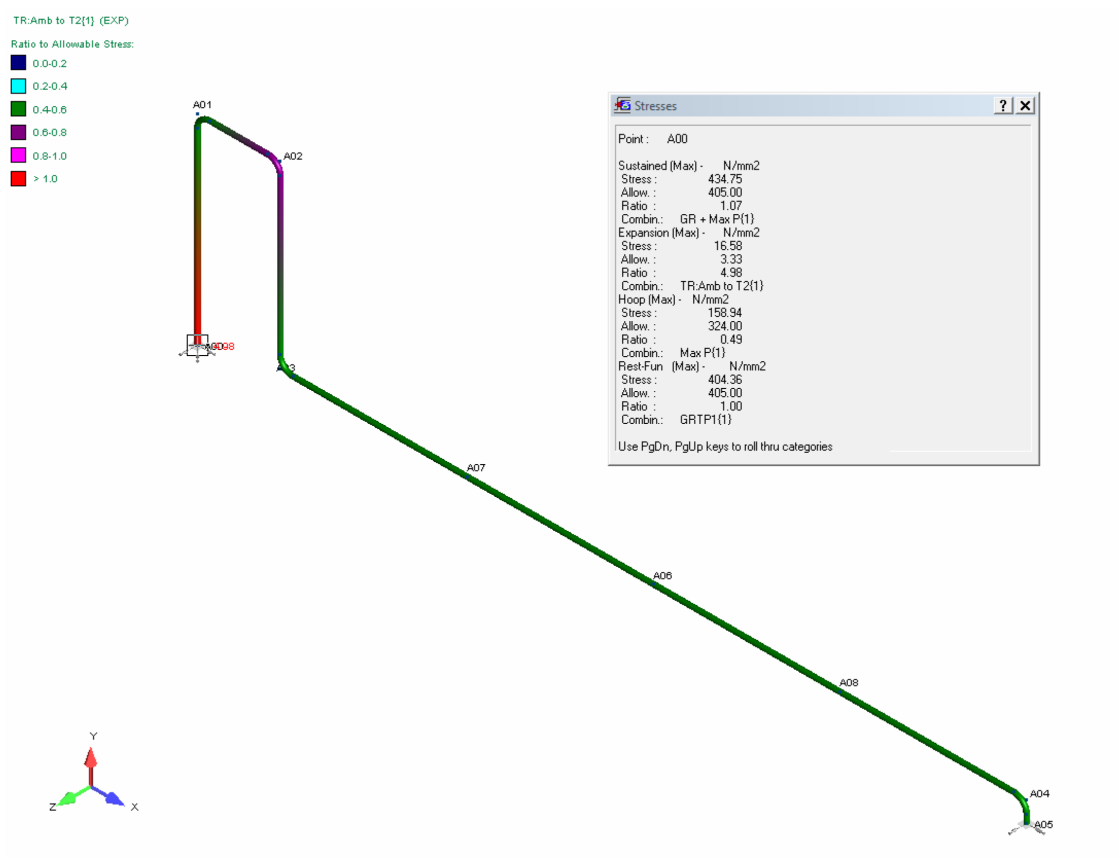


Figure 8-28 AutoPIPE spool ASME B31.8 Code stress results corroded condition

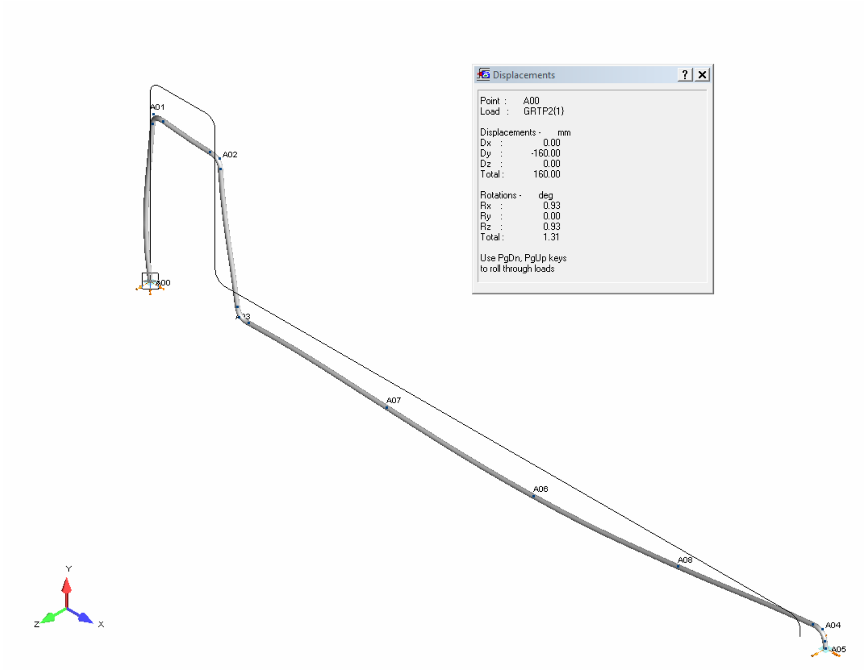


Figure 8-29 AutoPIPE spool displacement

By using the nominal wall thickness the following results are given by the program see Table 8-7

Table 8-7 AutoPIPE ASME B31.8 Code stress utilisations nominal wall thickness

Load combination	Code Stress [MPa]	Allowable [MPa]	Location	Utilisation UF
Sustained loading	379	405	MF end-(A00)	0.94
Ambient to T(1)	NA	231	NA	NA
Ambient to T(2)	15	59	Bend between leg 2 and 3	0.25
Max P(1)	159	324	XT-end –(A05)	0.49
GR+T(1)+P(1)	356	405	MF end-(A00)	0.88
GR+T(1)+P(1) Longitudinal stress S_L	355	360	MF end-(A00)	0.88
GR+T(2)+P(2)	343	405	MF end-(A00)	0.85
GR+T(2)+P(2) Longitudinal stress	340	360	MF-end (A00)	0.94

As seen from the table above the spool passes the ASME B31.8 Code check for the nominal wall thickness of the pipe. Figure 8-30 shows the utilisations towards the code limit.

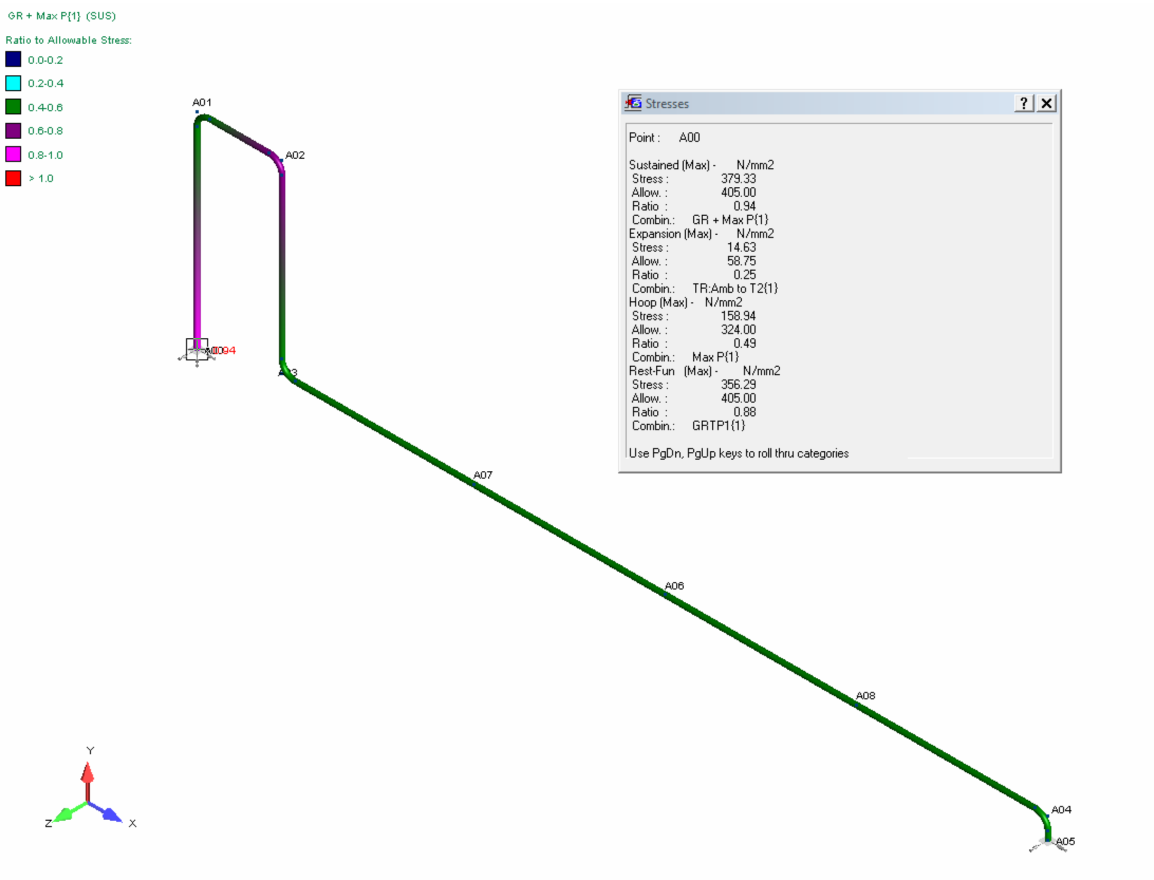


Figure 8-30 AutoPIPE spool ASME B31.8 Code stress results nominal wallthickness

General Stress Calculations-(Ref. AutoPIPE Documentation)

The General Pipe Stress Report, shown in Table 8-8 is produced by enabling the *General Stress* option. Since the bending and shear stresses vary around the cross section of the pipe, AutoPIPE looks at 15° intervals around the cross section to determine the maximum stresses. Only the total stress location is reported as a clockwise angle relative to the out-of-plane axis of the cross section. The total stress is user-specified as either the Max Shear stress or the Octahedral stress (von Mises stress).

Table 8-8 AutoPIPE General stress report

```
          G E N E R A L   P I P E   S T R E S S   R E P O R T
                                   (Stress in N/mm2  )
Point     Load                         Hoop    Longitudinal   Shear     Principal     Total
name    combination                   Stress    Max      Min  Stress   Max      Min   Stress  Loc
------ ----------------------        ------ ------ ------ ------ ------ ------ ------ ---

*** Segment A  begin ***

A00     SIFI= 1.00    SIFO= 1.00
        Gravity{1}                      0.00  341.20 -345.90   3.39  341.24-345.93  345.95   93
        Thermal 1{1}                    0.00    0.00    0.00   0.00    0.00   0.00    0.00  270
        Thermal 2{1}                    0.00   16.58  -16.54   0.00   16.58 -16.54   16.58   90
        Pressure 1{1}                 190.19   71.70   68.46   0.00  190.19  68.46  166.85  270
        Pressure 2{1}                 190.19   71.70   68.46   0.00  190.19  68.46  166.85  270
        GRTP1{1}                      190.19  409.67 -274.20   3.39  409.72-274.23  404.40   93
        GRTP2{1}                      190.19  393.15 -257.64   3.39  393.20-257.67  389.34   93
```

```
A02 N+ SIFI= 1.00    SIFO= 1.00
        Gravity{1}                      0.00  223.53 -219.27   6.07  223.70-219.44  223.78    0
        Thermal 1{1}                    0.00    0.00    0.00   0.00    0.00   0.00    0.00  270
        Thermal 2{1}                    0.00   17.99  -18.31   0.00   17.99 -18.31   18.31    0
        Pressure 1{1}                 190.19   71.84   68.29   0.00  190.19  68.29  166.87    0
        Pressure 2{1}                 190.19   71.84   68.29   0.00  190.19  68.29  166.87    0
        GRTP1{1}                      190.19  291.82 -147.43   6.07  292.18-147.54  293.36  180
        GRTP2{1}                      190.19  273.51 -129.44   6.07  273.95-129.55  278.66  180
```

The total stress report for the corroded condition is enclosed in Appendix A4.2

For comparison towards the ANSYS Pipe element model the general stress output is used. See Table 8-8 and Table 8-9. Max longitudinal and combined stress is reported for load case GR+T(2)+P(2). Reaction forces are reported for the same load case see Table 8-10.

Table 8-9 Max spool stress utilisation

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF
MF-end (A00)	389	405	0.96	393	360	**1.09**
Bend between leg 2 and 3 (A02)	279	405	0.68	274	360	0.76

Table 8-10 Reaction forces AutoPIPE

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
17.1	-14.4	1.2	4.0	-1.8	84
X-tree Side					
-7.6	-14.4	-0.8	-0.14	-4.8	37.8

8.5 Summary

The results from the different FEA computer models for the same load case and condition are presented in the table below:

Table 8-11 Summary of stress results spool analysis verification

Computer model	Combined stress S_C (von Mises) [MPa]	Longitudinal stress S_L[MPa]	Max UF	Location
Shell Element model	313	360	1.01	MF-end
Pipe Element model	352	385	1.07	Bend between leg2 /leg3
Solid Element model	337	384	1.07	MF-end
AutoPIPE General stress GR+T(2)+P(2)	389	393	**1.09**	MF-end
AutoPIPE Code stress GR+T(2)+P(2)	389	393	**1.09**	MF-end

Table 8-12 Combined stress difference- Computer models

Computer model	Shell Element model [%]	Pipe Element model [%]	Solid Element model [%]	AutoPIPE General stress [%]	AutoPIPE Code stress [%]
Shell Element model	-	12.5	7.7	24	24
Pipe Element model	12.5	-	4.5	11	11
Solid Element model	7.7	4.5	-	15	15

Table 8-13 Longitudinal stress difference- Computer models

Computer model	Shell Element model [%]	Pipe Element model [%]	Solid Element model [%]	AutoPIPE General stress [%]	AutoPIPE Code stress [%]
Shell Element model	-	7.0	6.6	9.2	9.2
Pipe Element model	7.0	-	0.26	2.1	2.1
Solid Element model	6.6	0.26	-	2.3	2.3

Commentary to results:

As seen from Table 8-12 the largest utilisation is the AutoPIPE code stress calculation with an utilisation factor of 109% towards the allowable ASME B31.8 code stress limit (9% above the limit of 405MPa). The ANSYS model's has an utilisation range between 101% - 107 %. The difference is the location of reported stress location where the pipe element model reports the max longitudinal stress at the bend between leg 2 and 3. Max bending stress is however reported at the same location (MF-end). Table 8-12 also shows that there is no difference between the calculated code stress and the general stress reported in AutoPIPE for this case.

All models however reports utilisation above the code limits for stress in corroded condition. The lowest element stress is the shell element model and the highest stress reported is the AutoPIPE beam element model with a difference of 24%. The longitudinal stress (see Table 8-13) has less difference in results with a maximum difference of 9.2 % between AutoPIPE model and the ANSYS shell model. The minimum difference found is 0.26% and is between the ANSYS solid model and the ANSYS pipe element model. The utilisation towards the code stress limits has a good match between all computer models except for the ANSYS shell model which seems to underreport the stress levels.

The spool however passes the ASME B31.8 Code check for *the nominal wall thickness* of the pipe max utilisation is *94%* for sustained loads and GR+T(2)+P(2). See results in Table 8-6

9. SPOOL WEIGHT AND LOAD MITIGATION

In order to minimize the loading from self-weight from a submerged free spanning spool, varies techniques can be performed such as:

- Buoyancy elements
- Seabed Support
- Pre-bending or mitigation of pipe

Each of the techniques is explained here.

9.1 Buoyancy Elements

The common way is to use high density polymeric foam buoyancy clamped onto the spools. Polymer foams are made up of a solid and gas phase mixed together to form a foam. The benefit of this is that the buoyancy can be mounted onto to areas of high weight. The main purpose of the buoyancy is to:

- Provide uplift buoyancy force and reduce submerged weight of the spool
- Reduce the force and bending moments into the connectors.
- Provide support for service lines
- Bending restrictor for pipelines and flexibles.

The buoyancy is clamped around the pipe with band straps which are mechanically locked.

The buoyancy has to withstand the pressure due to the water depth. This is can be a limitation for deep water application, as the weight of the buoyancy increases with the water depth see Table 9-1. This requires then that the volume of the buoyancy has to increase to sustain its weight. This usually means increasing the diameter of the buoyancy element. Drag forces increases with diameter increase and gives larger forces to the spool in the transverse direction and also increases the risk for larger VIV effects. One way to mitigate this is to apply so called “strakes” onto pipes and risers which reduces the VIV effects, See Figure 9-2 and Figure 9-3

Table 9-1 Buoyancy types versus water depth from DIAB

GRADE	Operational depth	Depth rating	Crush depth	Bouyancy	Kg/m^3
H60	30		50	965	60
H80	40		65	945	80
H100	55		85	925	100
H130	75		120	895	130
HCP30	190		300	825	200
HCP50	300		500	775	250
HCP70	450		700	725	300
HCP90	550		900	665	360
HCP100	650		1000	625	400
LD1000		1000		635	390

Figure 9-1 Buoyancy Element for piping -(Trelleborg Systems)

Figure 9-2 VIV Strakes on buoyancy-(Balmoral-group)

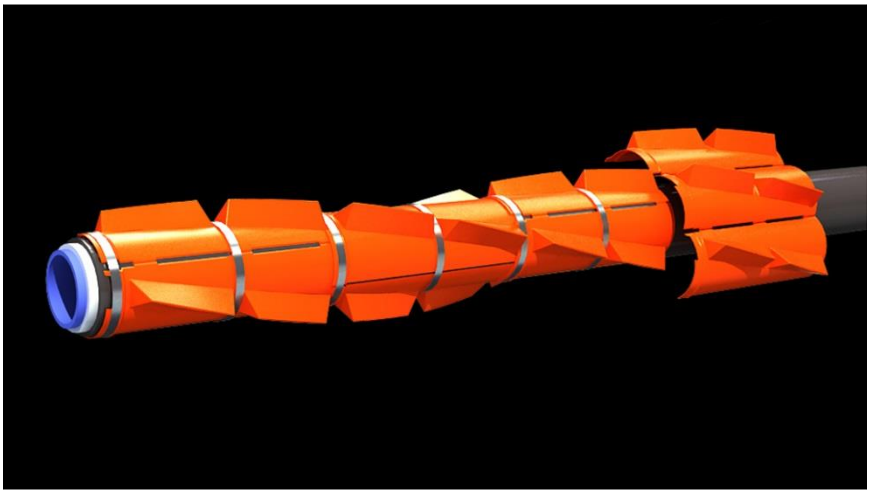


Figure 9-3 VIV Strakes for subsea piping-(Trelleborg Systems)

The effect of using strakes around a cylindrical object is shown in Figure 9-4. The figure shows a complex three dimensional flow around a bare riser Ref. *PRETechnologies.com*. The analysis shows high turbulent flow and high pressure regions, (right image) red areas shows high pressure. Figure 9-5 shows the resulting flow structures surrounding the riser with the helical strake installed. The flow is characterised by much smaller scale structures in the vicinity of the riser surface which are less correlated than in the bare riser configuration and hence less likely to lead to VIV. The riser surface pressures are also shown to be less correlated in the axial direction for the rake arrangement than with the bare riser configuration.

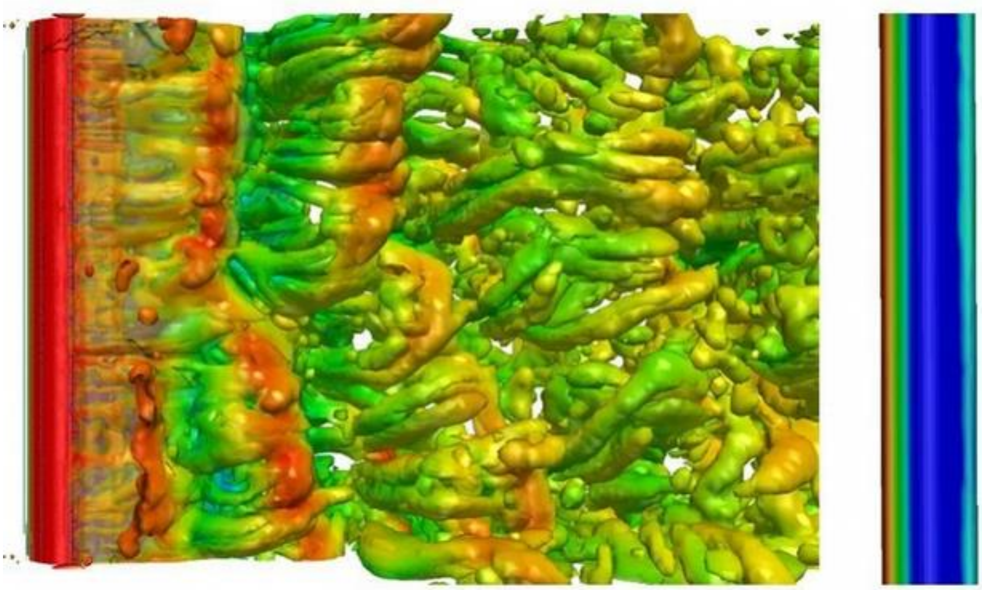


Figure 9-4 Three dimensional CFD flow around Riser (PRETech).

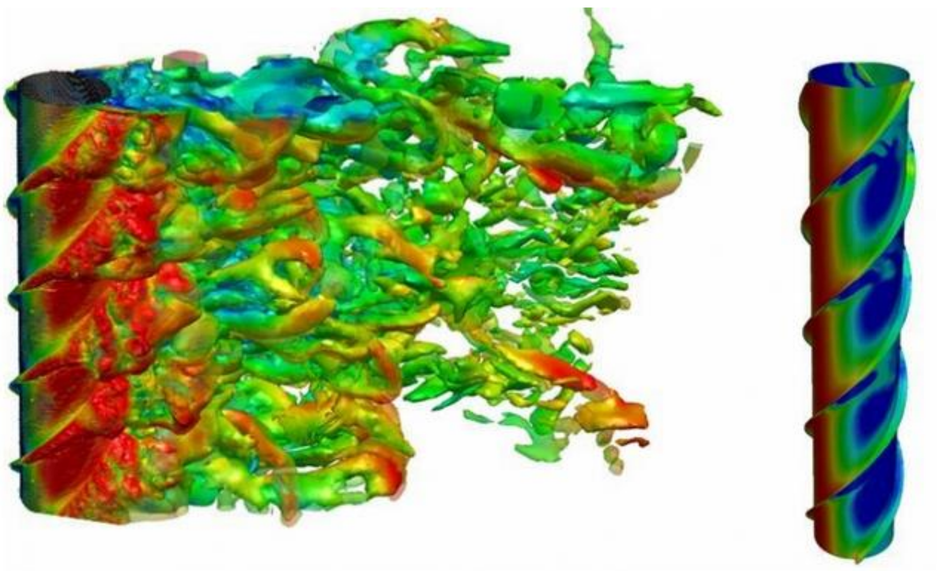


Figure 9-5 Three dimensional CFD flow around riser with strakes (PRETech).

The spool is analysed with a total calculated buoyancy uplift force of $F_{buoyancy}$=5105 N, This reduces the spool weight by 25% Ref. Appendix 3 for hand calculations

Table 9-2 Max spool stress element model with buoyancy

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF	Ref.
MF end	263	405	0.60	307	360	0.85	Figure 9-10 Figure 9-9

Table 9-3 Reaction forces with buoyancy

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
16.7	-99	1.4	4.9	-2.3	66
X-tree Side					
8.8	8.2	1.2	0.48	7.0	-9.1

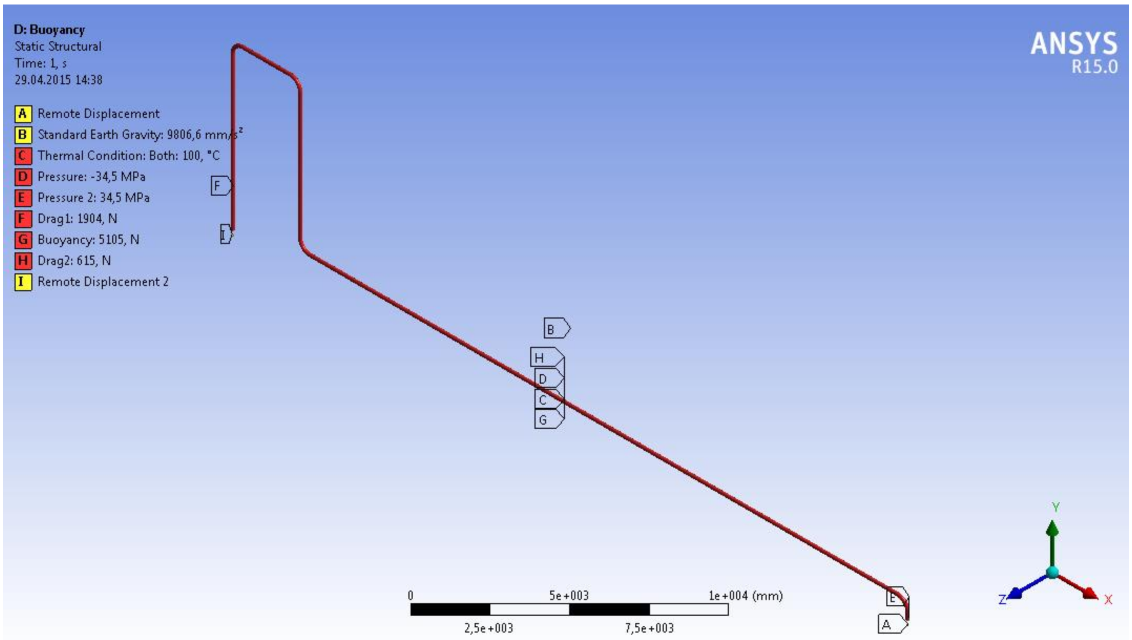


Figure 9-6 Spool with buoyancy uplift force

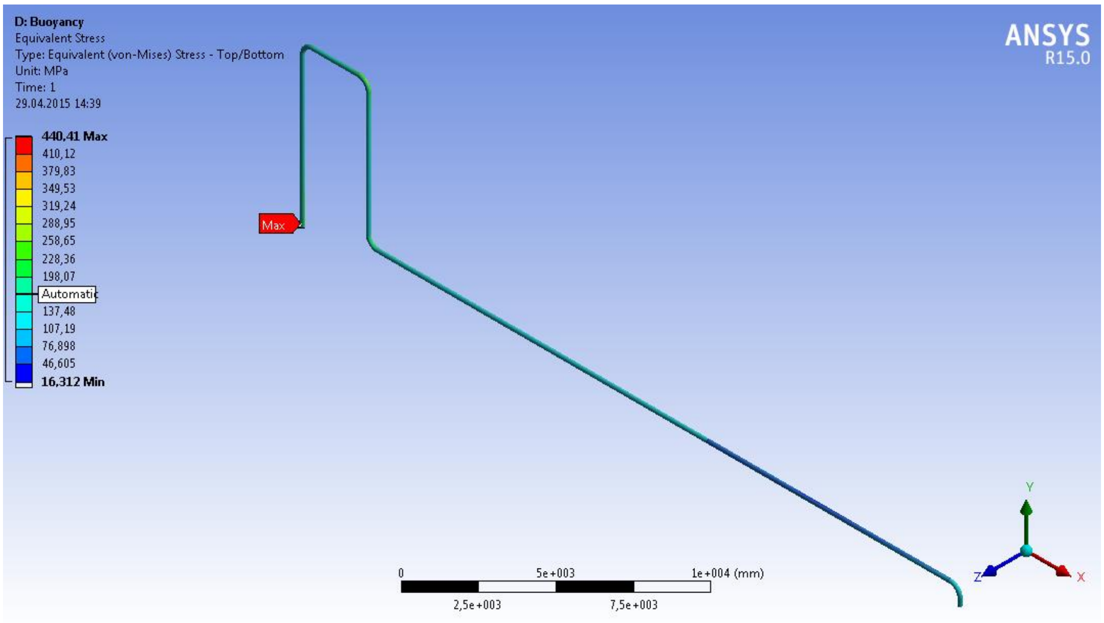


Figure 9-7 Max von Mises stress operational with buoyancy element

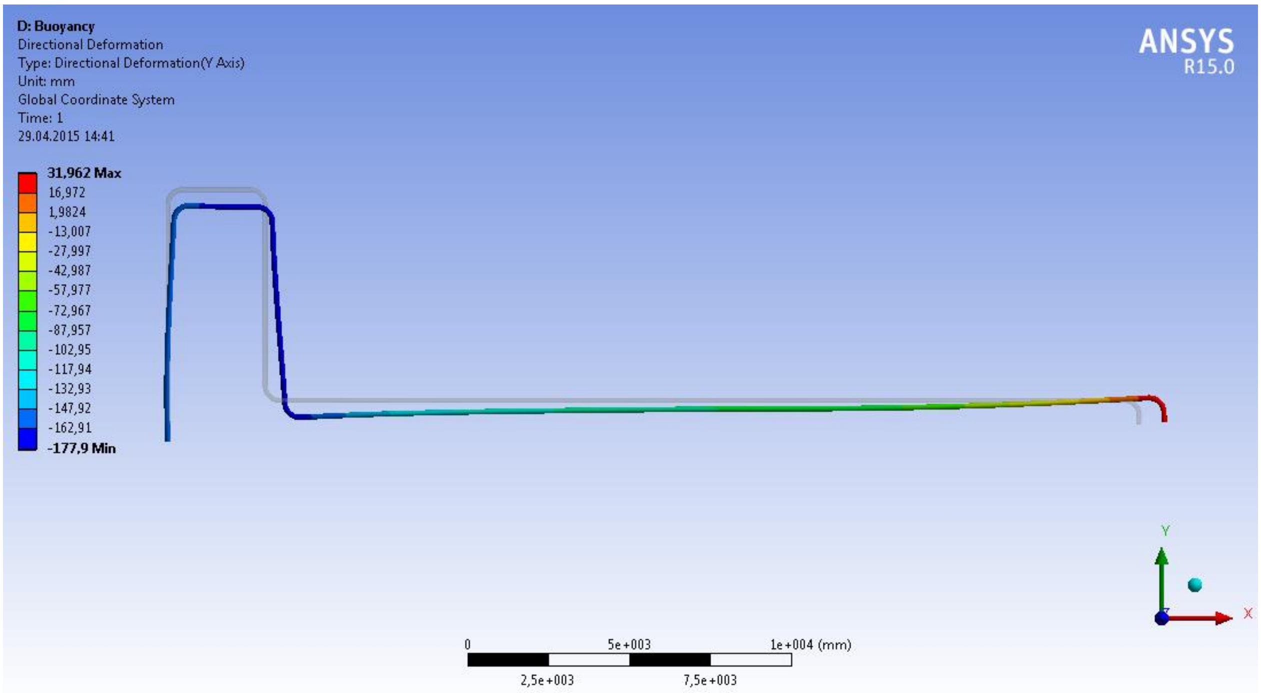


Figure 9-8 Max Displacement operational with buoyancy element

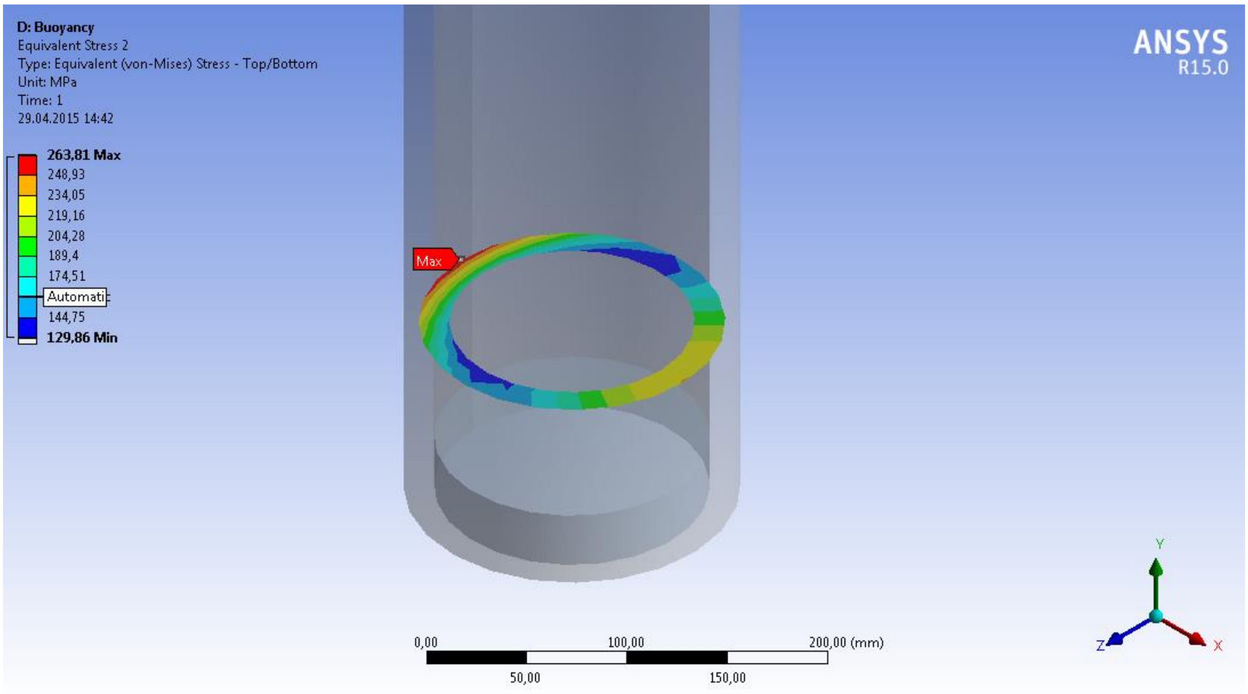


Figure 9-9 von Mises stress MF End operational with buoyancy element

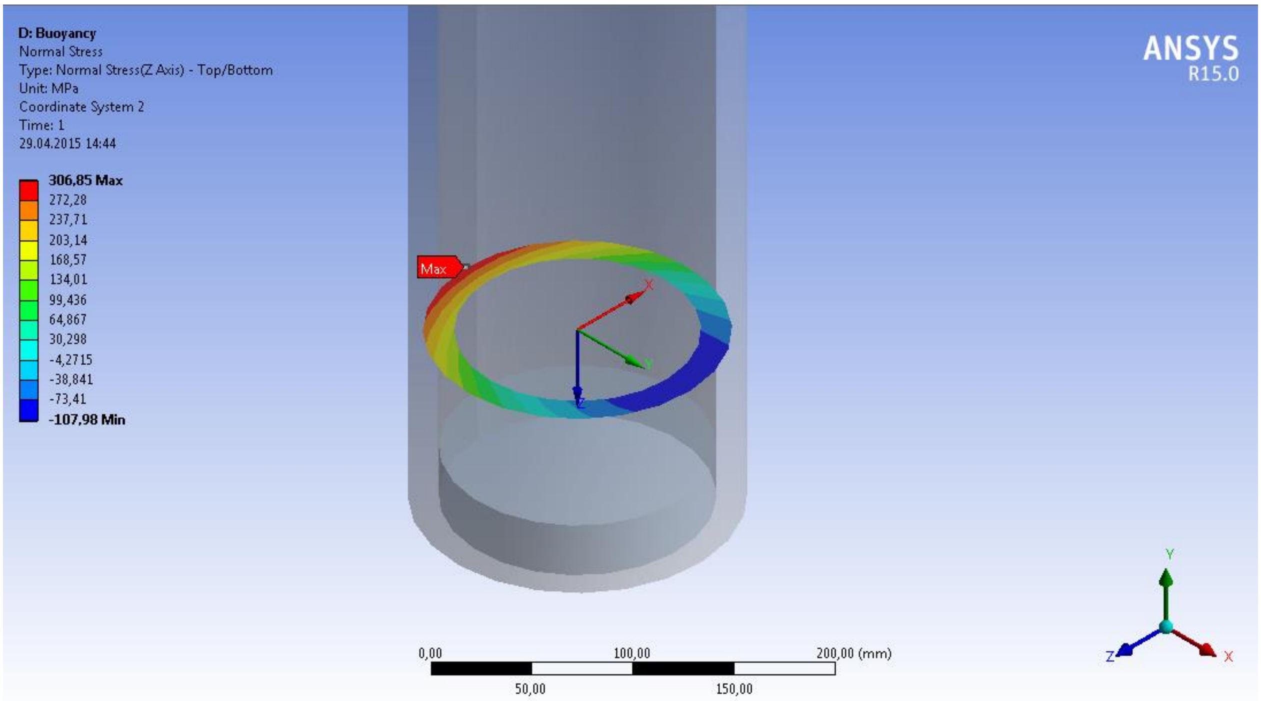


Figure 9-10 longitudinal stress MF end operational with buoyancy element

9.2 Seabed support

On way of reducing the forces and moments in the spool is to introduce a seabed mechanical spring support. This is traditionally used in topside piping (Ref. Figure 9-11 from wermac.org). This is not so common for subsea spools. A subsea spring design must be built to suit subsea environment. Such as:

- Corrosion resistant materials (Inconel-alloy spring or equivalent)
- Sliding seabed mudmat/anchored/clumpweights/suction anchor etc. onto seabed, dependent upon support condition required for spool.
- Protection from sand and debris.

During installation and handling the same amount of rigging equipment as for buoyancy mounted onto spool will be required, as the spring seabed support has to be clamped onto the spool prior to installation.

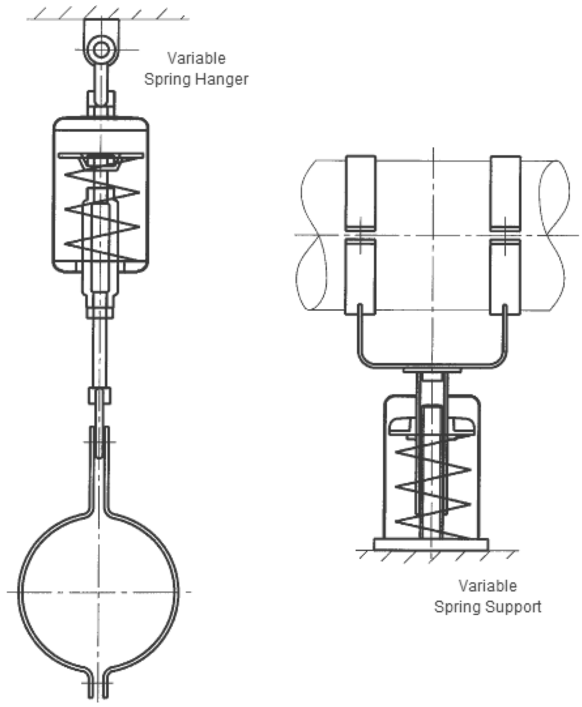


Figure 9-11 Typical piping spring support-[Wermac.org]

In the analysis of the spool a spring is positioned at the middle of the span of the spool. The stiffness of the spring is k_{spring}=100N/mm. The results are presented in Table 9-4 and Table 9-5

Table 9-4 Max spool stress element model with spring support

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF	Ref.
MF end	248	405	0.61	288	360	0.88	Figure 9-10 Figure 9-9

Table 9-5 Reaction forces with spring support

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
15.8	-100	1.4	4.1	-1.6	61
X-tree Side					
8.9	7.5	1.2	0.35	4.5	-8.2
Spring Support					
-5.8	NA	NA	NA	NA	NA

As seen from the table above a relative simple spring can achieve the same reaction forces level as for buoyancy modules, Ref.Table 9-2and Table 9-3.

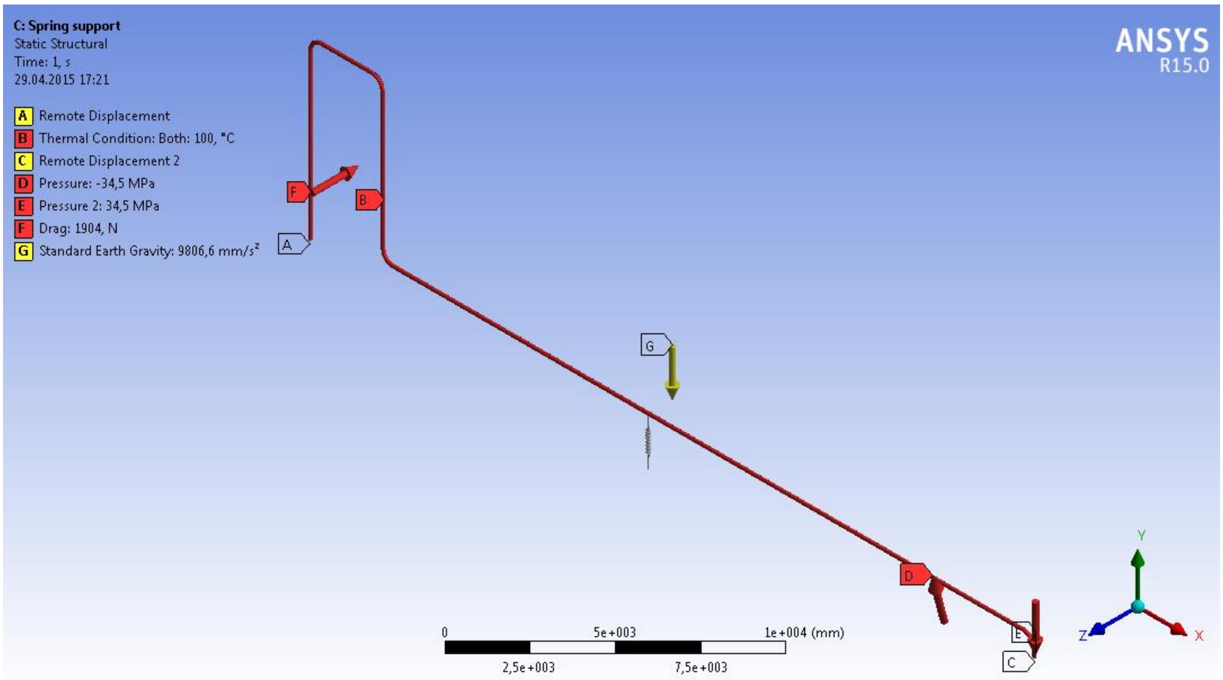


Figure 9-12 Spool with spring support loading

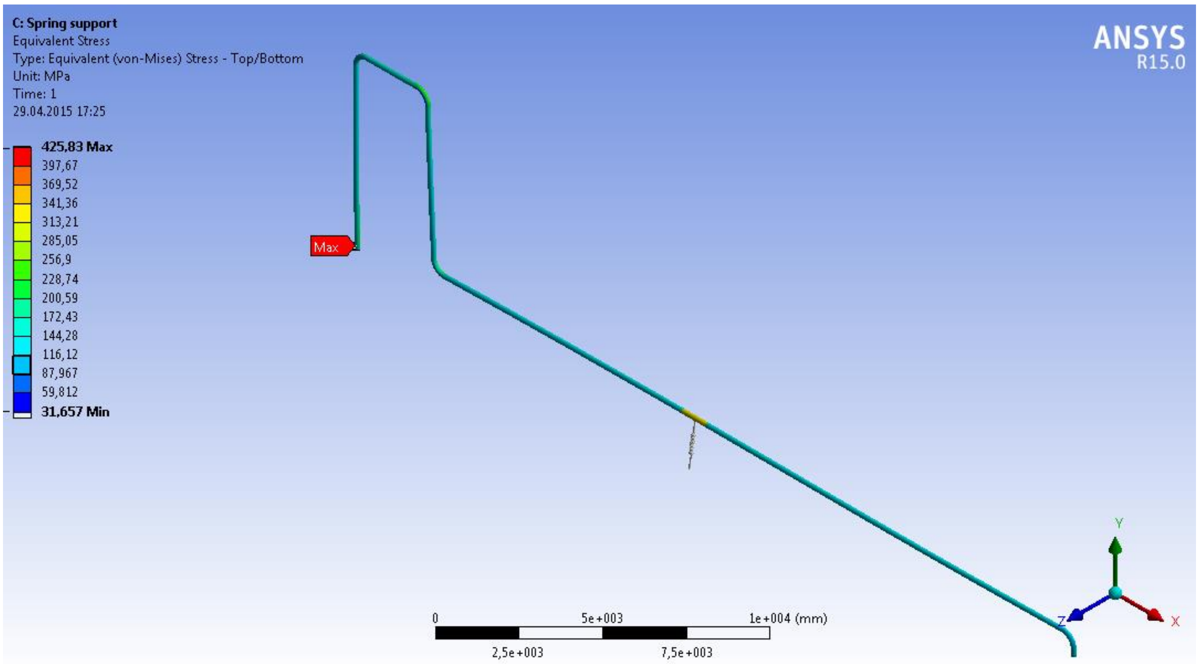


Figure 9-13 Max von Mises stress spool with spring support

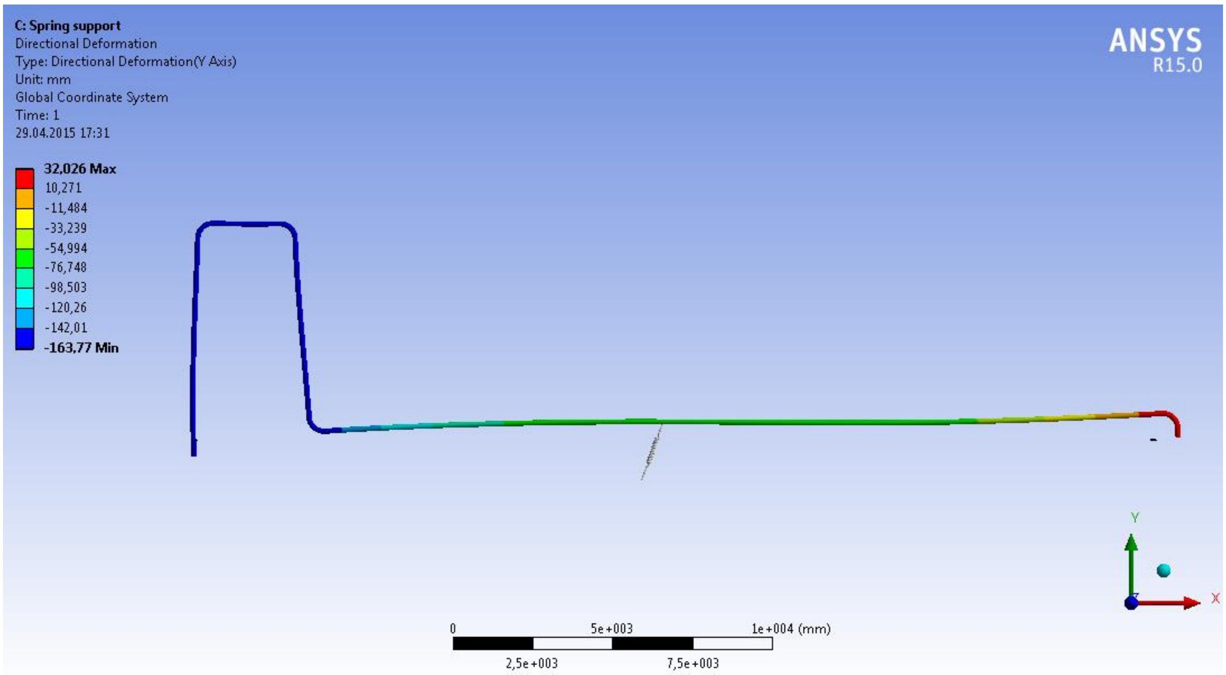


Figure 9-14 Max displacement spool with spring support

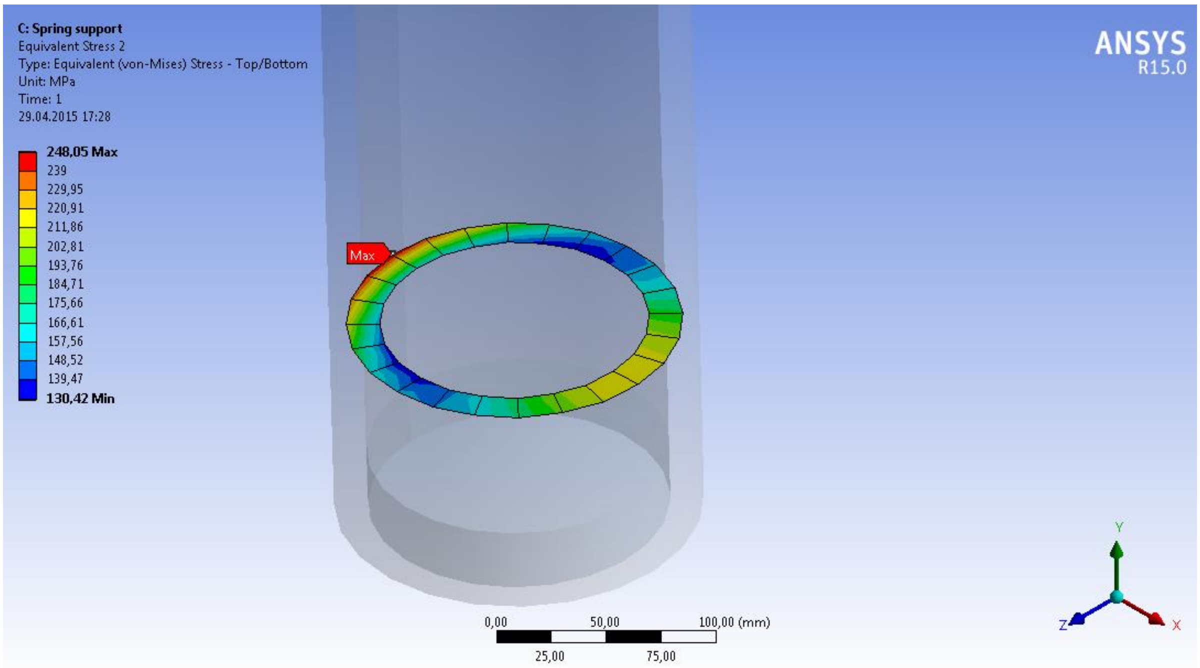


Figure 9-15 Max cross sectional von Mises stress at MF end

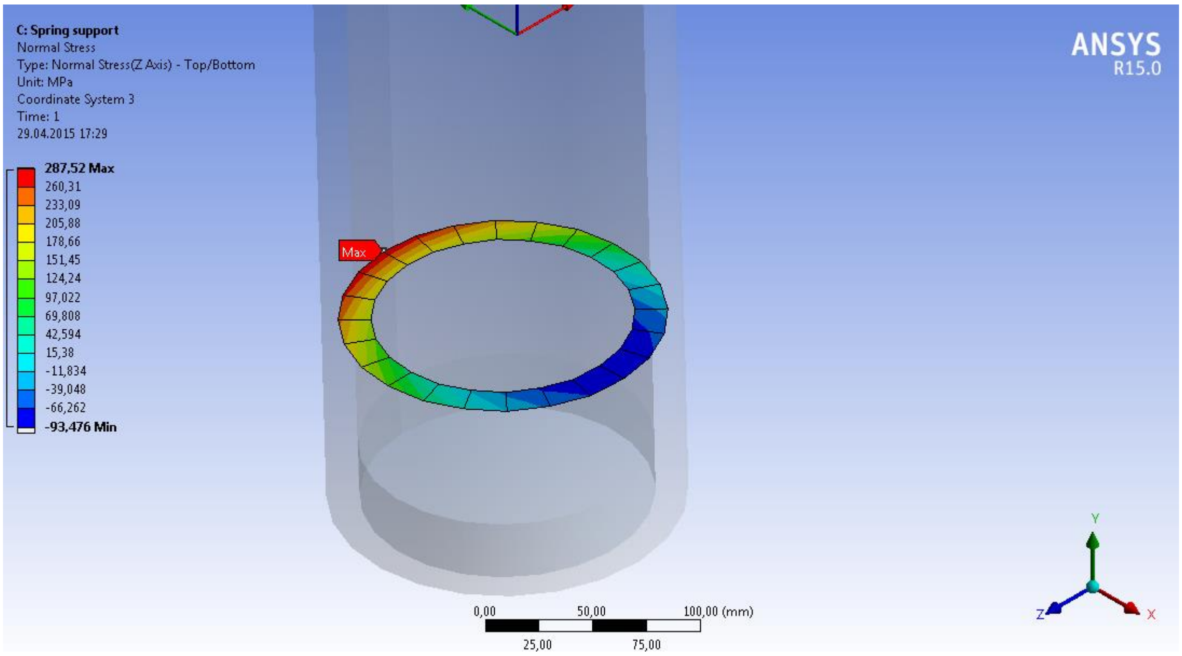


Figure 9-16 Max cross sectional longitudinal stress at MF end

9.3 Pre-bending of spool

Another way of reducing reaction bending forces in the spool is to pre-bend or initiate a positive upwards sagging of the spool. This can be achieved by mitigate the pipe at an angle of 3° which is the max allowance for bend mitigation according to ASME B31.8, the geometry is shown in Figure 9-17.

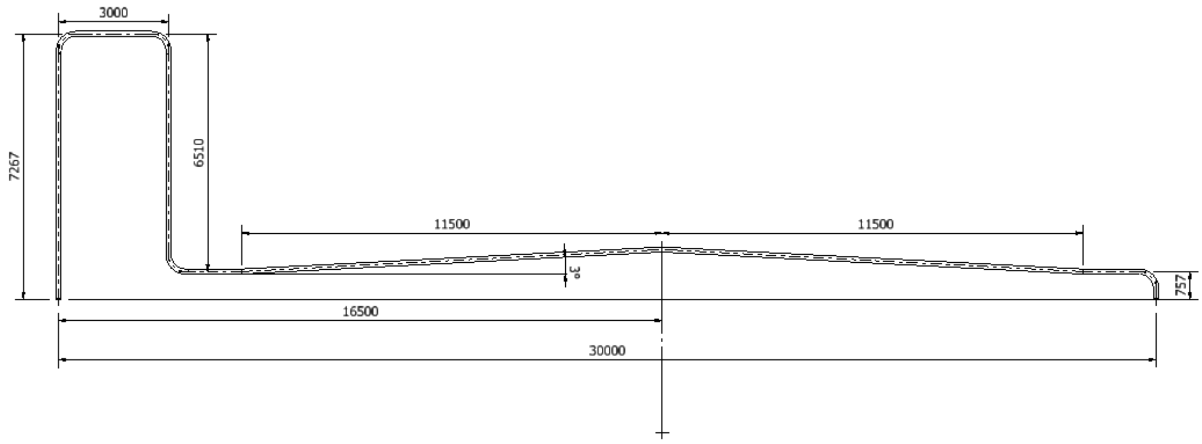


Figure 9-17 Max 1 configuration spool with pre-bending

The analysis result shows that the reaction forces have decreased for MF-end and XT tree end. The stresses however increased slightly compared to the results as presented in chapter 7.5.

Table 9-6 Max spool stress pre-bending of spool

Location	Combined stress S_C (von Mises) [MPa]	Stress limit F3 [MPa]	UF	Longitudinal stress S_L [MPa]	Stress limit F2 [MPa]	UF	Ref.
MF end	330	405	0.81	388	360	**1.07**	Figure 9-21 Figure 9-22

Table 9-7 Reaction forces pre-bending of spool

Manifold Side					
FY[kN]	FX [kN]	FZ [kN]	MX [kNm]	MY [kNm]	MZ [kNm]
18.8	32	1.1	4.3	-1.6	56
X-tree Side					
11.9	-16.0	0.8	0.7	4.6	-29

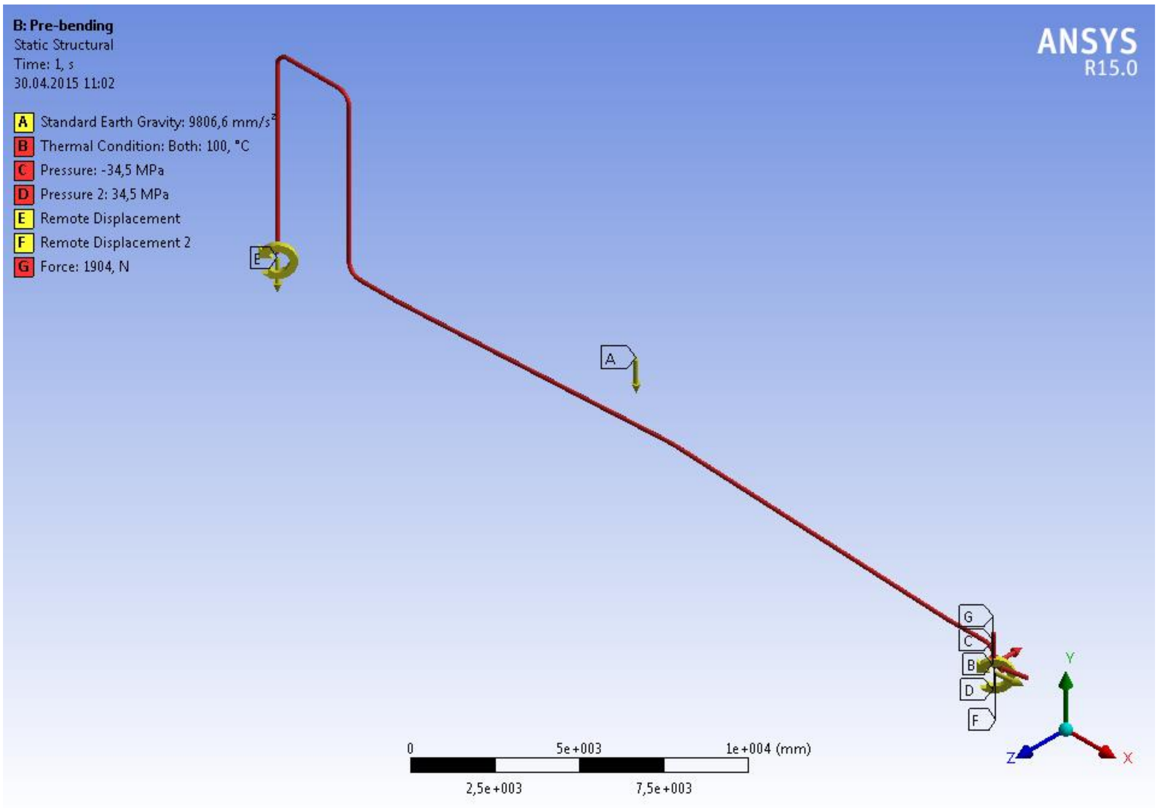


Figure 9-18 Spool Pre-bending loading

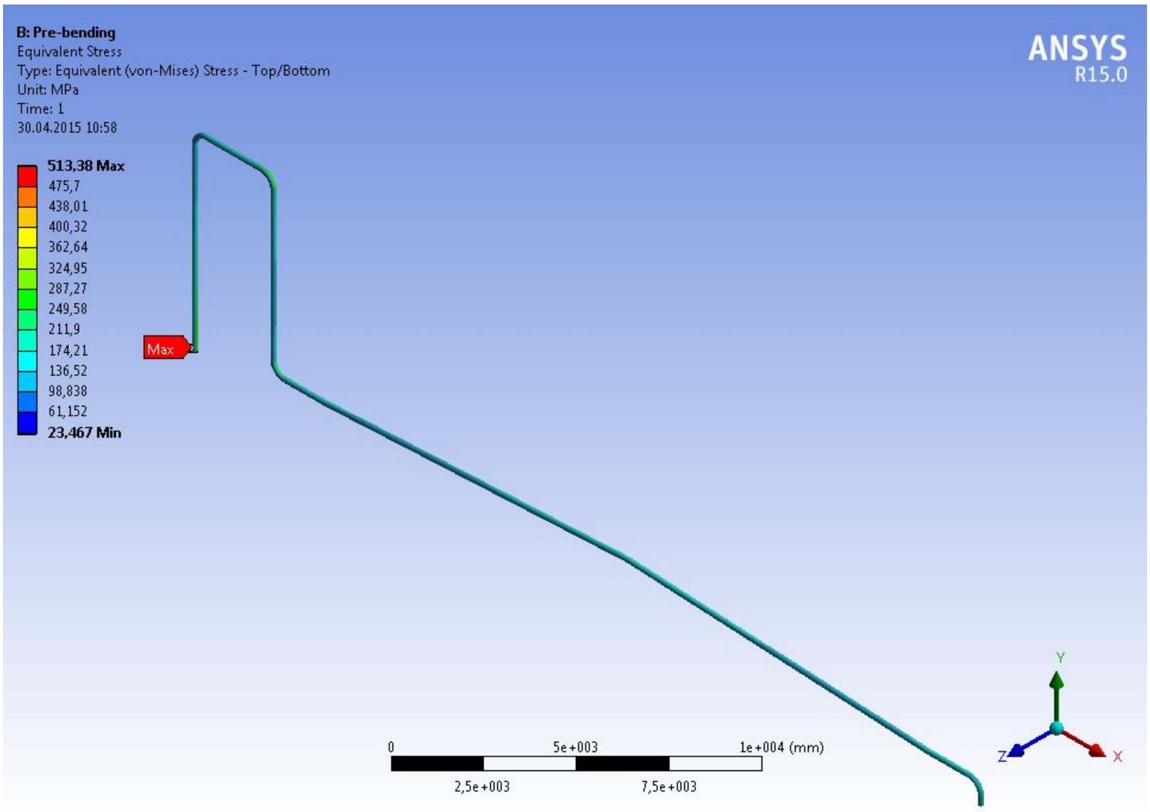


Figure 9-19Max von Mises stress Pre-bending of spool

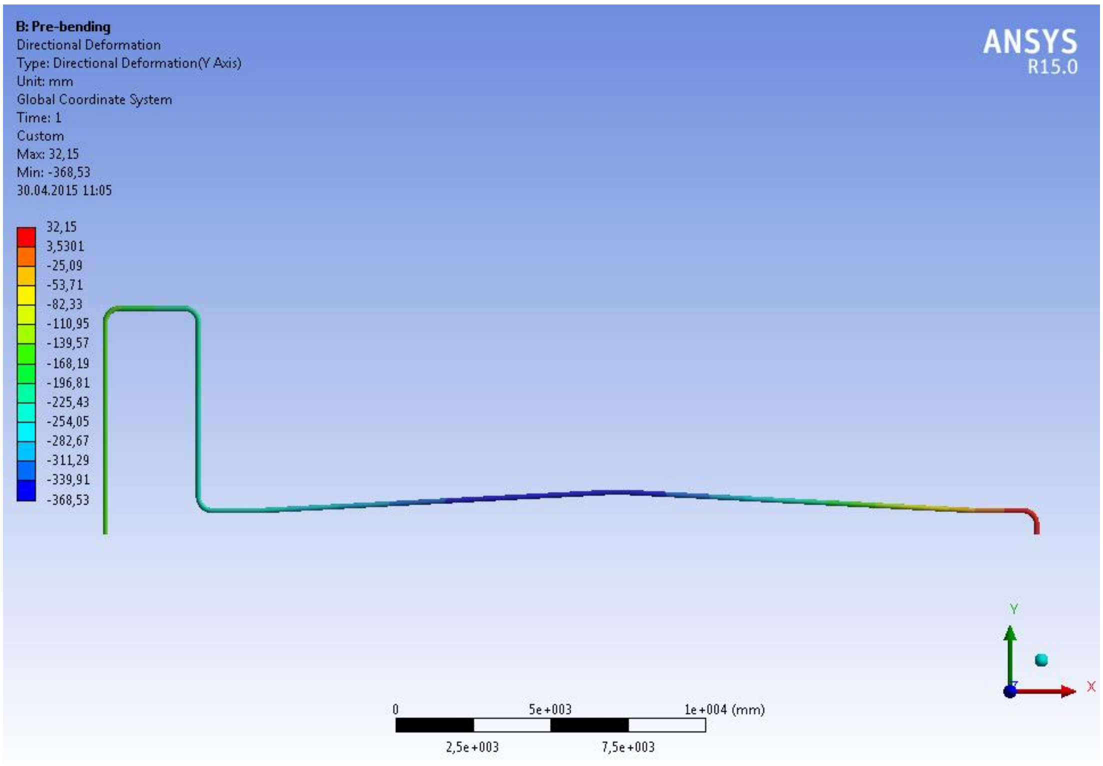


Figure 9-20 Max deflection Pre-bending of spool

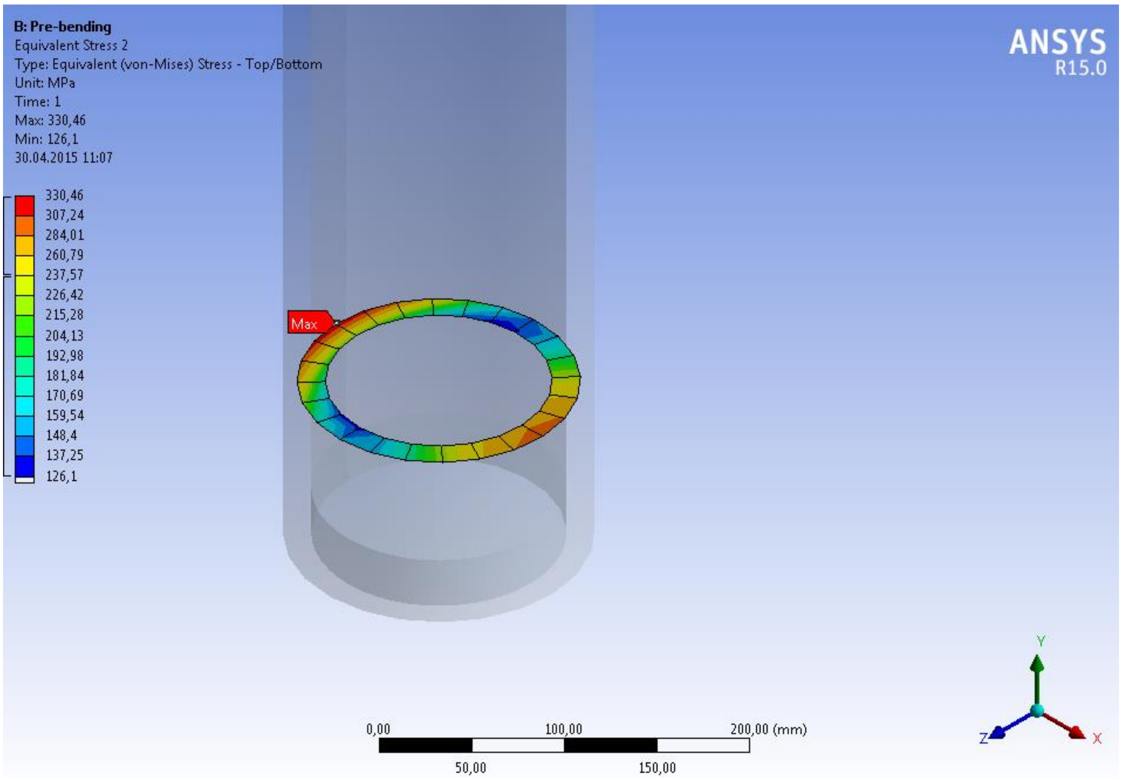


Figure 9-21 Max cross sectional von Mises stress MF end

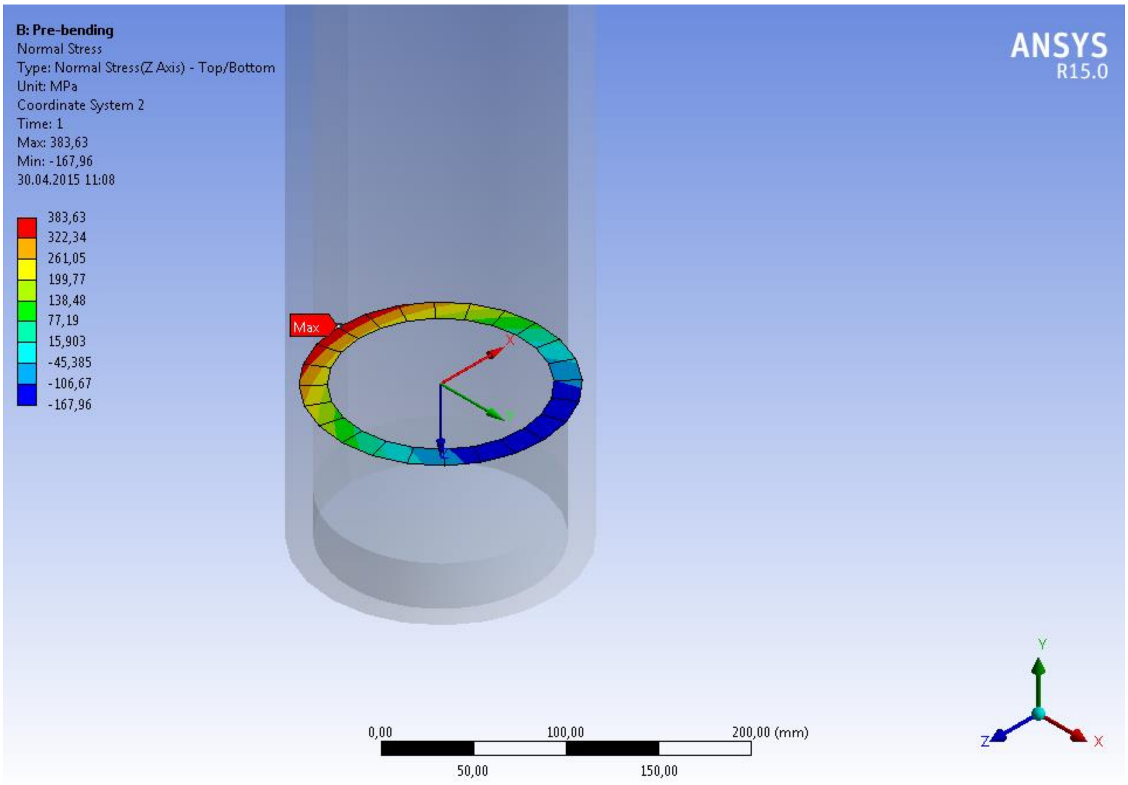


Figure 9-22 Max Cross section longitudinal stress MF-end

9.4 Chapter Summary

Table 9-8 shows the results from the load mitigation study. The result shows that by using a seabed support or a buoyancy element reduces the stresses in the spool by 25%. The largest effect was the reduction of bending moment at the *XT end with 76% reduction.* Hence this gave the best results. The pre-bending of the spool has a positive effect on the bending moments with a decrease of 30% for the bending moment on the XT tree side, but the utilisation for stresses at the manifold side was not largely effected in fact the stresses increased slightly by around~6%.

Table 9-8 Load mitigation effects on spool

Description	Max bending moment XT-end [kNm]	Reduction in bending moment [%]	Max UF
Spool without support operational case	-34.0	-	1.01
Buoyancy elements	9.1	74	0.85
Seabed support	-8.2	**76**	**0.88**
Pre-bending of spool	-24	30	1.07

10. VIV CHECK OF SPOOL

As presented in chapter 2.8 the vertical spool is a free spanning pipe and can be subjected to vibrations induced by current and waves. From chapter 5.4 the spool is installed in a deep water location so the spool is mostly subjected to current forces and not affected largely by waves as for shallow pipelines. As a result wave effects are not accounted for.

The spool is analysed by determining the natural frequency computed by the ANSYS FEM software. The 6 lowest frequency modes are computed.
Three configurations are considered:

- Max 1 configuration with buoyancy
- Max 1 configuration with a spring support
- Max 1 configuration without a spring support

10.1 Applicable codes

According to DnV-OS-F101 /7/ and the ASME B31.8 /15/ the piping system shall have adequate safety against fatigue failure within the design life. The recommended practice for free spanning pipelines is the DnV-RP-F105 /24/. This is an extensive and recommended guideline and involves a lot of time consuming computations with many parameters depending upon each other. The response models presented in this recommended practice is for a straight free spanning pipeline, see Figure 10-1. This geometry differs from a vertical spool in shape. In real the response of the vertical spool has to be analysed by use of and established hydrodynamic software such as SHEAR7 (MIT Research Institute) or equivalent. For the long horizontal span of the jumper spool there are however similarities to a long free spanning pipeline and recent projects in IKM Ocean design AS has proven to some extent the feasibility of using Ref. /24/ and compare it to results obtained by the ANSYS and SHEAR7 software.

Other earlier applicable codes is the old DNV Classification note 30.5 /25/ In this thesis due to the limited time, the method used for calculation is based upon calculating the reduced velocity and compare this to the in-line response model presented in the DNV Classification note 30.5.

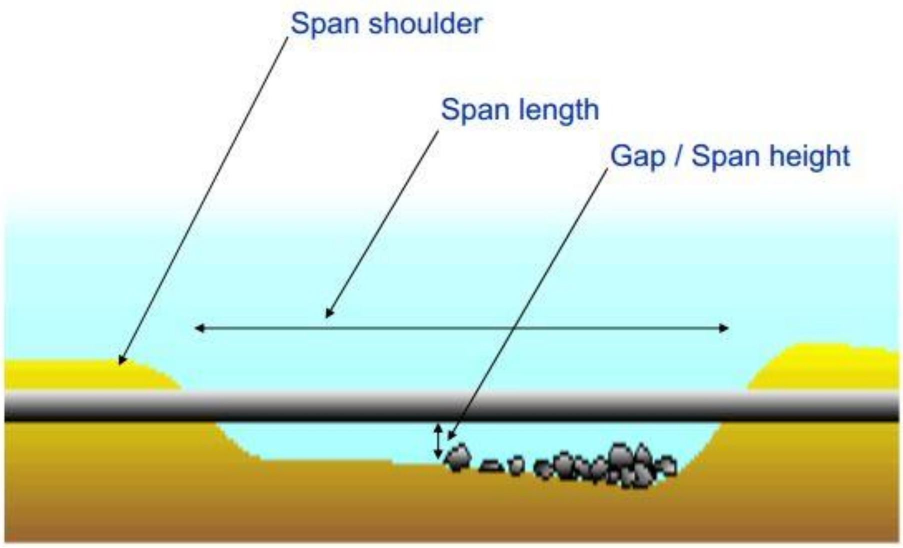


Figure 10-1 Free spanning pipeline

10.2 Modal Analysis

A modal analysis is carried out in the FE program ANSYS Workbench. ANSYS workbench uses a range of automatic solvers to establish the mode shapes. The ANSYS help documentation gives a detailed explanation about this. The subject is quite large and only a limited amount of the topic is given here:

For Static Structural and Transient Structural analysis types, by default, the Solver Type property is set to Program Controlled, which lets the program select the optimal solver. However you can manually select the Direct or Iterative solver. The Direct option uses the Sparse solver and the Iterative option uses the PCG or ICCG (for Electric and Electromagnetic analyses) solver. See the Help for the EQSLV command in the Mechanical APDL Command Reference for more information about solver selection.

For a modal analysis, additional solver type options are available and include:

- Unsymmetric
- Supernode
- Subspace

The Direct, Iterative, Unsymmetric, Supernode, and Subspace types are used to solve a modal system that does not include any damping effects During a Modal analysis, the Direct solver uses the Block Lanczos extraction method. The supernode solver is recommended for extracting a large number of modes.

Ansys Workbench has the feasibility of performing a static analysis, and then transfer's the results to a modal analysis. Damping effects can be incorporated in the analysis by setting the damping control values in the analysis setting or specify values in the materials setting. Here it is important to read the ANSYS help documentation in order to check some basics on how the software calculates the natural frequencies
In general the natural frequencies from an undamped free vibration can be found by solving the eigenproblem in the following form:

$$|[K] - \omega^2[M]| = 0 \tag{10.1}$$

Where:
[K] is the stiffness matrix
[M] is the mass matrix
ω^2 is an eigenvalue, and ω is a natural frequency.

The stability factor K_s. Ref. 7.1.6 /25/, plays a major part in controlling the motions of the system and is given as:

$$K_s = \frac{2m_e\zeta_T}{\rho D^2} \tag{10.2}$$

Where:
ρ = mass density of surrounding fluid
m_e= effective mass See 6.7.3 Ref /24/ and 7.1.6 /25/ .
ζ_T = Total modal damping ratio.
D = diameter of pipe.

The total modal damping ratio $\zeta_{T,}$ comprises structural damping, soil damping and hydrodynamic damping. The recommended practice gives some guidelines here, for instance the hydrodynamic damping in the lock in range is set to zero and the soil damping can vary between 0,5% to 2%. Assuming a very stiff manifold and X-tree, the damping value is assumed and set to 0.3%.

Hence (10.2) then becomes: (Ref Appendix 3 for calculation of effective mass m_e=82 kg/m)

$$K_s = \frac{2 \cdot 82 \cdot 0.03}{1026 \cdot 0.168^2} = 0.17 \qquad (10.3)$$

The added mass from the surrounding water is dependent upon the frequency and the mode shape and are included in the in-line response models and cross flow response models, see section 6.7.3 in Ref. /24/, In accordance with Ref. /24/ a simplification for the added mass can also be computed as:

$$M_{added} = \rho \pi r^2 L \qquad (10.4)$$

Where:
ρ = mass density of surrounding fluid
L=length of span or unit length

The added mass is calculated to be 23 kg/m and is included in the analysis (Ref. Appendix 3 for calculations)

The 6 first natural frequencies from the ANSYS modal analysis is listed in the tables below since they are assumed to be the critical ones. The initial pre-stressing from the remote displacements at the MF end and XT end is included as a pre-stress effect. The Pre stress effect must be included in the modal analysis because they will affect the natural frequencies. The natural frequency is for an un-damped system, except for the spring support model where a damping coefficient for the spring is included: A more accurate analysis would be to include the structural stiffness damping effect. This is taken into account in the limits used in the response models by use of K_s Ref. /24/. The purpose of this exercise is to evaluate if the spring has a positive effect on the spool.

The damping ratio of the spring is calculated based upon the following formulas, from mechanical vibrations theory:

$$\zeta = \frac{C}{C_c} = \frac{c}{2m\omega_n} = \frac{c}{2\sqrt{mk}} \qquad (10.5)$$

Where:
ω_n= natural frequency
m= mass of system
Cc=Critical damping
C= damping coefficient
k=stiffness of system/spring

The spring constant used here is k=100N/mm and the mass of the spring used is m=50kg. A typical value for springs with good damping ratio is in the range of ζ =0.2-0.4.

By solving for *c* in equation (10.5) gives the following damping coefficient for the spring used in the analysis:

$$C = 2\sqrt{mk}\zeta = 2\sqrt{0.05\frac{Ns^2}{mm}\cdot 100\frac{N}{mm}}\cdot 0.4 = 1.8\frac{Ns}{mm} \tag{10.6}$$

Note: The mass has been converted to Ns^2/mm in order to match the units under the root

The following natural frequencies are computed by the ANSYS software:

Table 10-1 Spool Frequencies-without spring support

Vertical spool without spring support		
Mode	**Frequency value [Hz]**	**Response**
1	0.63	In-line
2	0.80	Cross flow
3	1.32	In-line
4	1.53	In-line
5	1.64	Cross flow
6	3.22	In-line

Table 10-2 Spool Frequencies-with spring support

Vertical spool with spring support		
Mode	**Frequency value [Hz]**	**Response**
1	0.61	In-line
2	1.43	In-line
3	1.50	In-line
4	2.50	In-line
5	2.70	Cross flow
6	0	Damped out

Table 10-3 Spool Frequencies-with buoyancy uplift

Vertical spool with buoyancy uplift force		
Mode	**Frequency value [Hz]**	**Response**
1	0.37	In-line
2	0.63	Cross flow
3	1.33	In-line
4	1.64	In-line
5	1.77	Cross flow
6	3.92	In-line

The analysis shows that the frequencies for 1 mode is almost equal for the two models with spring support and without support see Table 10-1 and Table 10-2 . Since the spring used in the analysis acts in only in the vertical direction this seems natural. For the 2nd to 5th mode the spring support gave higher natural frequencies and for the 6th mode the spring has damped out the excitation. One important observation is that the amplitude of leg 2 and leg 3 seems to deflect at a higher value for the in-line response (mode2) with the spring support compared to the spool without any support see Figure 10-3 and Figure 10-4. The buoyancy case with an uplift force at the middle of the spool gave the lowest frequencies see Table 10-3 above.

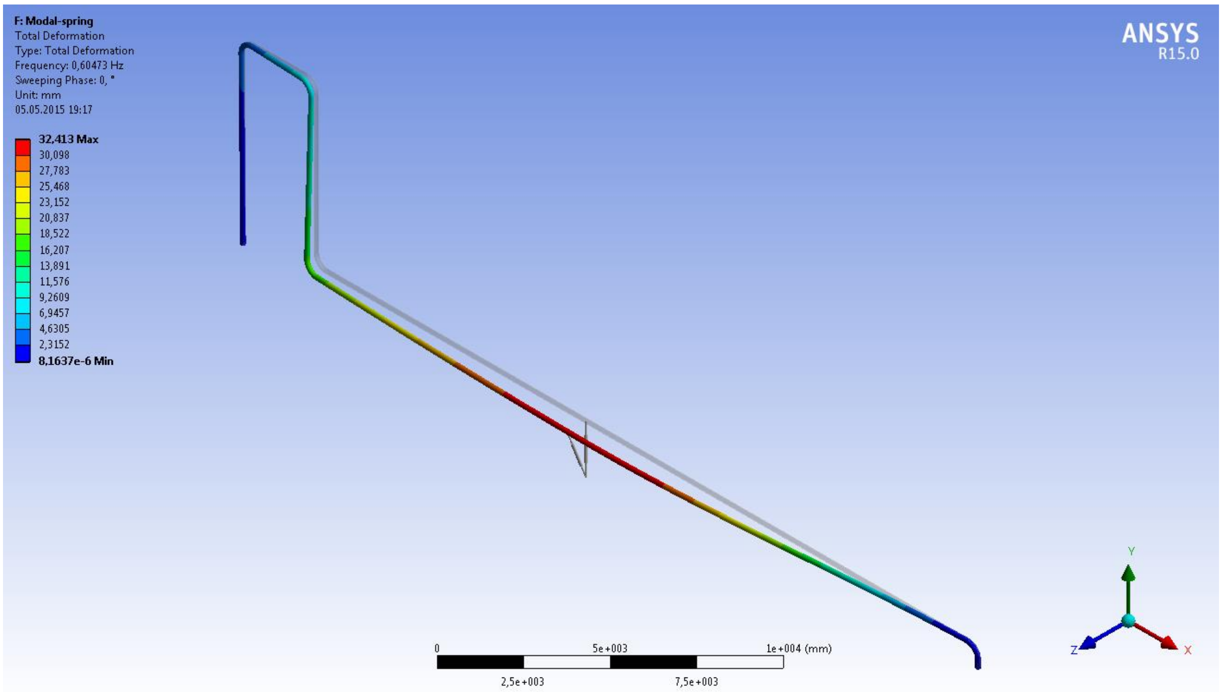


Figure 10-2 1'st mode frequency-spool with spring support

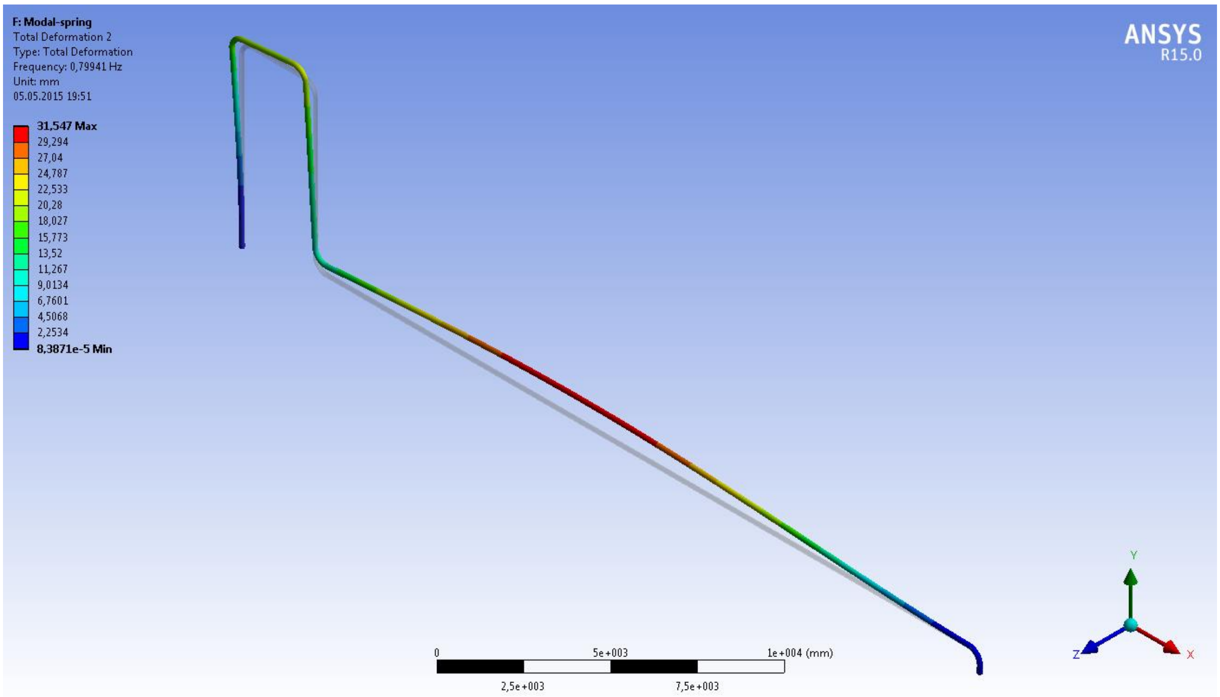


Figure 10-3 2nd mode frequency -spool without spring support

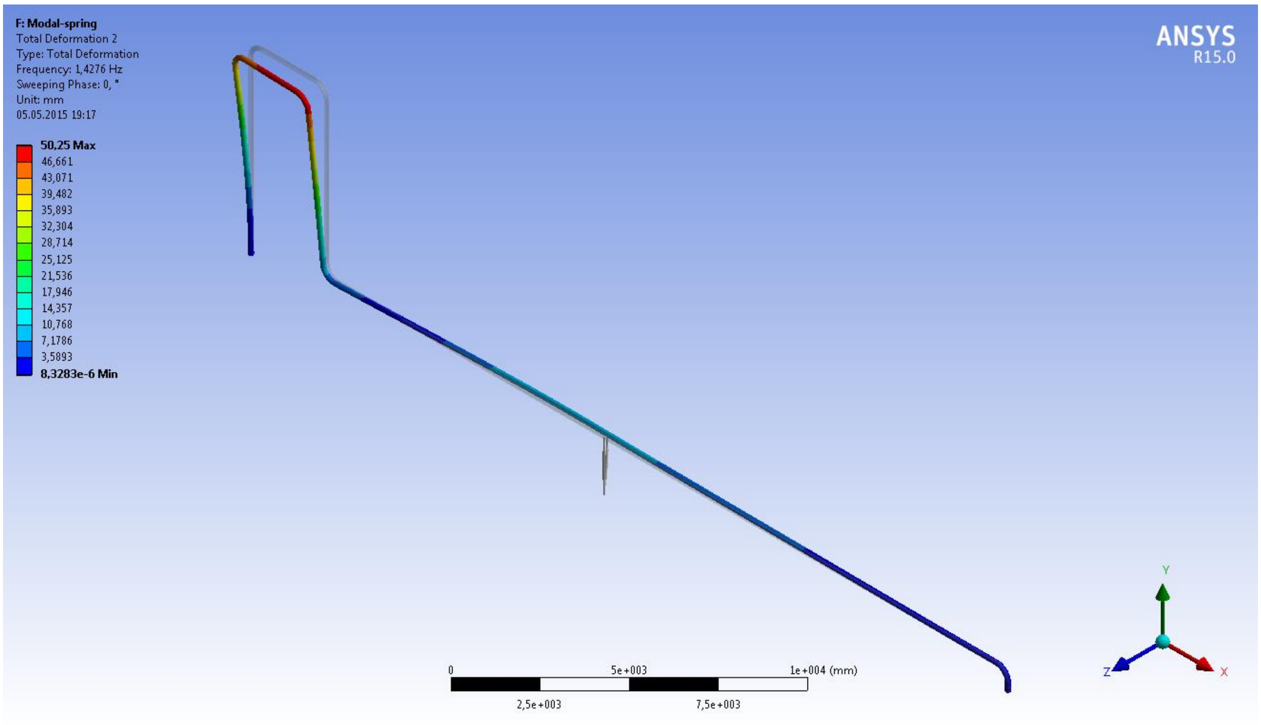


Figure 10-4 2nd mode frequency- spool with spring support

10.3 Code check Vortex Induced Vibrations (VIV)

A code check of the max 1 spool configuration supported with spring is checked here in accordance with/25/

According to this code the reduced velocity V_r shall be used to determine the velocity ranges where the vortex shedding induced oscillation may occur. The spool is located at height above seabed where the velocity and turbulence only slightly vary in the horizontal direction and is not influenced greatly by the seabed.

Reduced velocity is given by equation (2.37), here the code /25/operates with different notation and U_R is replaced with V_r:

$$V_r = \frac{V}{f_i \cdot D} \tag{10.7}$$

Where:
V= flow velocity normal to the pipe axis
f_i=natural frequency
D= pipe diameter

According to 7.3.1 in reference /25/, in line (flow parallel) excitations will occur when

$1.0 \leq V_r \leq 3.5$
And the stability parameter is
$K_s \leq 1.8$

When $1.0 \le V_r \le 2.2$, we have what is called the first instability region. The shedding will be symmetrical in this region and the max amplitude will be a function of the stability parameter K_s. For the spool in our case the first in-line oscillations of f=0.61 Hz See Table 10-2, in the first instability region will occur when the current velocity V is between:

$0.11 m/s \le V \le 0.23 m/s$

Hence this range of currents speed is within the range of max current speed of 0.7 m/s given in the design basis so 1st in-line lock on frequency will occur for the spool.

For $V_r > 2.2$ the shedding will be un-symmetric, the motion will take place in the second instability region for $K_s \le 1.8$

$2.2 \le V_r \le 3.5$

So for the next in-line mode the frequency is f= 1.5 Hz and this gives the following current speed range:

$0.55 m/s \le V \le 0.88 m/s$

This current speed range is within the max current speed of 0.7 m/s so the spool is subjected to the second instability region for in line movement and the current will lock onto this mode.

Cross flow excitations may occur when:

$3 \le V_r \le 16$ for all Reynolds number

But the maximum response is found in the range:

$4.8 \le V_r \le 8$.

The first cross flow frequency is f=1.43Hz which will give a current speed range of:

$1.15 m/s \le V \le 1.92 m/s$.

For the next cross flow frequency f=2.7 Hz will give a current speed range of:

$2.18/s \le V \le 3.62 m/s$

And hence are out of range for the max current speed of 0.7 m/s

The results are tabulated in Table 10-4 on the next page

Table 10-4 Lock on current speed's for spool with spring support

Mode	Lock on current speed V [m/s]	Response
1	0.11-0.23	In-line motion
3	0.55-0.88	In line motion
2	1.15-1.92	Cross flow
5	2.18-3.62	Cross flow

By using the stability parameter K_s , which are given by Eq.(10.3) in Figure 10-5 we see that we will have in-line motion for the reduced velocities in the range of $1.0 \leq V_r \leq 2.2$ and $2.2 \leq V_r \leq 3.5$,which are current speed of 0,11 m/s to 0,88 m/s.

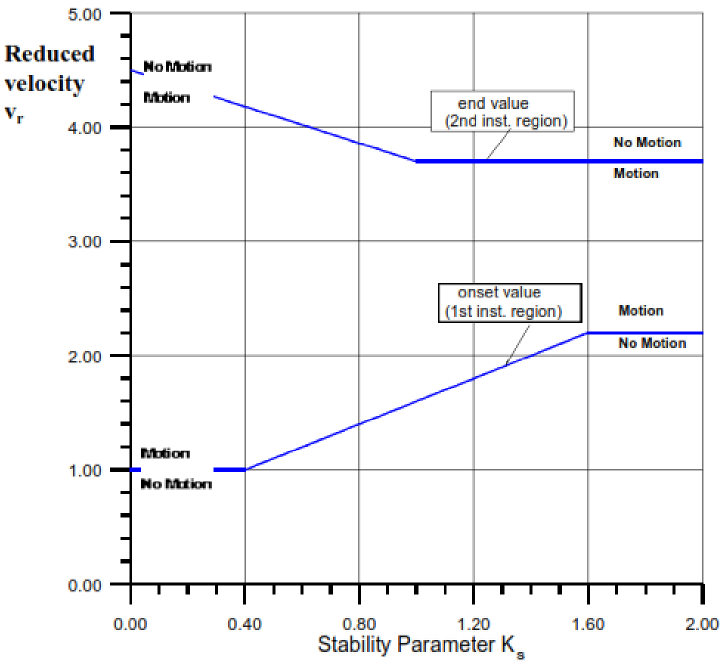


Figure 10-5 Criteria for onset of the motion in the first in line instability region (1.0<Vr<2.2 and end of second instability region Ref. /25/

Figure 10-6 shows the max amplitude to diameter A_{mpl}/D ratio for inline motions and Figure 10-7 shows the max amplitude to diameter ratio for cross flow motions as a function of the stability parameter K_s.

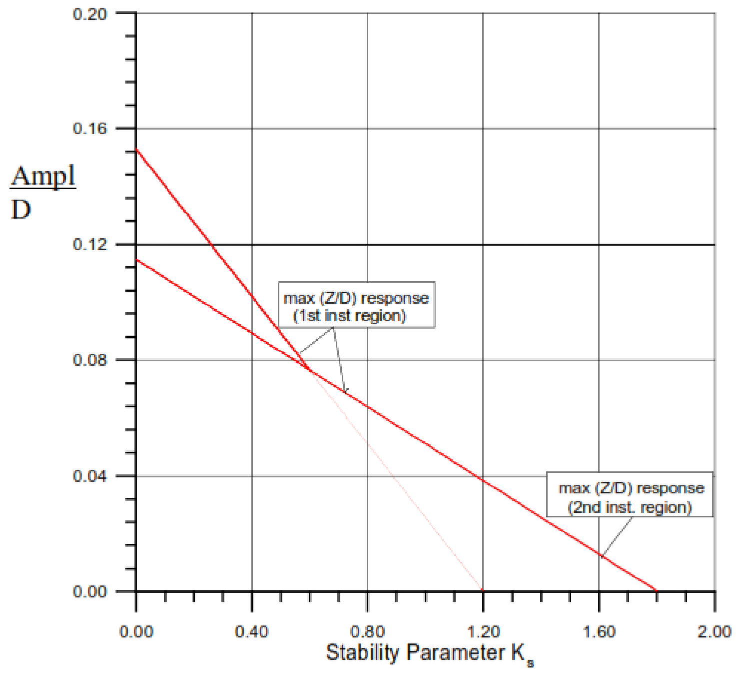


Figure 10-6 Amplitude of in-line motion as a function of K_s Ref. /25/

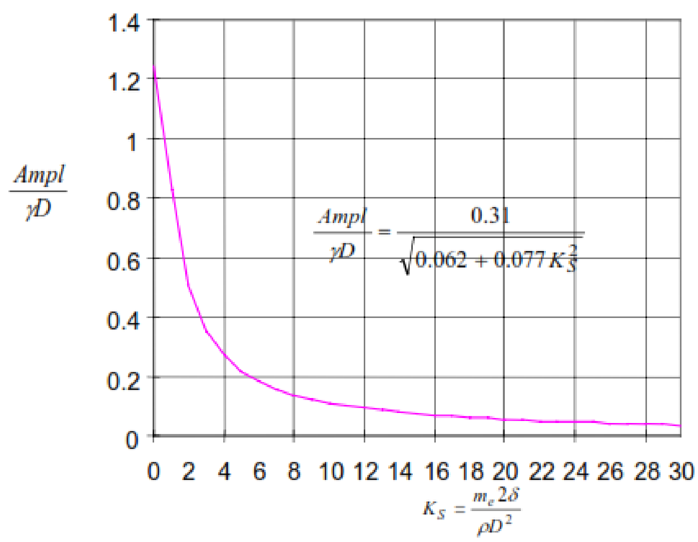


Figure 10-7 Amplitude of crossflow motions as functions of K_s Ref. /25/

Figure 10-7 has a mode shape parameter γ in the denominator of the amplitude ratio this value is listed in table 7.2 of Ref. /25/ for some typical structural elements. As seen from the figures above the cross flow motion gives the highest amplitude ratios.

10.4 Fatigue

So by applying this in a simplified fatigue evaluation of the spool due to VIV inline motion we get the following relation from Figure 10-7 with Ks=0.17 we get the following amplification ratio:

$$\frac{AMPL_1}{D} \approx 0.14 \tag{10.8}$$

$$\frac{AMPL_2}{D} \approx 0.11 \tag{10.9}$$

Where D=0.168m

The amplitude then becomes:

$$AMPL_1 = 0.14 \cdot 0.168m = 0.024\,m \qquad for\ 1.0 \leq \mathrm{Vr} \leq 2.2$$

And:

$$AMPL_2 = 0.11 \cdot 0.168m = 0.018\,m \qquad for\ 2.2 \leq \mathrm{Vr} \leq 3.5$$

The annual distribution and the current direction is assumed to be omnidirectional this means that the current is likely to come from all directions. The onset for VIV in line motion will be for all currents in the range of 0.11-0.23 m/s and 0.55-0.88 m/s with different onset frequencies. Table 10-5 shows the percent occurrence for these currents and is from based upon values from a deep water project.

Table 10-5 Current velocities percent occurrence

Current speed [m/s]	Percent occurrence [%]
0.1	16
0.2	33
0.3	29
0.4	16
0.5	5
0.6	0.75
0.7	0.14
0.8	0.11

The number of stress cycles can be taken as:

$$n = frequency \cdot \frac{sec}{hour} \cdot hours \cdot days \cdot \frac{probabillity \cdot years}{year} \tag{10.10}$$

For the first onset with f=0.61 Hz which is in range of 0 m/s to 0.2 m/s the number of cycles then becomes:

$$n_1 = 0.61 \cdot 3600 \cdot 24 \cdot 365 \cdot (0.16 + 0.33) \cdot 25 = 2.4E8$$

And for the second frequency f=1.5 Hz and in the current is in the range of 0.55-0.88 m/s with a limit of max 0.7 m/s is:

$$n_2 = 1.5 \cdot 3600 \cdot 24 \cdot 365 \cdot (0.05 + 0.0075 + 0.0014) \cdot 25 = 2.8E7$$

To get the stress amplitude, the spool has been analysed with a unit displacement of 1 meter at mid span (leg 4) and at the top of the spool (leg2) see Figure 10-8 and Figure 10-9 with fixed constraints at the ends and zero pressure and temperature.

Max principal stress amplitude for the two conditions then becomes by multiplying with the amplitude from Eq.(10.8):

$$\sigma_1 = \frac{Max\ stress}{1\ unit\ meter\ displacement} \cdot AMPl_1 = 1573 \cdot 0.024 = 38MPa$$

$$\sigma_2 = \frac{Max\ stress}{1\ unit\ meter\ displacement} \cdot AMPl_2 = 605 \cdot 0.018 = 11MPa$$

So max stress range then becomes:

$$\Delta\sigma_1 = 2\sigma_1 = 76MPa$$

$$\Delta\sigma_2 = 2\sigma_2 = 22MPa$$

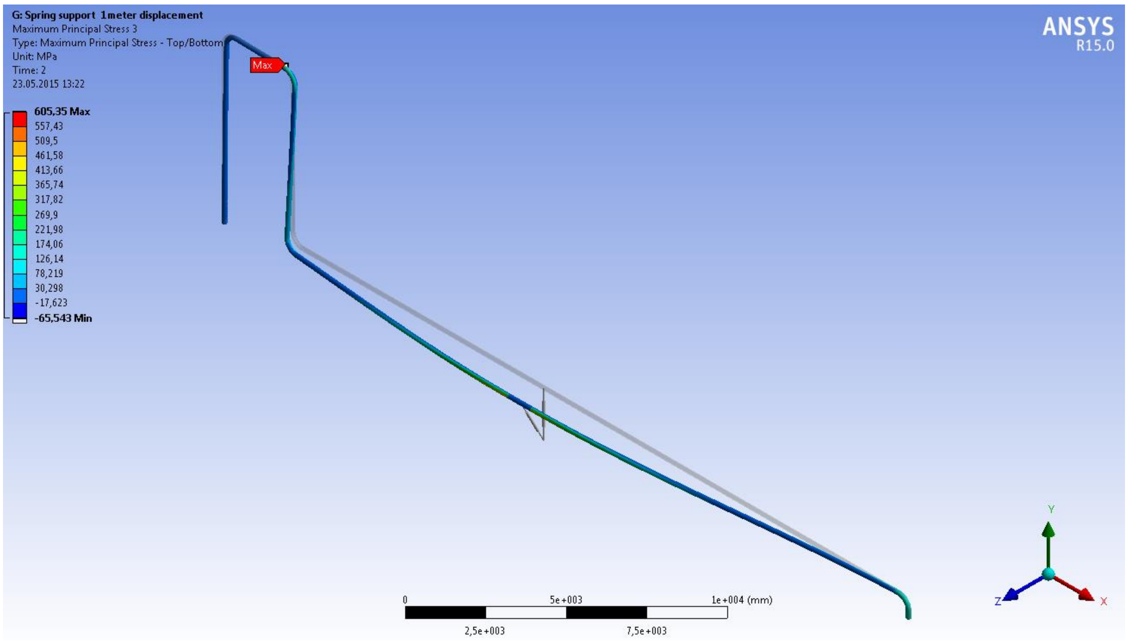


Figure 10-8 Max principal stress 1 meter displacement at leg 4

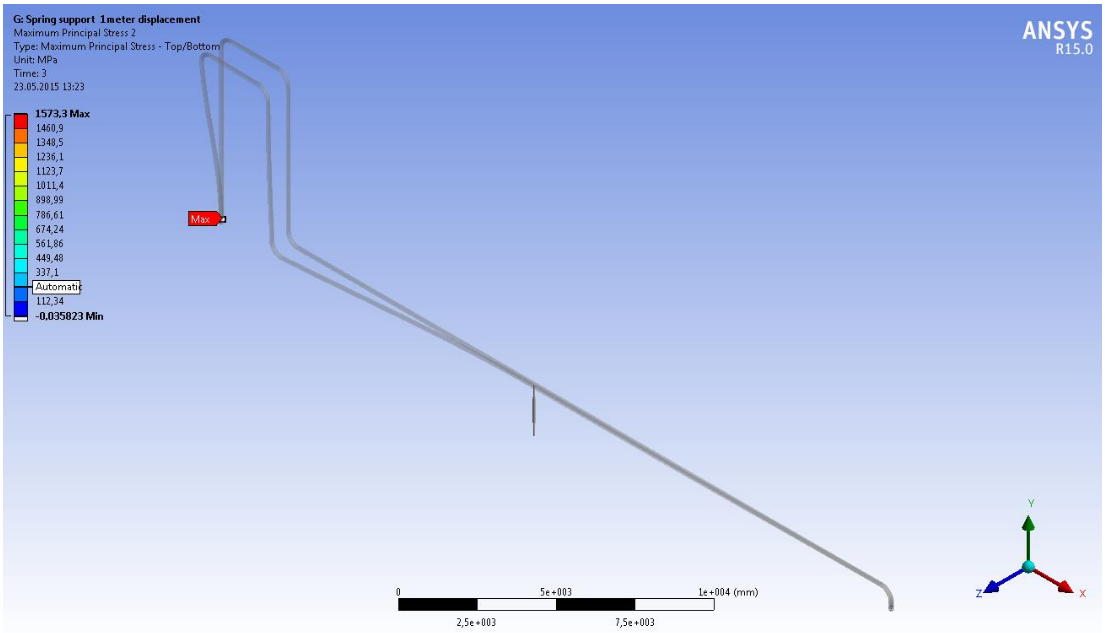


Figure 10-9 Max principal stress 1 meter displacement at leg 2

We have found the number of cycles for the lifetime and the stress range and can now compare with the S-N curves and perform a Miner summation for total cumulative damage for the spool

The fatigue life is based upon the S-N curves presented in DNV-RP-C203 Ref. /14/ section 2.2.

$$D = \sum_{i=1}^{k} \frac{n_i}{N_i} = \frac{1}{\bar{a}} \sum_{i=1}^{k} n_i \cdot (\Delta\sigma_i)^m \leq \eta \tag{10.11}$$

Where:

D= Accumulated fatigue damage

ā = Intercept of the design SN-curve with the log N axis

m= negative inverse slope of the S-N curve

k=number of stress blocks

n_i= number of stress cycles in stress block 1

N_i= number of cycles to failure at constant stress range $\Delta\sigma_i$

η= usage factor , =1/ Design Fatigue Factor (DFF)

The DFF is set equal to 10 based upon a high risk classification as given in DNV –OS-F101.

The total accumulated fatigue damage D is calculated in appendix A 3.5. The calculation shows that ***D=19>> greater than 1/10,*** and hence the spool does not have sufficient fatigue capacity for the 25 year lifetime.

VIV suppression strakes or other aids are required for the spool. The reason for failure is the long free span of the spool which is very sensitive to in line 1st mode VIV motion. Hence the vertical support should also have lateral support in order to minimize these effects.

10.5 Summary

In this chapter a VIV check have been performed based upon a modal analysis in ANSYS. The spool have been checked for different cases of support types such as a spring located at the centre of leg 4 and by use of buoyancy elements. The results show that the vertical spring support had a positive effect on the cross flow response of the spool for leg 4. The eigenvalues was increased and hence the spool became less sensitive to this motion the vibrations was also damped out for higher modes. For the inline motion there were minimal effects of the support and buoyancy. The spool was checked for a fatigue lifetime of 25years and the results shows that the spool does not have sufficient capacity. Mitigation measures such as VIV suppression strakes must be included on the spool and the effect of this must be checked by new calculations.

11. FUTURE SOLUTIONS FOR SUBSEA TIE-IN

11.1 Direct Tie-in method

This new method is presented by IKM Ocean Design AS and Statoil in a paper Ref. /30/. The paper describes how it is possible to connect a pipeline directly to the subsea structure without the use of spools. This method is manly for developed with the intention of replacing horizontal spools between pipeline and subsea structure (1st Tie in point) . it is also considered to be relevant for the 2nd Tie-in Point of the pipeline. "The background for proposing this is the reel lay method frequently used for installation of smaller diameter offshore pipelines. The installation process involves onshore fabrication of pipe sections, loading of the pipe onto the reel ship (spooling on), and finally unreeling during the offshore installation. The pipeline is normally plastically bent on the reel, and over the aligner, due to which the pipeline runs through a straightener system before leaving the reel ship, see Figure 11-1. The straightening scheme is usually a three point bending system, with the position setting of the system hydraulically adjustable. It can be quickly altered as required, e.g. by change in the pipeline outer diameter. The method is not intended to be used for short pipeline distances (<1 km) between subsea structures.

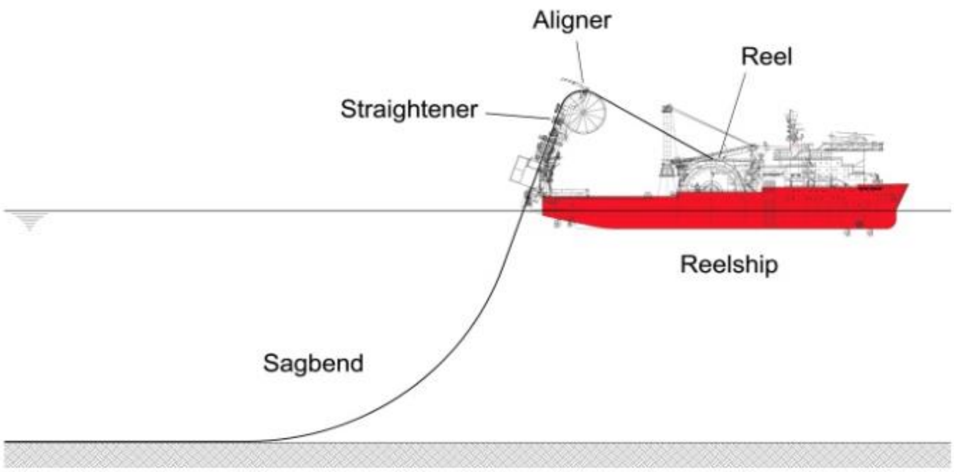


Figure 11-1 sketch of reel ship and pipeline

The quick, hydraulic adjustment of the straightener system can be utilized to create sections with residual curvature in the pipeline. This method was patented by Statoil in 2002 for thermal expansion and buckling control of reel-laid pipelines. The intention is that these residual curvature sections provide axial flexibility to accommodate thermal expansion effects.

The residual curvature method has been used successfully to control global buckling and expansion on the 14" - 16" dual diameter Skuld pipeline in the Norwegian Sea in 2012, Endal and Egeli (2014). The Skuld pipeline was installed by Subsea 7 using their reel ship "Seven Oceans". A total of 25 residual curvature sections were installed along the 26 km long route. Each section was 70 m long and had a residual strain of approximately 0.2%. Figure 11-2shows the as-laid survey data from two of these sections (KP 13.1 and 15.1)."

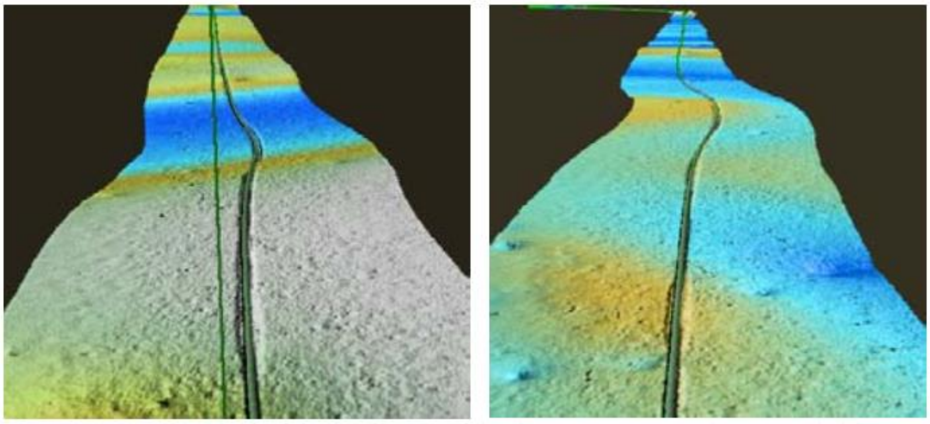


Figure 11-2 Typical Skuld Pipeline residual curvature sections

"In addition to showing the effectiveness of the method with regards to controlling global buckling, the Skuld project also demonstrated its installation friendliness.

General: The new direct tie-in method presented herein utilizes the reel-lay installation method to create local residual curvature in the pipeline, By installing such "prebent" sections at or close to the end of the pipeline, direct tie-in may be enabled without the need for tie-in spools or large start-up/lay-down areas while still achieving acceptably low tie-in and connection forces. It is believed that this approach has the potential to enable many more tie-ins by direct connection, hence permitting large cost and schedule savings for subsea development projects as separate spools or flexible jumpers become superfluous.

The same axial and lateral flexibility utilized for direct tie-in will be even more efficient when it comes to absorbing and controlling expansion forces/movements caused by pressure and temperature loads during the operating condition. In fact, pipeline expansion effects can be seen to neutralize the tie-in forces remaining in the system.
Advantages of the technique are considered to be:

- A residual curvature section can be quickly installed. Reference is made to the Skuld project where 25 pre-bent sections where installed with an average installation time of 10-20 minutes per location, Endal et al (2014).
- No straightening trials were found required, Endal et al (2014)
- Is considered a robust method even if residual strains vary.
- It will provide the means to efficiently reduce expansion forces for existing direct tie-in methods as well as spool tie-ins.

Figure 11-3 below shows the concept with implementing "tie-in and expansion loop" as an integrated part of the pipeline end section. 0.2 % to 0.3 % residual curvature strain can efficiently be created in the "tie-in and expansion loop" sections using the straightener system on the reel ship during installation.
Buoyancy and/or additional weight should be considered installed on or close to the pipeline end to ensure adequate rotation of the "tie-in and expansion loop", if required."

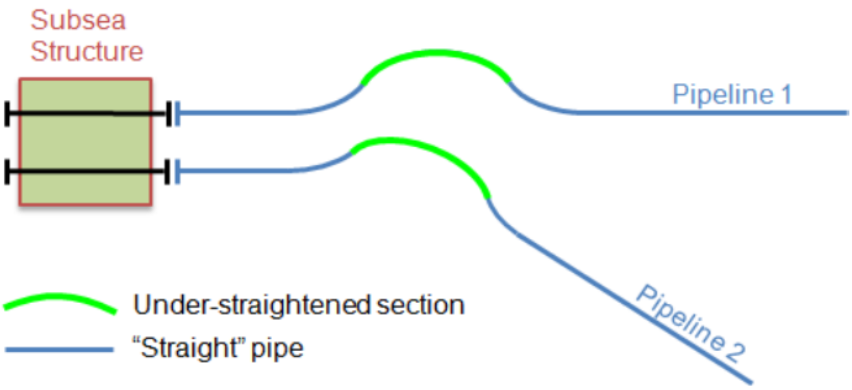


Figure 11-3 First End Direct Tie-in using the Residual Curvature Method

"A direct tie-in using the residual curvature method is considered to have more advantages at the pipelay initiation end (1^{st} tie-in) compared to the second end. At the first end, the pipeline can for instance be initiated against a return sheave arrangement on the subsea structure an lowered/docked in a controlled manner onto a guide post/landing frame etc. depending on the tie-in system being employed, see Figure 11-4 The residual curvature section is introduced approximately 100 m away from the pipeline end

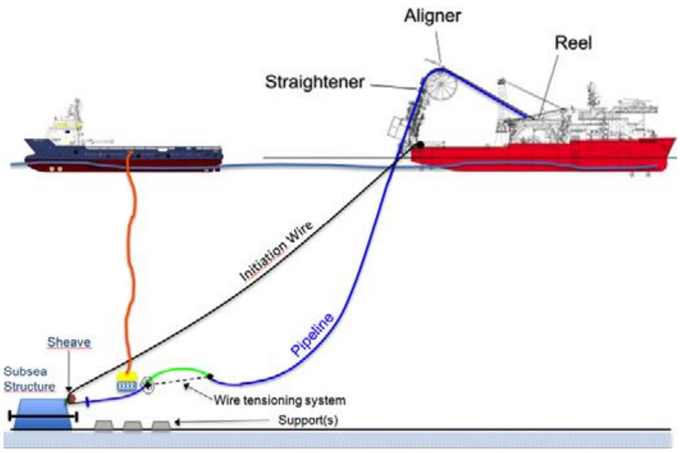


Figure 11-4 First End Direct Tie-in-Initiation overview

Hub capacity is often seen to be a governing factor for pipeline tie-ins using diver-less systems. Since hubs on subsea templates typically could be elevated 2.5 m above the seabed floor, vertical alignment between pipeline end and the hub is a key parameter. In order to compensate for this, the hub can typically be tilted slightly downwards, say 3°, 5° or 7°. However, further vertical alignment will in many cases be needed. This has on several projects been solved by introducing rock supports or adjustable mechanical supports in the adjacent free span.

After the pipeline is pulled down and safely landed in the tie-in porch/landing frame, the pipeline is ready for the final stroke-in and finally the clamp connector is made up and the seal can be tested".

"Even though the proposed method is considered more suitable for a first end tie-in, it is also considered relevant and suitable for a second end tie-in as well. Hence, the solution may be utilized generally on a project without having different solutions at first and second end tie-in points.

One feasible approach to a second end tie-in, is to install and lay down the pipeline end with a heading passing slightly on the outside of the subsea structure to be tied into with sufficient clearance, see Figure 11-5 and Figure 11-6."

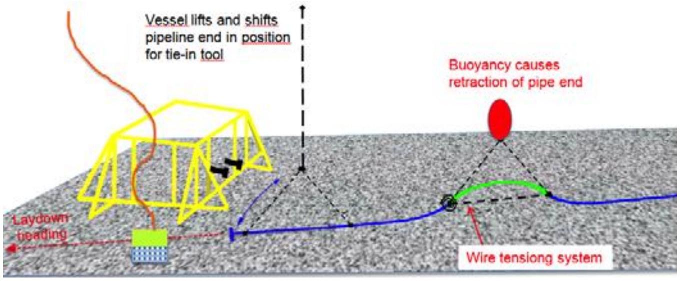


Figure 11-5 2nd Tie In laydown position, ready for Lift, Shift and Docking operation

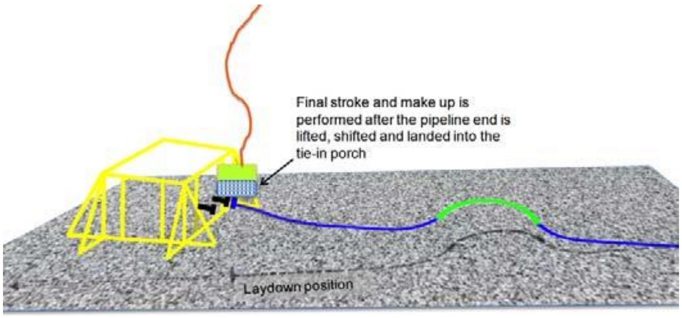


Figure 11-6 2nd end Tie-in Make up of clamp connector

The paper documents the method by use of FE model where the residual curvature is used. The paper concludes that the required stroke force is well within the capacity of applicable tie-in systems if buoyancy and/or a wire tensioning system are used see Figure 11-7 and Figure 11-8 (case *A* requires installation aids to be within the limits).

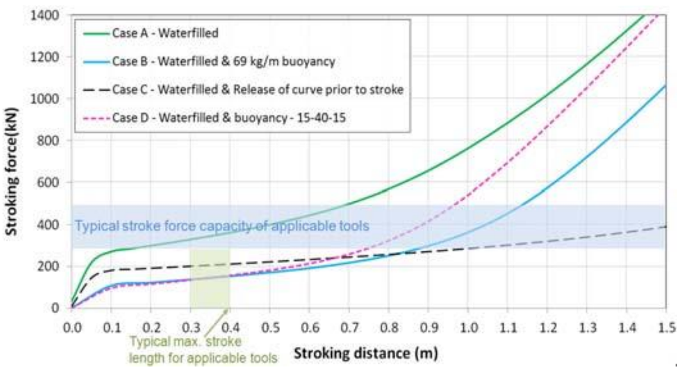


Figure 11-7 Stroke force versus distance

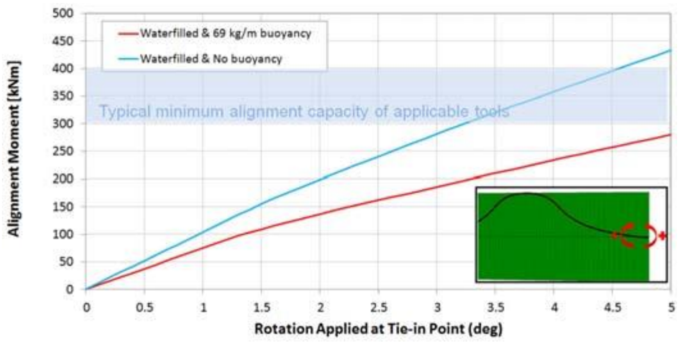


Figure 11-8 Bending Moment vs Rotation Applied at the Pipeline End

The second principal question to be checked is bending moment required for heading misalignment Figure 11-8 shows bending moment about the vertical axis as a function of rotation applied at the tie-in point. Most state-of-the-art tie-in tools have an alignment capacity of 300 kNm or more. Based on this, it can be seen that an alignment of more than 3.2° can be achieved for Case *A* (base case - waterfilled condition without buoyancy) and that more than 5°alignment can be achieved if buoyancy is used.

The paper Ref. /30/ highlights the following benefits with this method:

- All reel-lay contractors can do it.
- It also provides the means to efficiently reduce expansion forces for existing direct tie-in methods.
- The proposed method reduces the number of potential leak points in the pipeline system.
- The method allows one uniform pipeline code and criterion to be used throughout the entire pipeline system. In this paper the allowable bending moment criteria according to DNV OS-F101has been used.
- Construction vessel size may be reduced and schedule may be more relaxed (the need for spools are eliminated)
- Less parties involved (spool design, fabrication etc.).
- The proposed direct tie-in method will work very well in combination with residual curvature sections elsewhere along the route for free span reduction (Endal et al 2015).
- The method may be applicable for S-lay vessels as well.

Some of the challenges regarding this method are to achieve rotational control of the pipeline. When installing pipelines they tend to roll and twist, if the section does not roll over by a 90 deg angle and resting on the seabed the residual curvature section may be free spanning as well as being exposed to hydrodynamic loads trawl loads etc. By using the Direct Tie-in Method of the pipeline the following parameters must be considered:

- Torsional capacity of connectors
- The soil friction
- The size of the pipeline
- The length of the pipeline
- Roll angle versus touch down of pipeline

11.2 Flexible spools

Flexible spools is an alternative to rigid spools often utilised in subsea projects The main advantages of using a flexible tie in solution versus a rigid spool are the reduced tie-in forces, roomy installation tolerances, and no requirement for metrology and fabrication after pipeline/structure installation. On the negative side, industry experience indicates that flexible pipes, jumpers and tails are more vulnerable than rigid pipe solutions, and thus have a reduced lifetime compared to a rigid solution. In addition, flow assurance issues could also be a concern for flexible.

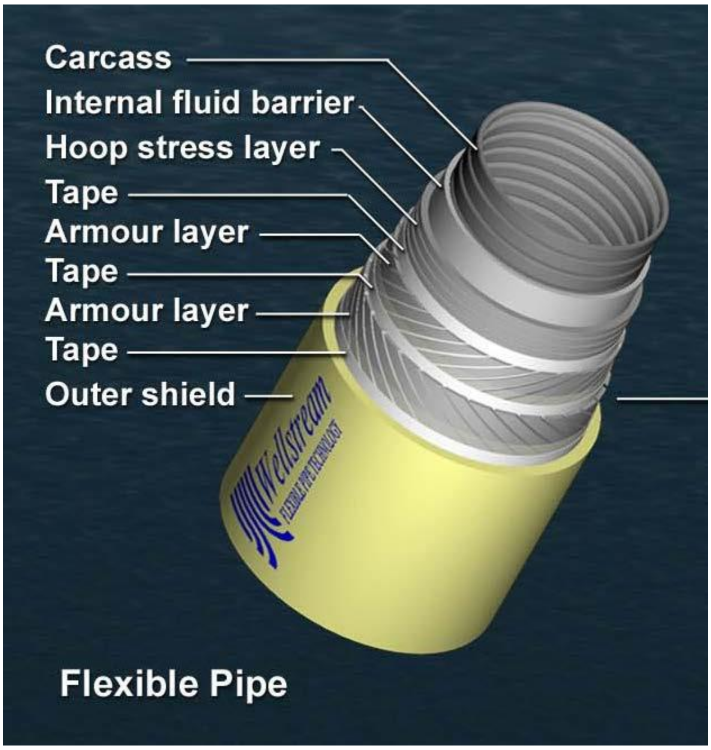


Figure 11-9 Layers of Flexible Pipe (Wellstream)

Figure 11-9 Show the typical layers of the flexible pipe. Each layer has its purpose:

- The inner steel Carcass protects against external overpressure

- The Internal fluid barrier consist of a thermoplastic sheath for sealing

- The Hoop stress layer (Zeta) spiral main function is to contain internal pressure
- Armour layer main function is to take the axial forces
- The Thermoplastic sheath (tape) main function is to reduce friction between tension layers and the zeta spiral
- The external sheet main function is to protect against outer forces such as abrasion protection

In the Norwegian offshore sector flexible pipes has been in use since the late 80's. Some of the first applications were already installed early in the 70's. The use of flexibles peaked between 1996 and 2000, the curve flattened out towards the year 2006. The Current status (2007) Ref. /31/ for the Norwegian Offshore Sector is still an increase in the total number of flexible pipes. See Figure 11-10

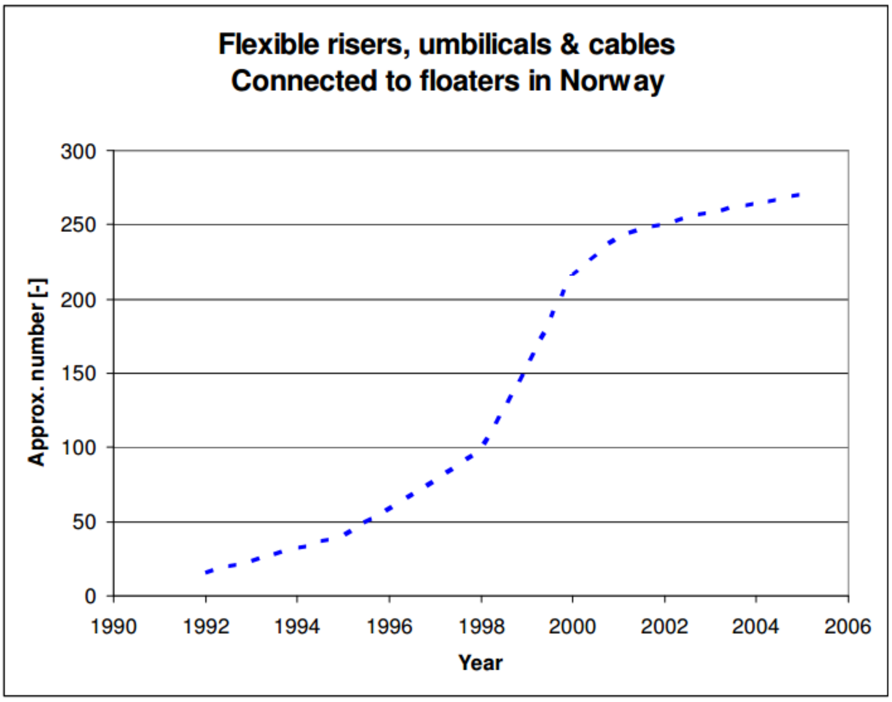


Figure 11-10 Installed flexibles in Norway Ref. /31/

For the Norwegian sector the report Ref. /31/ describes that over 200 flexibles has been installed in the Norwegian sector and the average servicetime for this has been 50% of its intended service life. However the technology has given cost effective solutions for a large number of field developments and the usage of flexibles is still growing. As of today there are several hundred kilometres of installed flexibles flowlines in the North Sea. Figure 11-11 gives an overview of the flexible riser incidents until 2007

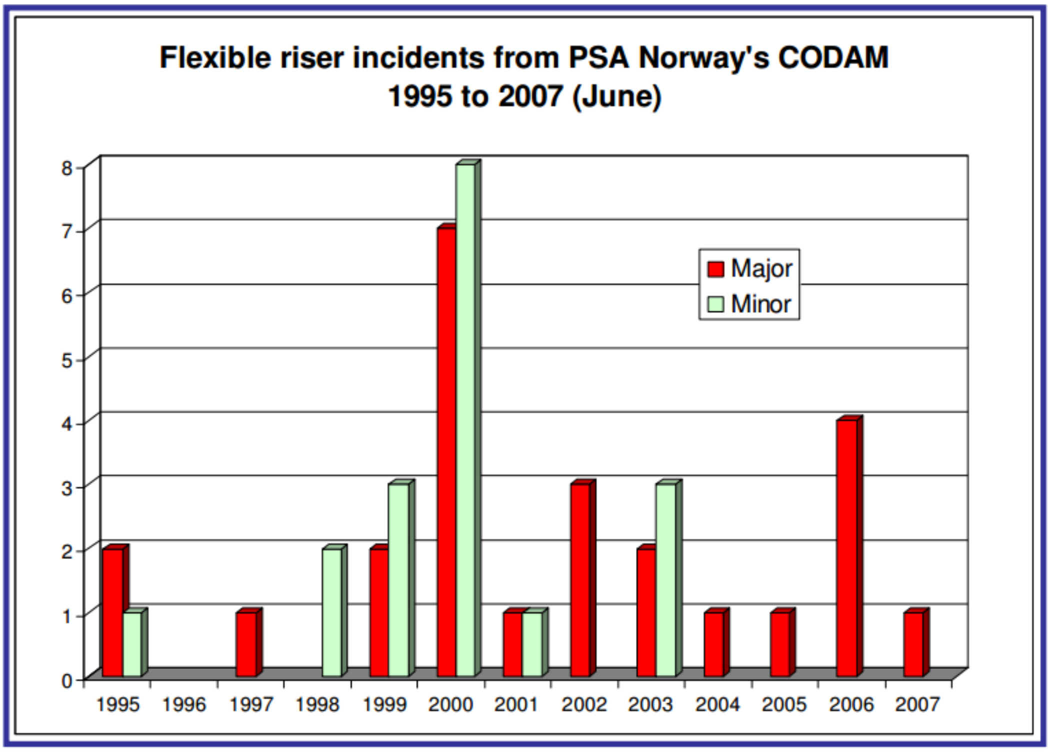


Figure 11-11 Norway Riser failure data Ref. /31/

Some of the problems related to failure have been:

- End Fitting failure
- Carcass collapse
- External Sheet damage
- Flooded annulus
- Wire/armour corrosion and fatigue
- Nylon ageing
- PVDF (Plasticized Poly Vinyl Di Fluoride) Pull out
- Slippage of bend stiffeners
- Marine growth

Some of the major failure modes have been usage outside operation design limits and that some of these failure modes has been resolved by the industry.

Figure 11-12 shows the flexible spools and flow lines used on the Tordis Template. The project was set in production during 1998 -2001. In 2012 Statoil awarded DOFSubsea Norway AS a contract for replacement of the flexible jumper and flow lines.

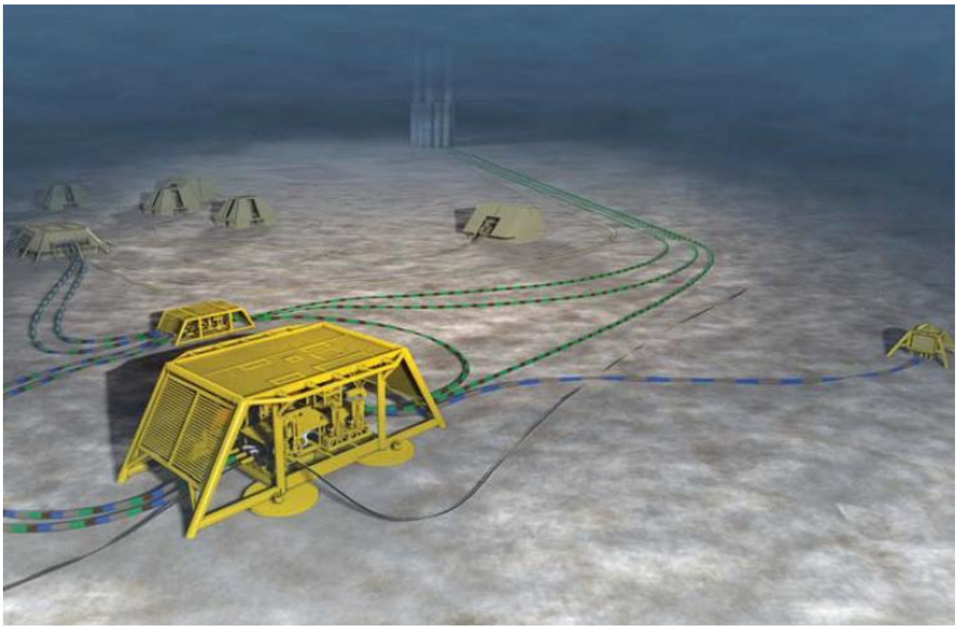


Figure 11-12 Tordis South East Field Flexible flow lines (Statoil)

The advancing technology of today offers more solutions for flexibles. Some of the new advancing products in flexibles are:

- DEEPFLEX Flexible Fibre Reinforced Pipe (FFRP) for deepwater applications Figure 11-13

  Can be installed in water depths > 3000meters. It consists of a high strength to weight ratio composite mixture. The composite structure eliminates the steel corrosion problems and allows for better flow assurance with a smoother bore. The fatigue life is improve

- IPB (Integrated Production Bundle) Technip

  This flexible provides active electrical heating and temperature monitoring. Used for ultra deep water applications. This allows for better control of hydrate and wax formations, developed for risers see Figure 11-14

- Magma m-pipe technology (TFRP) Thermoplastic Reinforced Pipe.

  Is a high performance composite material that can be used for jumper spools and is qualified in accordance with DNV –RP-A203. The product is capable of high strains and provides a high degree of flexibility. The material consists of PEEK material as the core and combination of glass and carbon fibre as the reinforcement. See Figure 11-15 and Figure 11-16.

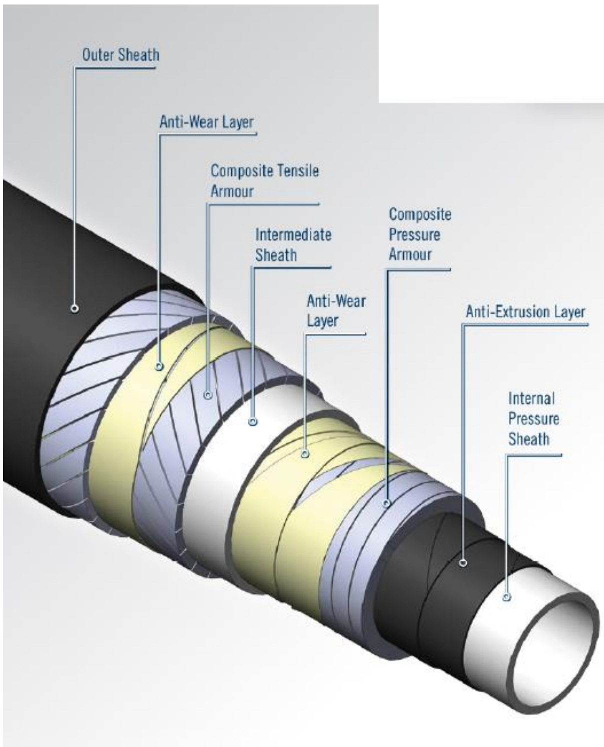


Figure 11-13 Multilayer Composite flexible (DEEPFLEX)

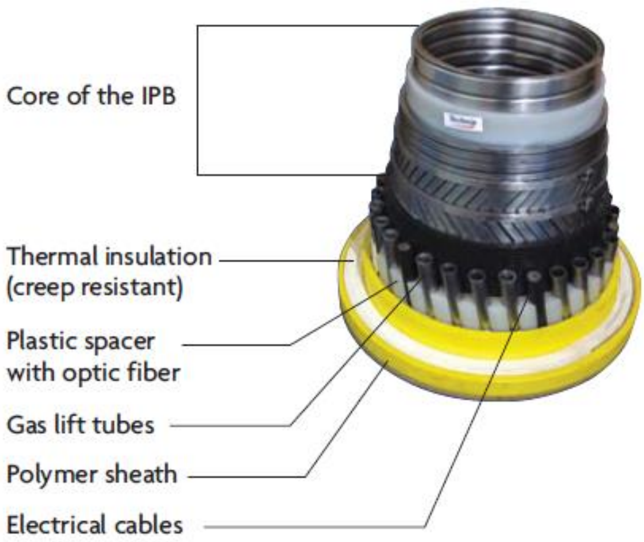


Figure 11-14 IPB Flexible With heat tracing and gas lift (Technip)

Figure 11-15 Jumper Solution m-pipe® (magmaglobal)

Figure 11-16 m-pipe® spool

The author of this thesis sees that for all composite flexibles it can be a challenge if the flow medium contains a high degree of sand content from the wells stream as this could lead to a faster abrasion of the product this should be consolidated with the manufactures of these flexibles

11.3 Design Concept Ideas

During the writing of this thesis some ideas regarding the improvement in the spool design has been proposed. Some of these concepts are:

- Introduction of a flexible joint at each end of a rigid spool

  This could potentially reduce the requirement for high flexibility and number bends in a rigid spool due to less dependent upon metrology and manufacturing tolerances.

- Pre-bending of spools by weld mitigation Ref. chapters 9.3

  This method is simple and reduces reaction forces at each end

- Develop lighter buoyancy material for deep water applications

  The buoyancy material used today is ~ ½ the density and of water and hence this causes the load mitigation aid to growth in diameter size due to increased submerged volume, this can give potential VIV issues for free span. By exploring the collapse limits for these materials or similar materials mixtures in order to optimize the density will benefit the spool design. The cost for buoyancy material may be reduced

- Introduce a spring seabed support

  This method is utilised in topside process piping and reduces stresses bending moments and can have positive effects on VIV sensitivity. Ref. Chapter 9.2.

- Integrate flexibles on reel's as part of subsea structures

  The idea is that an integrated flexible can be a part of the subsea structure. (It is the same principle as the house garden hose). This allows for installation cost savings. The principle is that the ROV only needs to perform a 1st Tie in operation to the other structure as the flexible is already connected and the ROV is basically flying out the other end to the tie-in point

- Integrate spool production facility and testing on board a pipe lay or installation vessels at regions in the world where there is lack of infrastructure for high quality pipe production. This could potentially save installation and transportation cost.

- Use of tension wires anchored to the sea bed can possibly reduce VIV effects for in-line motion

- Development of hybrid solutions between rigid and flexibles

12. SUMMARY, CONCLUSION AND RECOMMENDATIONS

12.1 Summary

In this thesis subsea Tie-in systems and spools used in the oil and gas market is studied. The purpose of this thesis was to explore the standard solutions used today and identify some of the main challenges regarding engineering of subsea Tie-in Spools. The other purpose was to perform strength verification of a spool by use of FEA. The study also explored and compared different computer software used in piping design. Another objective of the thesis is to propose possible new solutions and recommendations for Tie-in spool projects. An executive summary of the results is given here.

Tie-in Spool systems:

The study identified a variety of shapes and types used for spool design. When selecting and designing a Tie in system there are many considerations to be taken. A Pre-Study of the planned project is advisable to commence in order to determine the best spool design. The Pre-study or FEED should focus on the following key topics:

- Subsea routing and location of spool
- Spool type
- Installation time and constraints
- Use conservative data, previous experience is important
- Define optimum spool lengths (minimum and maximum)
- Identify critical values early in the project small changes can give rise to high cost later in project

For the North-Sea region most of the spools used today is of the horizontal type. Vertical spool type is most common for deepwater applications not exposed to risk of snagging or overtrawling. In this thesis industry experience is collected from publications and parent company. The evaluation shows that *vertical spools* are classified as *high risk* towards connector load capacity, increased complexity due to free span, can be sensitive to snagging, possible high risk for seal damage, and it can be difficult to perform pigging operations. For *horizontal spools* the limitation is as for vertical spools for the connector capacity towards bending moments and forces. However the spools are considered to have an overall lower risk compared with vertical spools. It is worth mention here that horizontals spools normally require longer offshore installation time and can give higher project costs.

Probabilistic assessment of fabrication tolerances and metrology used in spool design:

In this thesis the effect of fabrication, metrology tolerances and deformations is studied by use of ANSYS design explorer and the six sigma probabilistic tool. The statistical distribution of these effects has been discussed in chapters 3 and 7. Publications and recommendations on the topic have also been reviewed and discussed. The results in this thesis shows that the probability of reaching high stress levels due to the fabrication and metrology tolerance is very low given a normal statistical distribution for the variables and a uniform distribution for the displacements. It is assumed that there is a 95% probability of hitting the max and minimum angular tolerances with a 100% dimensional control. The analysis shows that the probability of reaching stress levels above the code allowable limits is equal or less to 10^{-4} (0.01%) or 1/10000 events (high sigma level). However the sensitivity analysis shows that the spool response is sensitive to small changes in the tolerance and displacement values and the response is of a non-linear character. Hence this must also be considered when addressing the required safety levels for the spool design.

For the piping analysis it is advisable to incorporate a statistical tool for assessing the fabrication tolerances in order to establish a reasonable safety level in-line with the code requirements. One question is on how the project should handle the standard deviation based upon the critically of the system i.e. should the level be in the range 1.65 - 3 times the standard deviation. How will the probability of exceedance be quantified (5% or 10%)? Most of the analyzed spools today are based upon “worst case” scenario which is a very rare event but should perhaps be analyzed with the most probably expected tolerances and loading. Different techniques for random probabilistic and parameter correlation effects are available and can be included in simulation software (Monte Carlo simulation is one example)

The benefit of using the engineering tool ANSYS Design Explorer is that it is easier for projects to quantify the uncertainty and do an optimisation decision based on a better understanding of the results. Another benefit is that the tool also allows for “what if studies” .The ANSYS Design Explorer tool is user friendly and requires lesser time compared to performing tedious computer programming. This can contribute to a better quality check of the engineering work, since computer scripting is often performed by one engineer and can be a source of human errors.

FEA software comparison:

The traditional way of analysing subsea spools by use of the recently new ANSYS Pipe elements PIPE 288, 289 and ELBOW 290 is shown to be feasible for usage in designing spools, however in some cases it seems that max equivalent stress reported can differ in the location compared to other computer software and solid element models. The reason is not clear it could be that the ANSYS pipe elements reports higher stresses at bend locations due to sharp curvature or radius in pipe bends and that there is less flexibility in ELBOW290. Here it is advisable to study more comparative models in order to conclude. If the location is regarded as critical it is advisable to perform a finite element analysis using solid elements, fabrication tolerances and weld should be included in the analysis. The analysis also shows that use of shell elements in the finite element model of the spool can lead to underestimation of the reported stress levels.
The AutoPIPE software usage is proven to be a good alternative for analysis of vertical spools the results are slightly more conservative and gives higher stress and utilisations compared to the ANSYS beam elements. The benefit of the software is that is very versatile to use for design code checks and it is by far superior to ANSYS when it comes to analysis time and design changes. In addition the user of the software is “forced” to learn the code and pipe fabrication more in depth by using this type of software. The minor side is that it does not seems to be suitable for strain based criteria’s as given in the DNV-OS-F101Pipeline codes and for large non-linear displacements which is often used in horizontal spool design.
For the vertical spools check which was based upon the ASD criteria the utilisation towards the code stress limits has a good match between all computer models except for the ANSYS shell model which seems to underreport the stress levels.

Results from strength verification of vertical Tie-in spool

The vertical spool has been checked for the hydro testing operational and seal replacement load case. The analysis shows that the spool is utilised above the code stress limit *in corroded condition* with a *max utilisation of 109%* using the AutoPIPE piping analysis software and *max utilisation of 107%* using the ANSYS FEA software. The utilisation is largely influenced by the ASME B31.8 longitudinal stress limit criteria. The spool also fails for the minimum fatigue criteria for flexibility stress as given in the ASME B31.8 code. The spool however passes the ASME B31.8 Code check for *the nominal wall thickness* of the pipe, here max utilisation is *94%* for the sustained load case and GR+T(2)+P(2).

For the VIV and fatigue check the spool fails even with a vertical support and the accumulated damage is calculated to *D=19>> greater than fatigue limit of 1/10,* and hence the spool does not have sufficient fatigue capacity for the 25 year lifetime. The vertical support has a positive effect for the cross flow motion on leg 4 of the spool and the vibrations are damped out for higher modes. However since the VIV is dominated by in-line motion, a possible seabed support must also include lateral support in order to reduce this motion. The spool will require mitigation measures such as VIV suppression strakes which must be included on the spool and the effect of this must be checked by new calculations and analysis. The spool is sensitive due to its long free span.

Results from load mitigation study

The result shows that by using a seabed support or a buoyancy element reduces the stresses in the spool by 25%. The largest effect was the reduction of bending moment at the *XT end with 76% reduction.* Hence this gave the best results. The pre-bending of the spool has a positive effect on the bending moments with a decrease of 30% for the bending moment on the XT tree side, but the utilisation for stresses at the manifold side was not largely effected in fact the stresses increased slightly by around~6%. The results are given in Table 9-8.

Future solutions for Subsea Tie-in

The subsea industry and the market have challenges due to a considerable drop in the oil prises. The industry has a large focus on how to reduce installation and development costs for future subsea oil and gas fields. An overview of alternative solutions which has potential to reduce cost is given in chapter 11. Direct Tie-in method for connection pipelines directly to subsea structures without the use of flow line spools is one example. New development of better materials and flexibles spools for smaller pipe sizes is also belived to enter the market in a larger scale. Rigid spools will however be a part of the industry for a long time due to its proven field of record and durability. The new tie-in Systems is less complex than the old ones and this should contribute to reduce costs. In this thesis simple measures which are adopted from the industry is proposed in order to reduce loading on vertical connectors and spool. Some conceptual ideas are presented which is believed to have a possibility of reducing the engineering, fabrication and installation scope such as using flexible joints in spool design, this could reduce the sensitivity for angular tolerance deviations. This would again minimize subsea metrology and survey requirements.

12.2 Conclusion

Subsea spool design projects involve many tasks and considerations. It is important that issues related to the life time cycle of the spools is understood and mapped. Each phase from project concept to end of life time for the spool must be taken into consideration. The work requires in depth knowledge of spool design and fabrication methods from the project members.

In this thesis a vertical spool has been analysed with a statistical and probabilistic approach for the metrology and tolerances, the results shows that it is beneficial to include such method in order to better document the safety level and the conservatism in the spool design. The approach also allows the engineer to make a better decision towards the optimisation process. The ANSYS Design Explorer used in this thesis uses an algorithm for the parametric variations. The software tool generates many possible load combinations and gives output results based upon a statistical distribution of the input values. This results in a faster analysis time. Another positive feature is that it is capable to perform “what if analysis” for many design options and variations.
Other applicable software analysis program is also feasible to use such as the AutoPIPE software from Bentley. The difference compared to ANSYS is small when it comes to piping design using beam element technology and formulas found in a piping code such as the ASME B31.8 Ref. /15/.For a more optimised spool design approach ANSYS solid element modelling is found to give less conservative results. The AutoPIPE does not have the capability for strain based LRFD design according to DNV pipeline code Ref. /7/; hence this also limits the possibility for further optimisation work. Another limitation is the software capability to perform large non-linear displacements which is often the case for horizontal spools which relay on stroking capability, however the software has an advantage when it comes to linking the engineering process and the requirements given in the piping code, the software may be used in an early screening process prior to a more optimised work by use of ANSYS.

Vertical spools have a more challenging design and gives larger engineering and fabrication scope since there are many more considerations to be taken. This gives a higher risk level compared to horizontal spools. The vertical spools have had a traditional advantage that they can be installed faster than horizontal spool and require less deck space which influences the total vessel time used in an offshore campaign. When it comes to installation time subsea this might not be true for the recently developed horizontal spool connectors, as they have become much simpler and would logically give less subsea installation time. Documentation regarding this has not been available and can be difficult to retrieve. An economical and technical analysis of each system prior to a subsea field development is recommended.

The thesis also shows that simple mitigation measures for a vertical spool such as pre-bending and introduction of a seabed support and buoyancy onto the spool has positive effects by reducing the resulting bending moments at connector ends, and can reduce the total stresses in the spool. The result also shows that vertical spool design is very sensitive to VIV, and hence fatigue capacity governs the design.

12.3 Recommendations

For future subsea projects involving spool design the following practice is recommended for the design work and optimisation process.

- Develop design basis
- Use conservative data, previous experience is important
- Identify critical values early in the project small changes can give rise to high cost later in project
- Determine which FEA software is most applicable for usage regarding spool design
- Use pipe beam element technology as far as possible this reduces the amount of analysis time. For a more optimised and requirement for detailed analysis of the spool, design solid element modelling is recommended.
- Implement a statistical and a probabilistic tool (such as ANSYS design Explorer) for assessing the distribution of tolerances, deflections and metrology, and load combinations.
- Perform parameter study and sensitivity study to changes.
- The spool should be analysed for the whole life cycle.
- Define optimum spool lengths (minimum and maximum)
- Perform manual calculations as a benchmark and for checking results

<u>Proposal for future studies:</u>

- Development of hybrid solutions between rigid and flexibles
- Integrate spool production facility and testing on board a pipe lay or installation vessels at regions in the world where there is lack of infrastructure for high quality pipe production. This could potentially save installation and transportation cost.
- Development of lighter or alternative and cheaper buoyancy material for deep water applications
- Develop a software tool that integrates DNV OS-F101 LRFD design into an typical piping software such as the AutoPIPE or similar.
- Perform further studies for the Direct Tie-in method for pipelines.

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Appendix 1 Pre-Study Master Thesis

University of Stavanger

Faculty of Science and Technology

Pre-study to Master Thesis

Study program/ Specialization: Master in Mechanical Engineering	Spring semester, 2015 Open
Writer: Loyd Kjetil Helland Andersen	Loyd Kjetil H. Andersen (Writer's signature)
Faculty supervisor: Hirpa G. Lemu, UIS External supervisor(s): Per Nystrøm, IKM Ocean Design AS	
Thesis title: Subsea Tie-in, Design Solutions and Optimisation Methods	
Credits (ECTS): 30	
Key words: -Tie-in System -Subsea installation -FE analysis -Design methods -Pipe Spool	Pages: 23 + enclosure: 0 Stavanger, 14.02.2015

Preface

This is a pre-study to the Master's Thesis to be conducted at UIS, department of Mechanical and Structural Engineering and Material Science during spring 2015. The Pre-study report is made with the intention to define the scope of work for this thesis. Key words are Subsea Tie-in spools and design methods used in the gas and oil industry. The design route for Tie-in spool used today is either the DNV-OS-F101 Submarine Pipeline Systems or the ASME Code for Pressure Piping, B31.8, B31.4 Pipeline Transportation. A recommended practice for the design of rigid spools is not yet set forth. According to the DNV list of Joint industry project (JIPS) a project for these types of spools was completed in 2013. The result of this project is not known to author at present day. This Thesis main objective is to investigate standard solutions, identify main challenges for Tie-in Spools, perform a spool analysis, and propose possible new solutions and recommendations for the commencing of such projects.

The Thesis is written in co-operation with IKM Ocean Design A/S, which has been my employer for the last 7 years where I have been working as a Structural and Mechanical Engineer. IKM Ocean Design Specializes in design and engineering of subsea pipelines, subsea structures and Tie-in solutions. The company is a sub company of the IKM Group in Norway which is a major sub supplier to the oil and gas industry.
During the years working for IKM two vertical Tie-in Spool systems for deepwater applications projects has been proven to be of great challenge when it comes to design optimization, analysis techniques and strength verification. Hence a requirement for a more standardized route and methods for these types of spool would indeed benefit future projects.

Table of contents

1. Introduction

A pipeline connection is normally used as a link between a pipeline, manifold, oil-well, storage tank, processing facility or other mechanical equipment used for the transportation of a fluid, gas, sand or a combination of all from one location to another. The pipeline link connection is called a spool which is an English terminology (in Norwegian it translates to “snelle” , which is a device for reeling something on like a fishing reel). When we use the word spool in piping terminology it is understood as piece of pipe with necessary bends tees and flanges for connection to another system. In simple terms it is the pipe from flange to flange. The concept is relatively simple. As the pipes are heated and pressurized they expand and since the piping is restrained in some way in a piping system stresses are developed. For subsea pipelines the spools is usually an infield pipeline connection to a trunk exporting pipeline, manifold, oil wells or other subsea facility. The transport medium is:

- Produced oil
- Gas injection
- Water injection
- Multiphase flow (oil, gas and water)

Spools must have enough flexibility to withstand the expansion deflection from facilities such as:

- Pipeline and Risers connected to subsea structures or other processing unit.
- Oil-wells and manifolds
- Environmental forces

1.1 Historical

Since the 1980’s, when the subsea industry started moving into water depths where divers could not be used, the industry has been challenged to provide a simple cost effective method of connecting two lines without divers.

The industry has responded to this challenge providing innovative methods of doing first end and second end tie-in methods including:

- stab & hinge-over
- Rigid jumpers/spools
- Flexible
- Deflect and connect

A multitude of vertical and horizontal connectors & tools:

> However, the use of rigid jumpers still remains the universal method of performing deepwater pipeline connections, possibly due its extensive proven track record, its cost effectiveness and high reliability. However, this system still has significant drawbacks which include the requirement for metrology, topsides fabrication (which may or may not be on the critical path), installation with a multipoint lift and its limited capability to accommodate pipeline expansion and two tie-in operations. Ref. /1/

Some of the early projects during the 1980,s utilizing the deflect to connect approach was

- East Frigg Project. June 1988. Connection of 2 production manifolds to a central manifold by 2 bundles in 24” carrier pipes to provide buoyancy.

Bundles connected by a first time diver less deflect to connect method.

- Troll Olje Project. August 1995. Connection of 16” oil and gas export pipelines. First time diver less "deflect to connect" directly on pipelines by attaching weight and buoyancy.

1.2 **Background**

Subsea Tie-in solutions provided by most of the major actors in the subsea market provides various systems for connecting pipelines to manifolds, wells and Trunk pipe lines. These pipelines are usually called “spools” or tie in spool. This is usually a steel pipe oriented either vertically or horizontally with a connector system in each end, other types used is of a flexible types similar to what is used in risers. These spools are often designed to withstand large forces and displacements due to pressure and temperature in the pipeline during installation and operation; hence the requirement for flexibility and strength is one of the key design features. Various computer optimization techniques such as the use of FEA and CFD are utilized in order to analyze and verify strength of these spools towards numerous load combinations in order to document required design life and governing codes. Experience has shown that some of these solutions are sensitive to parameter changes such as:

- Flow and process data
- Material choice
- Metrology and fabrication tolerances
- Environmental factors.
- Size and shape
- Connector solutions

Typically main issues related to design of rigid spools can be listed as follows:

- Size
- Stresses
- Conflict between company standard and code requirements
- Lack of recommended practice
- Corrosion and (HISC) problems
- Insulation
- VIV
- Weight
- Fatigue
- Erosion
- Slugging
- Pressure loss sour service
- Requirement for MEG inhibitors
- Sour service
- Seabed
- Size and limitation of connector systems
- Requirement for structural support equipment

In order to reduce project cost, time and complexity for subsea Tie-in projects the following areas are considered to be of interest to study in a FEED (Front End Engineering and Design) phase:

1. Computer analysis techniques and design methods
2. An early identification and mapping of critical design values and limitations
3. Other relevant Tie-in solutions
4. Mapping of complexity in the project
5. Vessel installation time.
6. Use of standard solutions and previous project experiences.
7. Usage of design standards, company practices and codes.

In order to limit the work in this Thesis, point 1, 3 and 7 is chosen as main areas to study.

1.3 Scope of work

This Thesis major purpose is to investigate some of the standard solutions of the tie-in system as used by the major actors in the oil and gas industry.
The Thesis will utilize other studies, company experience, papers and Master Thesis on this topic.
Main objective is to analyze a vertical jumper spool by use of a commercial Finite Element Analysis Software, and to study spool design such as:

- Investigate the effect of a flexible joint or seabed support in order to reduce moment and forces in a rigid spool.
- Optimize the computer analysis by parametric variation

The study will include

- Development of design basis for analysis
- Theory
- Use of applicable standards

Other topics such as

- Conceptual ideas
- Further studies and development for Tie-in
- Limitations

Propose recommendations for commencing of such projects and the result of the case study

1.4 Flexibility in piping design

Flexibility design in piping systems is of great importance to study as temperature, pressure and gravity forces are inducing stresses, strains and deformations in the pipe system when it is restrained. Furthermore as the piping system heats up and shuts down the piping system is exposed to changes in stresses, this causes a fatigue situation. For a piping system exposed to environmental forces such as current and waves typically for subsea pipes, VIV (Vortex Induced Vibration) can cause the pipeline to be excited into harmonic low frequency vibration. This can result in fatigue failure or unintentional high displacement ranges. The designer must calculate the stresses allowed by a particular code. One of the significant differences between flexibility analysis and pressure design is that flexibility is related to stress range rather than a specific stress. For subsea piping the requirement for flexibility is how to make a system flexible enough in order to be able to handle the deformations from pipeline expansion, Tie-in forces, metrology and fabrication tolerances environmental forces etc. The acceptance criteria are usually a strain and stress based criteria set forth by a piping code.

The principle for flexibility design is shown by a simplified calculation known as the guided cantilever approximation /2/. A pipe is restrained against axial movements causing a deformation shape as shown by the dotted line. The deflection is assumed to occur in a single plane system under the guided cantilever approximation.

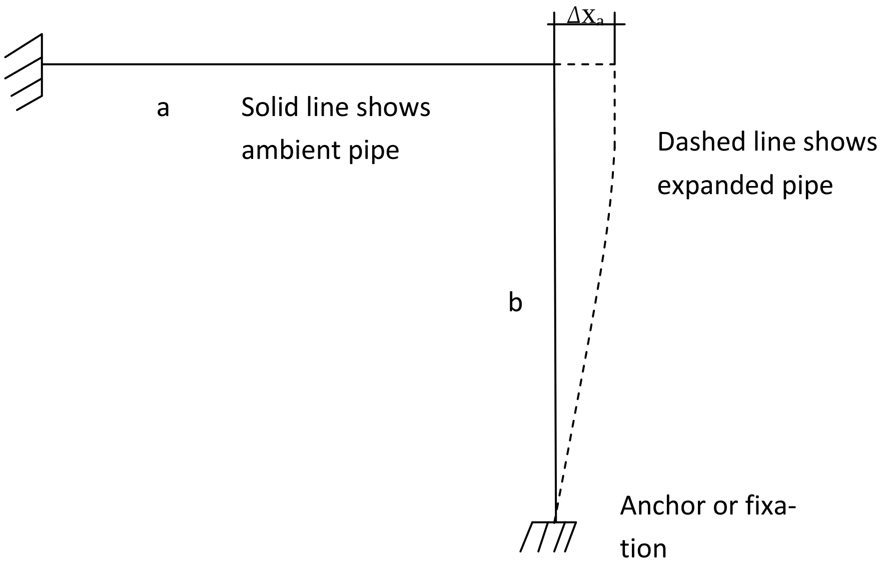

Figure 1 simple Restrained pipe flexibility design

The deflection capacity of a cantilever under this assumption can be given by the following equation in (US customary units)

$$\Delta X_a = \frac{144L^2 S_A}{3ED_o} \quad (1)$$

Where

Δ= permissible deflection

L= Length of pipe element to absorb the expansion

E=Youngs modulus at cold temperature

D_o=Outside diameter of pipe

S_A=Allowable stress range

The limitation of the method is:

- The system has only two terminal points and is composed of straight legs of pipe with uniform size and thickness
- All legs are parallel to the coordinate axes
- Thermal expansion is absorbed only by legs in a perpendicular direction
- The amount of thermal expansion that a given leg can absorb is inversely proportional to its stiffness. Because the legs are of inverse value of the cube of their lengths.
- In accommodating thermal expansion the legs act as a guided cantilever, that is they are subjected to bending under end displacement no end rotation is permitted

Consider the following example as to calculate the required length of leg (b) if the system is exposed to a displacement in x direction of 2.3in (58mm) due to thermal expansion of leg (a).

Pipe data:
D_o=4.5 in
Δ_{xa}=2.3 in
S_A=15000 psi
E=29.7x10^6psi
La=20ft

By rearranging Eq. 1.1 the required length for Leg (b) becomes in:

$$L_b = \sqrt{\frac{3ED_o \cdot \Delta xa}{144S_A}} = \sqrt{\frac{3 \cdot 29.7 \cdot 10^6 \cdot 4.5 \cdot 2.3}{144 \cdot 15000}} = 20.7ft(6298mm) \tag{2}$$

This simple calculation shows that Leg (b) has to be longer than that of leg (a) in order to be within a safe stress limit for the pipe. By observing this formula one sees that the allowable deflection is depended up-on the pipe length leg squared hence doubling the expansion Δ of pipe length L_a requires the other length L_b of the pipe to increase by a factor of: $\sqrt{2} = 1.41$, in order to maintain stresses within the limit.

Other flexibility criteria as given in ASME B31.4 Ref. /3/section 403.9 can also be used as a simplification. This formula is an empirical criteria for checking the expansion of unrestrained pipelines.

$$\frac{Dy}{(L-U)^2} \leq K \tag{3}$$

Where
D= Outside Diameter of the pipe
L= Developed Length of the pipe between anchors
U=Straight line distance between anchors
K=numerical factor =208 (SI Units) and 0.03 for US customary units
y=Resultant of total displacement strains to be absorbed by the pipe ($U\alpha\Delta T$)

The code specifies the following criteria to be met
"Pipelines shall be designed to have sufficient flexibility to prevent expansion or contraction from causing stresses in the pipe material or pipeline components that exceed the allowable specified herein, including joints, connections, anchor points, or guide points. Note that allowable forces and moments on equipment may be less than for the connected pipe. Analysis of adequate flexibility of unrestrained pipe is not required for a pipeline that
(a) Has been previously analyzed
(b) Is of uniform size, has no more than two anchor points, no intermediate restraints, and falls
Within the limitations of the following empirical formula given by (3):
Any pipeline not meeting the requirements given above shall be analyzed by a simplified, approximate, or comprehensive method as appropriate. The effects of all parts of the pipeline and components and of all restraints, including friction, shall be accounted for."
By comparison to the example given by the cantilever method the calculation using formula (2) becomes:

Table 1-1 ASME B31.4 Flexibility Criteria

ASME B31.4 Section 403.9 Check			
Outside diameter of pipe	D_{od}=	4,50	in
Length a	L_a=	30	ft
Length b	L_b=	34	ft
Length of spool	L_{spool}=	64	ft
Straight line distance between anchors	$U=(L_a^2+L_b^2)^{0.5}$	45,3	ft
Total imposed displacement	y=	2,30	in
Numerical factor ASME B31.4	K=	0,03	
Criteria	$a=(D_{od}\cdot y)/(L_{spool}-U)^2$	0,030	
	Flexibillity Check a≤K	OK	

According to this empirical formula given by (3) the spool has to increase leg (a) to 30ft and leg (b) to 34ft in order to pass the criteria. There is no general proof of this formula as to its accuracy or conservatism. The formula might only be valid for temperature rise change and not imposed external displacements. As shown the simplified calculation has limitations and the designer must decide whether or not it is required to proceed to a more accurate analysis.

1.5 Analysis Software

Various methods exits such as simplified analysis, charts , graphical analysis, computer analysis tool such as Autopipe by Bentley Ref. Figure 2, CAESAR II from Intergraph and Triflex from piping solutions.com. The Software checks that stresses are within the code limit. The software usually has a limitation of number of load combinations and large non-linear displacement functions. The software is mostly utilized in topside /onshore process piping systems but is also feasible for process system used subsea

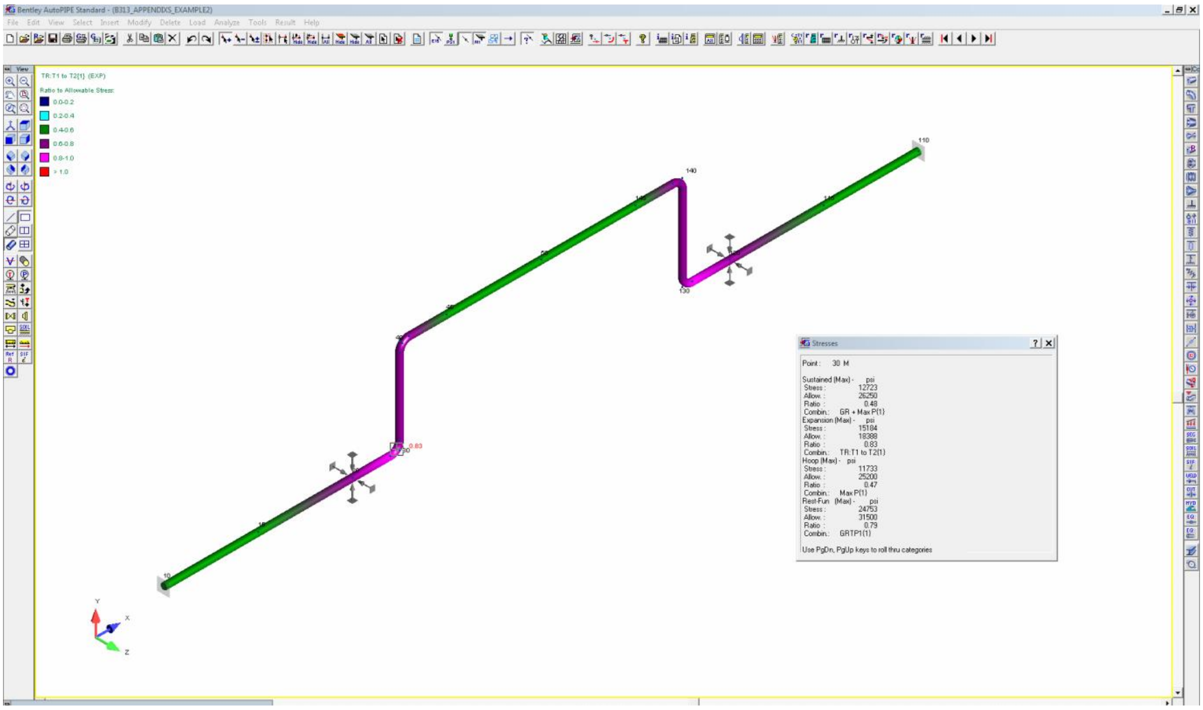

Figure 2 Autopipe typical Stress Analysis Software by Bentley

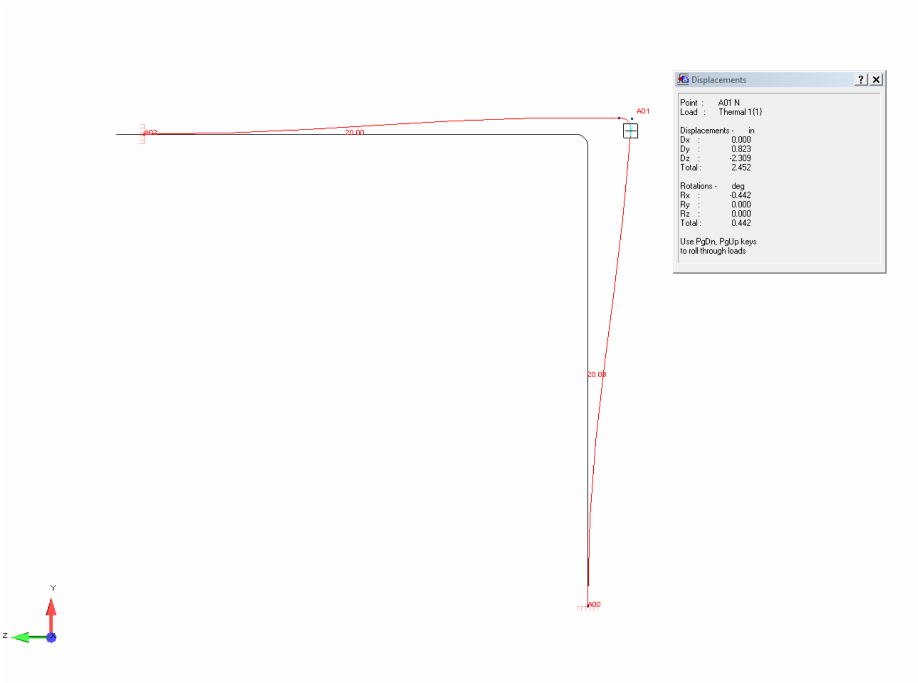

Figure 3 Autopipe Stress Analysis of 4”pipe spool-max deflection

Figure 3 and Figure 4 shows the result of the stress analysis for the guided cantilever method. For a max displacement of 2.3in (imposed +thermal expansion) and leg (b) equal to 20.03ft the software reports back a max utilization ratio of **UF= 1.07** towards allowable stress ratio which is almost identical to the analytical result (7% difference)**.** This indicates that the guided cantilever method is a good approximation for simple calculation and estimation but lacks code approval given in the ASME codes.

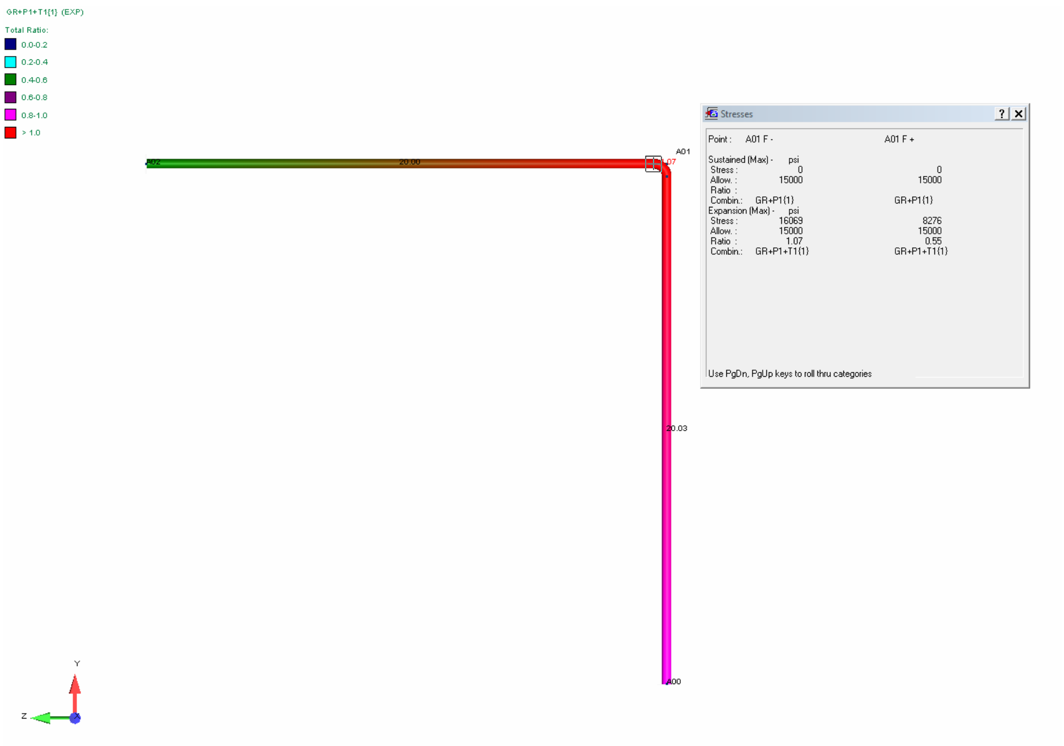

Figure 4 Autopipe Stress Analysis of 4”pipe spool-max stress ratio

For a more advanced study or special case such as subsea flow lines and tie-in spools the most utilized FEA software used today is the ANSYS software package. Here the pipe model can be set up by use of script commands in the APDL (ANSYS Parametric Design Language) and the whole sequence of installing and operation the pipe within many hundreds or thousands of load combinations can be run in a loop. The software also has the capability to model non-linear contact problems such as seabed friction. Figure 5 and Figure 6 shows a typical classic FEA model of a spool. Pipe elements with physical properties are meshed along a line element. The challenge is to vary the combinations of loads and to check that the configuration of the spool is within a safe stress and deflection limit. Examples of design data for a rigid spool is shown in Table 1-2to Table 1-4.The spool is normally fabricated based on the field metrology report after installation of the Manifold and PLET (Pipeline End Termination). The installation tolerances however are covered for by changing the jumper geometry in the analyses to find the most unfavorable configuration, governing for the spool design. However, the settlements and the fabrication and metrology tolerances will be unknown Project experience has shown that changing these values can have a major impact on the design hence giving rise to more uncertainties in combinations with all the other design parameters.

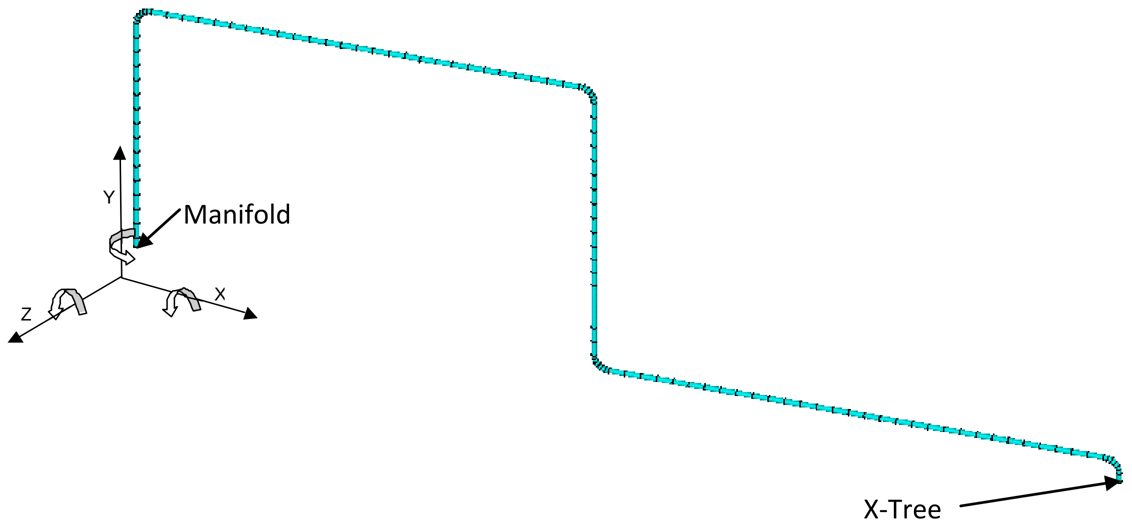

Figure 5 ANSYS Pipe element model vertical spool

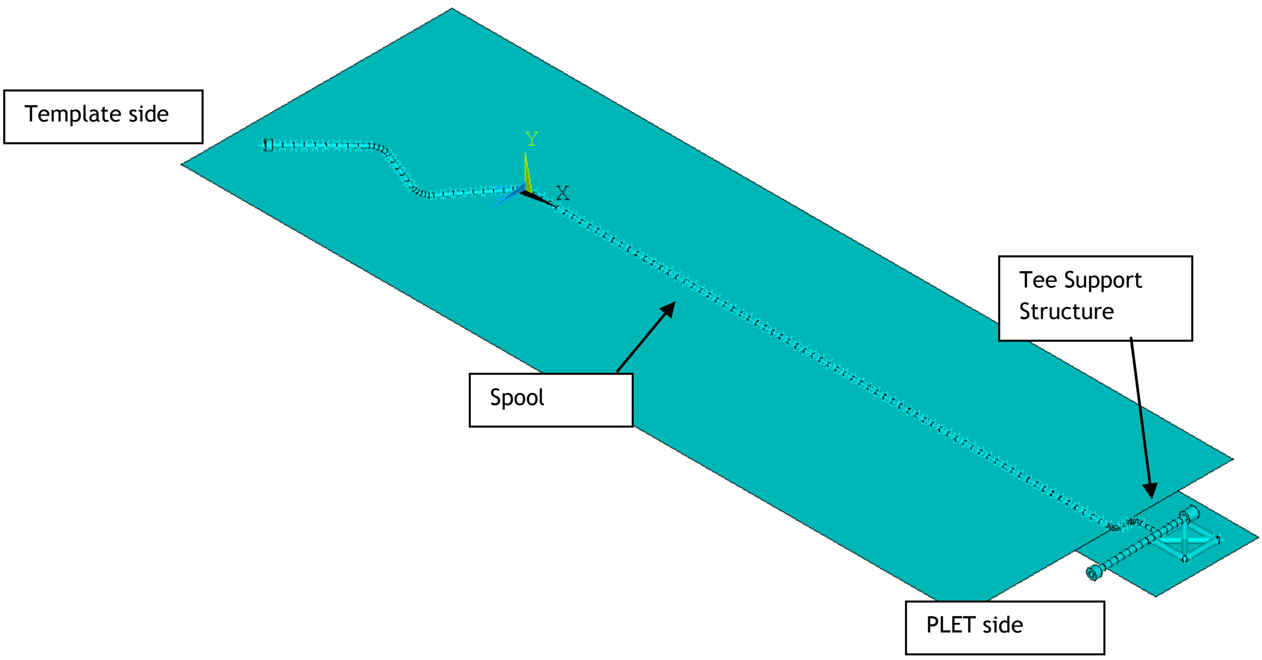

Figure 6 ANSYS Pipe element model horizontal spool with seabed

Table 1-2 Design data for a typical deep water project

Design parameter	Value
Subsea ambient temperature	4.3°C
Density of fluid	747 kg/m^3
Seawater density	1026 kg/m^3
Current speed (Operational)	0.7 m/s
Max. design water depth collapse	914 m
Min. design water depth to obtain design gauge pressure	700 m
Max design pressure	5000 psi (345 bar)
Max temperature range	-29°C to **+**121°C
Slugging force from well stream flow (multiphase flow)	v_s=10 m/s density slug 900 kg/m^3
Pipe size	6 "ND Sch.120
Pipe material	25% Cr Super Duplex

Table 1-3 Typical Installation, Tolerances and Settlements for a subsea project

Parameter	PLET	Manifold	X-tree	Spool/pipe
Position	± 3.0 m	2.0 m Radius		
Heading	± 5 °			
Vertical angle	± 5 °	± 2.5 °		
Long term settlement	- 200 mm	-130mm		
Short term settlement	- 20 mm			
Vertical position		+0.3m		
Horizontal deflections	Pipeline expansion		216 mm	
Vertical deflections	200 mm		25 mm	
Fabrication tolerance angular				± 0.5 ° / ± 1 °
Fabrication tolerance linear				5 to 10mm

Table 1-4 Typical Clamp Connector/Hub Capacities

Parameters	Data
Max axial force Tension	100 kN
Max bending Moment at max pressure	250 kNm
Max bending Moment clamp closing	260 kNm
Max allowable bending moment	200 kNm
Torsional capacity	109 kNm

2. Work Break down Structure (WBS)

Table 2-1 shows the WBS for this Thesis. The study is divided into chapters where each WBS reflects a topic. The objective and scope of each topic is described in a standard IKM CTR format (Cost, Time and, Resource).The cost estimate for this thesis is not necessary. The CTR is linked to the schedule given in chapters 3. The thesis is mainly one resource dedicated to the task. Some assistance of IKM engineers will be required during the period as to give input to software usage, technical advices checking of results and literature on the topic.

Table 2-1 Thesis work break down structure

CTR No.	Description	Reference
E.1.0	Pre-study and Introduction to Master Thesis	2.1
E.1.1	Introduction	2.2
E.2.0	Tie-in Spool Systems	2.3
E.3.0	Connector Systems	2.4
E.4.0	Spool optimization FEA	2.5
E.5.0	Future solutions for Tie-in systems	2.6
E.6.0	Engineering Route Subsea Spools	2.7

COST, TIME, RESOURCE			IKM Ocean Design AS	
Project	**Master thesis Subsea Tie-In** **Design Solutions and Optimisation Methods**			
CTR No. **E.1.0**	**Title :**	2.1 **Pre-study and Introduction to Master Thesis**		
Objective	Identify main challenges and technical issues related to subsea tie-in			
Scope of work:	• An extensive literature review • Pros and cons of current subsea solutions • Identify the industry technical challenges			
Duration	: Ref. Schedule			
Estimate	:NA		**Prepared by**	: LKHA
			Date	: 08.02.2015

COST, TIME, RESOURCE			IKM Ocean Design AS
Project	**Master thesis Subsea Tie-In** **Design Solutions and Optimisation Methods**		
CTR No. E.1.1	**Title :**	2.2 **Introduction**	
Objective	Describe main purpose with subsea Tie-in spools		
Scope :	• An extensive literature Review • Background • Historical development • Main functionality and purpose of Tie-in systems		
Duration	**:** Ref. Schedule		
Estimate	:NA	**Prepared by**	LKHA
		Date	08.02.2015

COST, TIME, RESOURCE				IKM Ocean Design AS
Project	Master thesis Subsea Tie-In Design Solutions and Optimisation Methods			
CTR No. E.2.0	**Title :**	2.3 **Tie-in Spool Systems**		
Objective	Identify typical Tie-in System used in today's industry			
Scope :	• Classification of systems • Main purpose • Configurations and geometrical layouts • Limitations			
Duration	**:** Ref. Schedule			
Estimate	NA	**Prepared by**	LKHA	
		Date	08.02.2015	

<table>
<tr><td colspan="2">COST, TIME, RESOURCE</td><td colspan="3">IKM Ocean Design AS</td></tr>
<tr><td colspan="5"></td></tr>
<tr><td>Project</td><td colspan="4">Master thesis Subsea Tie-In
Design Solutions and Optimisation Methods</td></tr>
<tr><td>CTR No. E.3.0</td><td>Title :</td><td colspan="3">2.4 Connector Systems</td></tr>
<tr><td>Objective</td><td colspan="4">Identify typical Mechanical Connector Systems used for Subsea Tie-Back</td></tr>
<tr><td>Scope :</td><td colspan="4">• Major Suppliers
• Differences
• Functionality and operability
• Limitations</td></tr>
<tr><td>Duration</td><td colspan="4">Ref. Schedule</td></tr>
<tr><td>Estimate</td><td colspan="2"></td><td>Prepared by</td><td>LKHA</td></tr>
<tr><td colspan="3"></td><td>Date</td><td>08.02.2015</td></tr>
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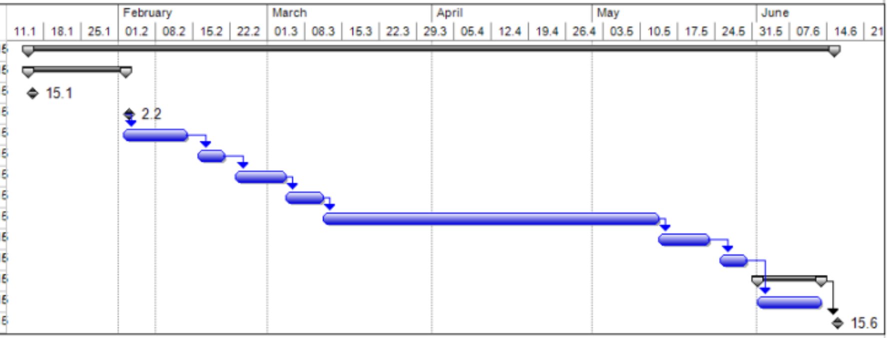
<tr><td colspan="5"></td></tr>
<tr><td>Project</td><td colspan="4">Master thesis Subsea Tie-In
Design Solutions and Optimisation Methods</td></tr>
<tr><td>CTR No. E.4.0</td><td>Title :</td><td colspan="3">2.5 Spool optimization FEA</td></tr>
<tr><td>Objective</td><td colspan="4">Verify and optimize the structural strength of a typical spool by use of FEA</td></tr>
<tr><td>Scope :</td><td colspan="4">Design basis
Identification of load cases
Flexibility design
Wall thickness design
Perform computer analysis of spools
Compare different solutions
Establish criteria based upon industry standard such as ASME ,DNV</td></tr>
<tr><td>Duration</td><td colspan="4">Ref Schedule</td></tr>
<tr><td>Estimate</td><td colspan="2">NA</td><td>Prepared by</td><td>LKHA</td></tr>
<tr><td colspan="3"></td><td>Date</td><td>08.02.2015</td></tr>
</table>

COST, TIME, RESOURCE				IKM Ocean Design AS
Project	Master thesis Subsea Tie-In Design Solutions and Optimisation Methods			
CTR No. E.5.0	**Title :**	2.6 **Future solutions for Tie-in systems**		
Objective	Propose other possible Tie-in Solutions			
Scope :	• Describe other methods not utilised today such as Direct Tie-in Deflect to connect etc. • Flexible spools • Other conceptual Ideas			
Duration	Ref. Schedule			
Estimate		**Prepared by**	LKHA	
		Date	08.02.2015	

COST, TIME, RESOURCE			IKM Ocean Design AS	
Project	Master thesis Subsea Tie-In Design Solutions and Optimisation Methods			
CTR No. E.6.0	**Title :**	2.7 **Engineering Route Subsea Spools**		
Objective	Establish a best practice route for Spools project			
Scope :	• Describe experience from projects • Establish a flow chart for choice of best suited spool type and configuration			
Duration	Ref. Schedule			
Estimate			**Prepared by**	LKHA
			Date	08.02.2015

3. Schedule

		Name	Duration	Start	Finish
1		**Master Thesis Subsea Tie-In Design Solutions and Optimisation Metho**	**107 days**	**15.1.15**	**15.6.15**
2		**Application delivered at UIS**	**12 days**	**15.1.15**	**2.2.15**
3	✓	Application Approved	0 days	15.1.15	15.1.15
4		Official Start of thesis	0 days	2.2.15	2.2.15
5		**CTR E1.0 Prestudy**	10 days	2.2.15	13.2.15
6		**CTR E1.1 Introduction**	1 wk	16.2.15	20.2.15
7		**CTR E2.0 Tie-In Spool systems**	1,5 wks	23.2.15	4.3.15
8		**CTR E3.0 Connector systems**	1 wk	4.3.15	11.3.15
9		**CTR E4.0 Case Study Spool Optimisation FEA**	9 wks	11.3.15	13.5.15
10		**CTR E5.0 Future Solutions Tie-in**	**1,5 wks**	**13.5.15**	**22.5.15**
11		**CTR E6.0 Engineering Route for Subsea Spools**	**1 wk**	**25.5.15**	**29.5.15**
12		**Draft Issue**	**10 days**	**1.6.15**	**12.6.15**
13		Include comments and finnish report and copies	**2 wks**	**1.6.15**	**12.6.15**
14		**Delivery of the master thesis**	0 days	15.6.15	15.6.15



Project: Schedule
Date: 8.2.15

Task | Progress | Summary | External Tasks | Deadline
Split | Milestone | Project Summary | External Milestone

Page 1

4. References

/1/	Per Richard Nystrøm and Gawain Langford, “Jumper Arch System”, OPT-2002-03, ABB Offshore systems presented at the Offshore Pipeline Technology Conference, 2002
/2/	Sam kannappan John Wiley & sons Inc. “Introduction to pipe stress analysis” Tennessee Valley Authority, Knoxville Tennessee 1986
/3/	*Pipeline Transportation systems for liquids and Slurries* ,ASME B31.4, 2012

Appendix 2 ANSYS Design Explorer Spool Deflection Combinations

Table A2-1 Spool Imposed Displacement and Rotation Combination's

Design Point	Dy1 (mm)	Rx1 (degree)	Rz1 (degree)	Dx2 (mm)	Dy2 (mm)	Rx2 (degree)	Rz2 (degree)
1	-130,00	0,00	0,00	0,00	-31,00	0,00	0,00
2	-160,94	0,00	0,00	0,00	-31,00	0,00	0,00
3	-99,06	0,00	0,00	0,00	-31,00	0,00	0,00
4	-130,00	-0,93	0,00	0,00	-31,00	0,00	0,00
5	-130,00	0,93	0,00	0,00	-31,00	0,00	0,00
6	-130,00	0,00	-0,93	0,00	-31,00	0,00	0,00
7	-130,00	0,00	0,93	0,00	-31,00	0,00	0,00
8	-130,00	0,00	0,00	-246,51	-31,00	0,00	0,00
9	-130,00	0,00	0,00	246,51	-31,00	0,00	0,00
10	-130,00	0,00	0,00	0,00	-55,95	0,00	0,00
11	-130,00	0,00	0,00	0,00	-6,05	0,00	0,00
12	-130,00	0,00	0,00	0,00	-31,00	-0,93	0,00
13	-130,00	0,00	0,00	0,00	-31,00	0,93	0,00
14	-130,00	0,00	0,00	0,00	-31,00	0,00	-0,93
15	-130,00	0,00	0,00	0,00	-31,00	0,00	0,93
16	-146,37	-0,49	-0,49	-130,42	-44,20	-0,49	0,49
17	-113,63	-0,49	-0,49	-130,42	-44,20	-0,49	-0,49
18	-146,37	0,49	-0,49	-130,42	-44,20	-0,49	-0,49
19	-113,63	0,49	-0,49	-130,42	-44,20	-0,49	0,49
20	-146,37	-0,49	0,49	-130,42	-44,20	-0,49	-0,49
21	-113,63	-0,49	0,49	-130,42	-44,20	-0,49	0,49
22	-146,37	0,49	0,49	-130,42	-44,20	-0,49	0,49
23	-113,63	0,49	0,49	-130,42	-44,20	-0,49	-0,49
24	-146,37	-0,49	-0,49	130,42	-44,20	-0,49	-0,49
25	-113,63	-0,49	-0,49	130,42	-44,20	-0,49	0,49
26	-146,37	0,49	-0,49	130,42	-44,20	-0,49	0,49
27	-113,63	0,49	-0,49	130,42	-44,20	-0,49	-0,49
28	-146,37	-0,49	0,49	130,42	-44,20	-0,49	0,49
29	-113,63	-0,49	0,49	130,42	-44,20	-0,49	-0,49
30	-146,37	0,49	0,49	130,42	-44,20	-0,49	-0,49
31	-113,63	0,49	0,49	130,42	-44,20	-0,49	0,49
32	-146,37	-0,49	-0,49	-130,42	-17,80	-0,49	-0,49
33	-113,63	-0,49	-0,49	-130,42	-17,80	-0,49	0,49
34	-146,37	0,49	-0,49	-130,42	-17,80	-0,49	0,49
35	-113,63	0,49	-0,49	-130,42	-17,80	-0,49	-0,49
36	-146,37	-0,49	0,49	-130,42	-17,80	-0,49	0,49
37	-113,63	-0,49	0,49	-130,42	-17,80	-0,49	-0,49
38	-146,37	0,49	0,49	-130,42	-17,80	-0,49	-0,49
39	-113,63	0,49	0,49	-130,42	-17,80	-0,49	0,49
40	-146,37	-0,49	-0,49	130,42	-17,80	-0,49	0,49
41	-113,63	-0,49	-0,49	130,42	-17,80	-0,49	-0,49
42	-146,37	0,49	-0,49	130,42	-17,80	-0,49	-0,49

Design Point	Dy1 (mm)	Rx1 (degree)	Rz1 (degree)	Dx2 (mm)	Dy2 (mm)	Rx2 (degree)	Rz2 (degree)
43	-113,63	0,49	-0,49	130,42	-17,80	-0,49	0,49
44	-146,37	-0,49	0,49	130,42	-17,80	-0,49	-0,49
45	-113,63	-0,49	0,49	130,42	-17,80	-0,49	0,49
46	-146,37	0,49	0,49	130,42	-17,80	-0,49	0,49
47	-113,63	0,49	0,49	130,42	-17,80	-0,49	-0,49
48	-146,37	-0,49	-0,49	-130,42	-44,20	0,49	-0,49
49	-113,63	-0,49	-0,49	-130,42	-44,20	0,49	0,49
50	-146,37	0,49	-0,49	-130,42	-44,20	0,49	0,49
51	-113,63	0,49	-0,49	-130,42	-44,20	0,49	-0,49
52	-146,37	-0,49	0,49	-130,42	-44,20	0,49	0,49
53	-113,63	-0,49	0,49	-130,42	-44,20	0,49	-0,49
54	-146,37	0,49	0,49	-130,42	-44,20	0,49	-0,49
55	-113,63	0,49	0,49	-130,42	-44,20	0,49	0,49
56	-146,37	-0,49	-0,49	130,42	-44,20	0,49	0,49
57	-113,63	-0,49	-0,49	130,42	-44,20	0,49	-0,49
58	-146,37	0,49	-0,49	130,42	-44,20	0,49	-0,49
59	-113,63	0,49	-0,49	130,42	-44,20	0,49	0,49
60	-146,37	-0,49	0,49	130,42	-44,20	0,49	-0,49
61	-113,63	-0,49	0,49	130,42	-44,20	0,49	0,49
62	-146,37	0,49	0,49	130,42	-44,20	0,49	0,49
63	-113,63	0,49	0,49	130,42	-44,20	0,49	-0,49
64	-146,37	-0,49	-0,49	-130,42	-17,80	0,49	0,49
65	-113,63	-0,49	-0,49	-130,42	-17,80	0,49	-0,49
66	-146,37	0,49	-0,49	-130,42	-17,80	0,49	-0,49
67	-113,63	0,49	-0,49	-130,42	-17,80	0,49	0,49
68	-146,37	-0,49	0,49	-130,42	-17,80	0,49	-0,49
69	-113,63	-0,49	0,49	-130,42	-17,80	0,49	0,49
70	-146,37	0,49	0,49	-130,42	-17,80	0,49	0,49
71	-113,63	0,49	0,49	-130,42	-17,80	0,49	-0,49
72	-146,37	-0,49	-0,49	130,42	-17,80	0,49	-0,49
73	-113,63	-0,49	-0,49	130,42	-17,80	0,49	0,49
74	-146,37	0,49	-0,49	130,42	-17,80	0,49	0,49
75	-113,63	0,49	-0,49	130,42	-17,80	0,49	-0,49
76	-146,37	-0,49	0,49	130,42	-17,80	0,49	0,49
77	-113,63	-0,49	0,49	130,42	-17,80	0,49	-0,49
78	-146,37	0,49	0,49	130,42	-17,80	0,49	-0,49
79	-113,63	0,49	0,49	130,42	-17,80	0,49	0,49

Appendix 3 Hand Calculations

A 3.1 ASME B31.8 Section VIII Pipe wall calculation

PIPE CODE CHECK ACCORDING TO ASME B31.8-2012
UNRESTRAINED PIPE

User Input	Calculation

Project Details:

Project Name:	Master Subsea Tie-in, Design Solutions and Optimization Methods
Project Number:	3200140
Pipeline Component:	water injection spool
Eingineer:	LKHA
Calculation Date:	31.03.2015
Revision:	1

Design Parameters:

Design Pressure:	$P_d =$	345	bar	***Differential pressure at 700m Waterdepth***
External Pressure:	$P_e =$	0	bar	

Note! FAT pressure (Ref. Sec. A847.2):
$P_d \cdot 1.4$ (Offshore platform piping)
$P_d \cdot 1.25$ (Offshore Pipeline system)

Construction Pressure 1: FAT	$P_{c1} =$	517,5	bar	***Project values based upon old standard***
Construction Pressure 2: Subsea Test	$P_{c2} =$	431,3	bar	***Project values based upon old standard***
Outside Diameter:	$D =$	168,3	mm	
Corrosion/Erosion Allowance:	$C =$	3	mm	

Material:

Material designation:		**X65**	
SMYS @ Construction Temperature:	S_C	450	MPa
Temperature derating factor: (Table 841.1.8-1)	T	1,0	
SMYS @ Design Temperature:	S_d	450,0	MPa

Allowable Stress values:

Utilisation Factor (Ref. Table A842.2.2-1):

Maximum Hoop Stress:	F_1	0,72	F_{1test}	0,72
Maximum Longitudinal Stress:	F_2	0,80	F_{2test}	0,80
Maximum Equivalent Stress:	F_3	0,90	F_{3test}	0,96 Ref. Statoil TR1230

Construction		**Operation**	
$F_{1test} \cdot S_C$	324,0 MPa	$F_1 \cdot S_d$	324,0 MPa
$F_{2test} \cdot S_C$	360,0 MPa	$F_2 \cdot S_d$	360,0 MPa
$F_{3test} \cdot S_C$	432,0 MPa	$F_3 \cdot S_d$	405,0 MPa

Note ! For platform piping and risers: F_1 is 0.5 and combined stress should include corrosion allowance.
Note! Hoop stress limit for pressure test may be based upon 0.96SMYS if t_{min} is used
Note! If Pipe is made from Duplex or Super Duplex (ref. Sec. 4, D302, DNV-RP-F112, Oct. 2008)
F2 = 0.68 (Coarse austenite spacing: 0.8*0.85=0.68)
F2 = 0.80 (Fine austenite spacing: 0.8*1.0 = 0.80)

Minimum Required Nominal Wall Thickness (Sec. A842.2.2):

Based on construction pressure P_{C1}: (FAT)	$t = P_{c1} \cdot D / 2 \cdot F_1 \cdot S_c$	t =	13,4 mm
Based on construction pressure P_{c2}: (Subsea Test)	$t = (P_{c2} - P_e) \cdot D / 2 \cdot F_1 \cdot S_c$	t =	11,2 mm
Based on differensial pressure ΔP_d: (Operation)	$t = (P_d - P_e) \cdot D / 2 \cdot F_1 \cdot S_d + C$	t =	12,0 mm
Chosen Wall Thickness:		$t_{nom} =$	18,3 mm
Resulting inside diameter:	$d = D - 2 \cdot t_{nom}$	d =	131,7 mm

Stress/Utilisations Calculations: Constructions Loads - Subsea Test :

Pipeline loads:

Axial Force:	F_A	14,80	kN	Reaction forces at XT side
Shear Force:	F_s	6,05	kN	$Sqrt(Fx^2+Fz^2)$
In-plane bending moment:	M_i	-51,20	kNm	
Out-of-plane bending moment:	M_o	-0,2	kNm	
Torque:	M_A	-4,7	kNm	

Hoop Stress: $D/t_{nom} \geq 30$ $S_h = (P_{c2} - P_e) \cdot D / 2 \cdot t_{nom}$ $S_h =$ 176,8 MPa

Sec. A842.2.2 $D/t_{nom} < 30$ $S_h = (P_{C2} - P_e) \cdot (D - t_{nom}) / 2 \cdot t_{nom}$

$UF = S_h / F_1 \cdot S_c$ UF = **0,55**

Longitudinal Stress (Unrestrained): $S_{L1} = (P_{c2} \cdot d^2 - P_e \cdot D^2)/(D^2 - d^2) + F_A \cdot 4/\pi \cdot (D^2 - d^2) + (M_i^2 + M_o^2)/Z$

Sec. 833.2/833.3/833.6 $S_{L1} =$ 244,9 MPa

$S_{L2} = (P_{c2} \cdot d^2 - P_e \cdot D^2)/(D^2 - d^2) + F_A \cdot 4/\pi \cdot (D^2 - d^2) - (M_i^2 + M_o^2)/Z$

$S_{L2} =$ -128,4 MPa

$UF = \max(S_{L1}, S_{L2})/F_2 \cdot S_C$ UF = **0,68**

Shear Stress: $S_S = M_A/2 \cdot Z + 2 \cdot F_s/A_s$

Sec. A842.2.2 $S_S =$ -6,62996 MPa

Equivalent stress: $S_{EQ1} = (S_h^2 + S_{L1}^2 - S_h \cdot S_{L1} + 3 \cdot S_S^2)^{0.5}$

$S_{EQ1} =$ 219,2 MPa

Sec. A842.2.2 $S_{EQ2} = (S_h^2 + S_{L2}^2 - S_h \cdot S_{L2} + 3 \cdot S_S^2)^{0.5}$

$S_{EQ2} =$ 265,6 MPa

$UF = \max(S_{EQ1}, S_{EQ2})/F_3 \cdot S_C$

UF **0,61**

Stress/Utilisations Calculations: Operational Loads :

Pipeline loads:

Axial Force:	F_A	18,30	kN	Reaction forces at manifold side
Shear Force:	F_s	33,70	kN	$Sqrt(Fx^2+Fz^2)$
In-plane bending moment:	M_i	64	kNm	
Out-of-plane bending moment:	M_o	-3,8	kNm	
Torque:	M_A	1,6	kNm	

Hoop Stress: $D/t_{nom} \geq 30$ $S_h = (P_d - P_e) \cdot D/2 \cdot (t_{nom} - C)$ $S_h =$ 169,1 MPa

Sec. A842.2.2 $D/t_{nom} < 30$ $S_h = (P_d - P_e) \cdot (D - t_{nom})/2 \cdot (t_{nom} - C)$

$UF = S_h / F_1 \cdot S_d$ UF = **0,52**

Longitudinal Stress (Unrestrained):

Sec. 833.2/833.3/833.6/

$S_{L1} = (P_d \cdot d^2 - P_e \cdot D^2)/(D^2 - d^2) + F_A \cdot 4/\pi \cdot (D^2 - d^2) + (M_i^2 + M_o^2)^{1/2} / Z_C$

$S_{L1} =$ 304,8 MPa

$S_{L2} = (P_d \cdot d^2 - P_e \cdot D^2)/(D^2 - d^2) + F_A \cdot 4/\pi \cdot (D^2 - d^2) - (M_i^2 + M_o^2)^{1/2} / Z_C$

$S_{L2} =$ -195,1 MPa

$UF = \max(S_{L1}, S_{L2})/F_2 \cdot S_d$ UF = **0,85**

Shear Stress: $S_S =$ $M_A/2 \cdot Z_C + 2 \cdot F_s/A_s$ $S_S =$ 10,91313672 MPa

Sec. A842.2.2

Equivalent stress: $S_{EQ1} =$ $(S_h^2 + S_{L1}^2 - S_h . S_{L1} + 3 \cdot S_S^2)^{0.5}$

Sec. A842.2.2 $S_{EQ1} =$ 265,2 MPa

$S_{EQ2} =$ $(S_h^2 + S_{L2}^2 - S_h . S_{L2} + 3 \cdot S_S^2)^{0.5}$

$S_{EQ2} =$ 316,2 MPa

$UF = \max(S_{EQ1}, S_{EQ2})/F_3 \cdot S_d$ UF **0,78**

A 3.2 Buoyancy Calculation

Calculation of buoyancy force for spool

Buoyancy type HCP 100 depth rating 625 m, chrusing depth=1000m

Seawater density $\rho_{seawater} := 1026 \cdot \frac{kg}{m^3}$

Density of buoyancy $\rho_{buoy} := 400 \cdot \frac{kg}{m^3}$

Length of spool $L_{spool} := 44 \cdot m$

Weight of spool pr length $m_{spool} := 68 \cdot \frac{kg}{m}$

Dry weigth of spool without connectors $M_{spool} := m_{spool} \cdot L_{spool}$

$M_{spool} = 2992 \cdot kg$

Wall thickness of pipe $w_t := 18.3 \cdot mm$

Outer/inner diameter of pipe $DO_{pipe} := 168.3 \cdot mm \quad DI_{pipe} := DO_{pipe} - 2w_t = 131.7 \cdot mm$

Cross section area of pipe $A_{cross} := \frac{\pi}{4} \cdot \left[DO_{pipe}{}^2 - \left(DO_{pipe} - 2 \cdot w_t\right)^2\right] = 8624 \cdot mm^2$

Submerged volume $V_{sub} := A_{cross} \cdot L_{spool} = 0.379 \cdot m^3$

Buoyancy force from submerged spool $F_{sub1} := V_{sub} \cdot \rho_{seawater} \cdot g = 3818\,N$

Ratio dry weight versus submerged weigth $R_{sub} := 1 - \frac{F_{sub1}}{\left(M_{spool} \cdot g\right)} = 0.87$

Submereged weight $m_{spoolsub} := M_{spool} \cdot R_{sub} = 2603\,kg$

In order to reduce the submerged weight by say 25% the following buoyancy force is required:

$$F_{sub2} := \frac{\left(M_{spool} \cdot R_{sub} \cdot g\right)}{1.25} = 20419\,N$$

$$F_{buoy} := \left(M_{spool} \cdot R_{sub} \cdot g - F_{sub2}\right) = 5105\,N$$

Required buoyancy element diameter calculation:

F_{sub} Submerged weight of element
F_{buoy} Buoyancy force required
F_{dry} Dry weigth of element
$V_{element}$ Volume of element
L Length of element

$$-F_{sub} = F_{buoy} - F_{dry} \qquad (1)$$

$$-F_{buoy} = F_{sub} - F_{dry} = V_{Element} \cdot \left(\rho_{seawater} - \rho_{buoy}\right) \cdot g \qquad (2)$$

Solved for the element volume gives the following equation:

$$V_{element} = \frac{-F_{buoy}}{\left(\rho_{seawater} - \rho_{buoy}\right) \cdot g} \qquad (3)$$

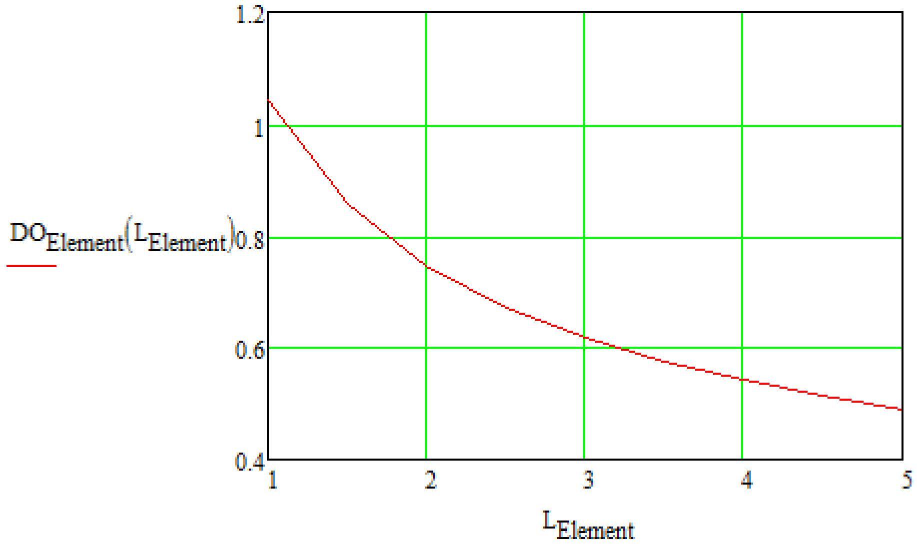
The volume is area x length. Use length L=1 meter $L_{Element} := 1 \cdot m, 1.5 \cdot m .. 5 \cdot m$

Volume for element $$V_{Element} = \frac{\pi}{4} \cdot \left(DO_{Element}{}^{2} - DO_{pipe}{}^{2}\right) \cdot L_{Element} \qquad (4)$$

By inserting (4) into equation (3) and solving for $DO_{Element}$ gives the following equation for required diameter pr meter element for a required uplift buoyancy force

$$DO_{Element}\left(L_{Element}\right) := \sqrt{\left[\frac{F_{buoy} \cdot 4}{g \cdot \pi \cdot \left(\rho_{seawater} - \rho_{buoy}\right) \cdot L_{Element}}\right] + DO_{pipe}{}^{2}}$$

Graph showing buoyancy element diameter as a function of the length



By using 5 meter as length the required outer diameter of the buoyancy element is equal to:

$DO_{Element}(5\cdot m) = 0.49\,m \qquad D_o := DO_{Element}(5\cdot m)$

Control of answer equation 2:

$$F_{buoy.check} := \left[\frac{\pi}{4}\cdot\left(D_o^{\,2} - DO_{pipe}^{\,2}\right)\right]\cdot 5\cdot m\cdot g\cdot\left(\rho_{seawater} - \rho_{buoy}\right) = 5105\,N$$

Which is eqaul to the required buoyancy uplift force hence OK

Calculating effective mass for spool without buoyancy element

The effective mass includes structural mass, added mass and the mass of the content

Added mass

$$m_{add} := \rho_{seawater} \cdot \pi \cdot \left(\frac{DO_{pipe}}{2}\right)^2$$

$$m_{add} = 22.825 \frac{kg}{m}$$

Structural mass-inclusive content

$$m_{struct} := \frac{m_{spoolsub}}{L_{spool}}$$

$$m_{struct} = 59.152 \frac{kg}{m}$$

Effective mass

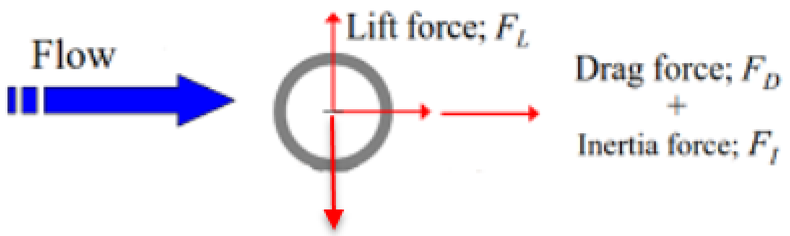
$$m_e := m_{struct} + m_{add} = 82 \frac{kg}{m}$$

A 3.3 Current Force Calculation

Drag force calculations

__Note to calculations__

-Inertia forces from waves is not included, as this is deepwater >700m and wave forces at this depth is assumed to have little effect.
-Lift force on pipe is assumed to be symmetrical on both sides of pipe and is neglected
-This calculation calculates max static dragforce
-The drag force is assumed to be perpendicular to pipe- worst direction.
-Variations in current,inertia and lift force requires VIV Check (Vortex Induced Vibration)



Formula (Morrison Equation): $F_d = 0.5\rho\, C_d \cdot OD \cdot (V_r)^2$

Seawater density: $\rho := 1026 \frac{kg}{m^3}$

Kinematic viscosity: $\nu := 1.05 \cdot 10^{-6} \cdot \frac{m^2}{s}$

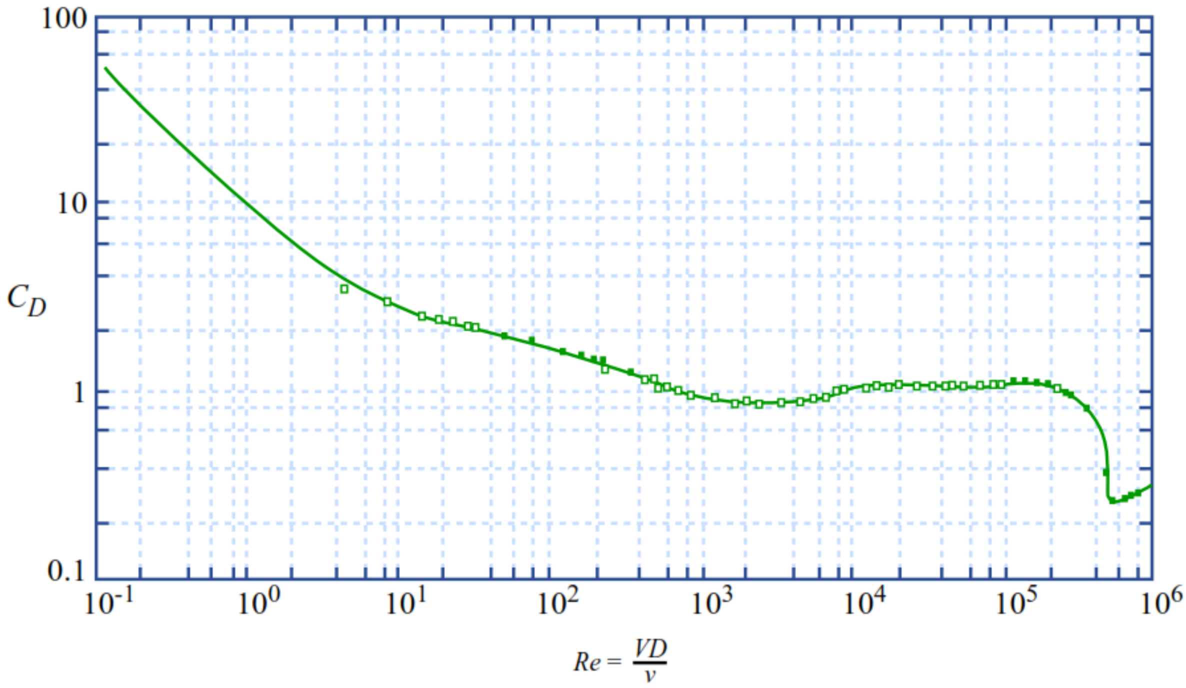
Current speed: $Vr := 0.7 \frac{m}{s}$

Outer diameter of pipe: $OD_{pipe} := 168.3 \cdot mm$

Isolation thickness: $t_{ins} := 0mm$

Outer diameter of buoyancy: $OD_{buoy} := 0mm$

NO STRAKES,INSULATION OR BUOYANCY APPLIED



Drag coefficient for a smooth circular cylinder as a function of Reynolds number.

Figure by MIT OpenCourseWare.

Reynolds numbers:

Reynolds number for buoyancy:

$$Re1 := \frac{OD_{buoy} \cdot Vr}{\nu} \qquad Re1 = 0$$

Reynolds number for pipe:

$$Re2 := \frac{OD_{pipe} \cdot Vr}{\nu} \qquad Re2 = 1.122 \times 10^{5}$$

Drag coefficients:

Drag coefficient for a smooth outer surface: $C_{D1} := 1.0$

Drag Force per unit length for different sections of pipe:

Bare pipe section:	$F1 := 0.5 \cdot \rho \cdot C_{D1} \cdot OD_{pipe} \cdot Vr^2$	$F1 = 42.306 \cdot \frac{N}{m}$
Insulated pipe section:	$F2 := 0.5 \cdot \rho \cdot C_{D1} \cdot \left(OD_{pipe} + 2 \cdot t_{ins}\right) \cdot Vr^2$	$F2 = 42.306 \cdot \frac{N}{m}$
Buoyancy section:	$F3 := 0.5 \cdot \rho \cdot C_{D1} \cdot OD_{buoy} \cdot Vr^2$	$F3 = 0$
Total length of pipe	$L_{pipe} := 45m$	
Total dragforce on pipesection	$F_{dragpipe} := F1 \cdot L_{pipe}$	$F_{dragpipe} = 1.904 \cdot kN$

A 3.4 Thick Wall Vessel Calculation

Calculation of hoop stress in thick cylindrical vessel based upon Lamès equations

Internal pressure $Pi := 345 \cdot bar$

Wall thickness $t_{wall} := 18.3 \cdot mm$

Outside and inside diameter $D_o := 168.3 \cdot mm$ $D_i := D_o - 2 \cdot t_{wall}$

Outside radius $R_o := \frac{D_o}{2} = 84.15 \cdot mm$

Inside radius $R_i := \frac{(D_o - 2 \cdot t_{wall})}{2} = 65.85 \cdot mm$

Thick wall check D/t<30 $\frac{D_o}{t_{wall}} = 9.197$ Less than 30 hence thick wall

$r_1 := R_i$

Max stress Inside of vessel

$$\sigma\theta\theta_{inside} := Pi \cdot \left[\frac{R_i^{\,2}}{\left(R_o^{\,2} - R_i^{\,2}\right)}\right] \cdot \left(1 + \frac{R_o^{\,2}}{r_1^{\,2}}\right) = 143\,MPa$$

$$\sigma rr_{inside} := Pi \cdot \left[\frac{R_i^{\,2}}{\left(R_o^{\,2} - R_i^{\,2}\right)}\right] \cdot \left(1 - \frac{R_o^{\,2}}{r_1^{\,2}}\right) = -34\,MPa$$

$r_2 := R_o$

Max stress outside of vessel

$$\sigma\theta\theta_{otside2} := Pi \cdot \left[\frac{R_i^{\,2}}{\left(R_o^{\,2} - R_i^{\,2}\right)}\right] \cdot \left(1 + \frac{R_o^{\,2}}{r_2^{\,2}}\right) = 109\,MPa$$

$$\sigma rr_{outside2} := Pi \cdot \left[\frac{R_i^{\,2}}{\left(R_o^{\,2} - R_i^{\,2}\right)}\right] \cdot \left(1 - \frac{R_o^{\,2}}{r_2^{\,2}}\right) = 0\,MPa$$

Comparison hoop stress formula DNV 1996 and ASME B31.8

$$\sigma_{mean} := \frac{(Pi - 0)\cdot(D_o - t_{wall})}{2t_{wall}} = 172.5\cdot MPa$$

Comparison Barlow Equation

$$\sigma_{barlow} := Pi\cdot\frac{D_o}{2t_{wall}} = 190\cdot MPa$$

The calculation shows that the difference in hoop stress between the DNV/ASME mean stress formula and the Lamés equations for the given wallthickness is small for the max stress calculation the difference is largest for the barlow wall formula with 11% difference

A 3.5 Fatigue calculation

DNV-RP-C203

In accordance to DNV-RP-C203 classification detail F1 is used for the SN-curve
Figure 2-5 S-N Curve in seawater with cathodic protection Ref. Section 2.10 Pipelines and Risers for limitations on welding and geometry

Design Fatigue factor $DFF := 10$

<u>SN Curve</u>

Parameters Design curve $a_1 := 10^{14.832}$ $m_1 := 5$ $t_{ref} := 25 \cdot mm$ $k := 0.25$

Pipe thickness $t := 18.3 \cdot mm$

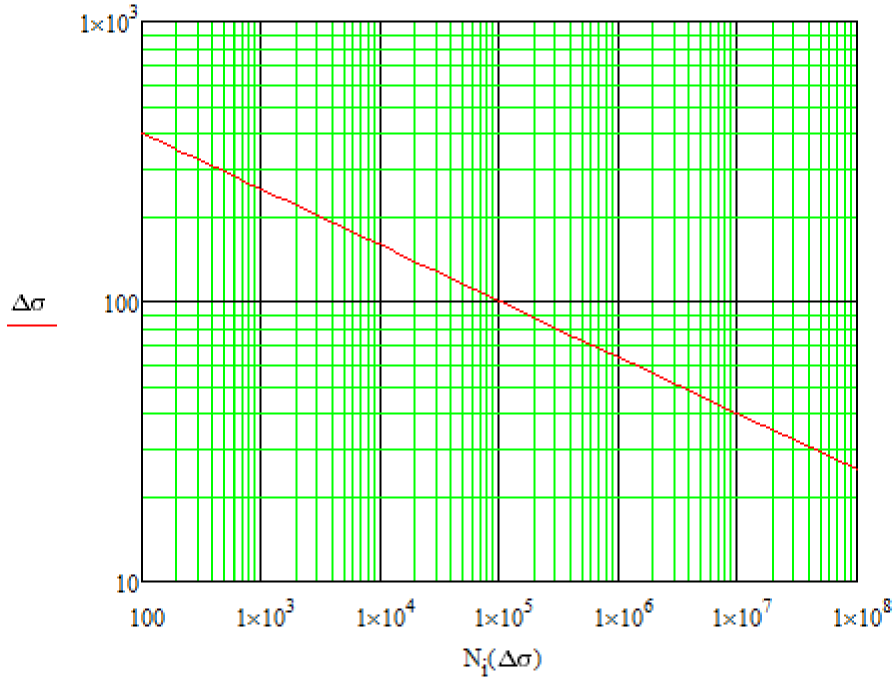
$\Delta\sigma := 10, 20 .. 500$

$$\log(N_i) = \log(a_1) - m_1 \cdot \log\left[\Delta\sigma \cdot \left(\frac{t}{t_{ref}}\right)^k\right]$$

Max allowable cycles

$$N_i(\Delta\sigma) := \frac{a_1}{\left[\Delta\sigma \cdot \left(\frac{t}{t_{ref}}\right)^k\right]^{m_1}}$$

S-N Curve



Max calculated cycles and stress range

$n_1 := 2.4 \cdot 10^8$ $\Delta\sigma_1 := 38$

$n_2 := 2.8 \cdot 10^7$ $\Delta\sigma_2 := 11$

Max allowable cycles from S-N curve

$$N_1(\Delta\sigma_1) := \frac{a_1}{\left[\Delta\sigma_1 \cdot \left(\frac{t}{t_{ref}}\right)^k\right]^{m_1}} \qquad N_1(\Delta\sigma_1) = 1.27 \times 10^7$$

$$N_2(\Delta\sigma_1) := \frac{a_1}{\left[\Delta\sigma_2 \cdot \left(\frac{t}{t_{ref}}\right)^k\right]^{m_1}} \qquad N_2(\Delta\sigma_2) = 6.23 \times 10^9$$

Summation of cummulative damage then becomes

$$D := \left(\frac{n_1}{N_1(\Delta\sigma_1)}\right) + \left(\frac{n_2}{N_2(\Delta\sigma_2)}\right) = 19$$

The max allowable usage factor with a DFF of 10 equals:

$$\eta := \frac{1}{10} = 0.1 \qquad D < \eta = 0$$

Hence the spool fails for 25 years fatigue life

VIV suppresion strakes are required for the spool in order to supress the vibration

A 3.6 Local Buckling-External Overpressure

Local buckling collapse of a pipe in accodance with DNV-OS-F101,2013

Ref. Sec. 5 & 13 D400, D700

$MPa := 1 \cdot 10^6 \cdot Pa$

Youngs modulus		$E := 2.0 \cdot 10^5 \cdot MPa$
Yield strength at specified temp.		$f_y := 450 \cdot MPa$
Poisson ratio		$\nu := 0.28$
Material factor:	(Table 5-2 and 5-3)	$\gamma_m := 1.15$
Safety class factor		$\gamma_{SC} := 1.138$
Load factors ULS	(Table 4-4 and 4-5)	$\gamma_F := 1.2$ $\gamma_c := 1.07$ $\gamma_p := 1.05$
Nominell Diameter of pipe and thickness		$D := 168.3 \cdot mm$ $t_{nom} := 18.3 \cdot mm$
Max and min measured diameter		$D_{max} := 172.9 \cdot mm$
		$D_{min} := 167 \cdot mm$
Ovality:		$f_0 := \frac{(D_{max} - D_{min})}{D}$ $f_0 = 0.04$
External over pressure		$p := 9 \cdot MPa$ $p_e := p \cdot \gamma_p$
Bend wall thinning and tolerance		$t_{thinning} := t_{nom} \cdot (10\% + 12.5\%) = 4.12\,mm$
Corrosion allowance		$t_{corr} := 3 \cdot mm$
Thickness of pipe $t_1 = t - t_{fab} - t_{corr}$	(Table 5-6)	$t_1 := t_{nom} - t_{thinning} - t_{corr} = 11.18\,mm$
Fabrication factor (seamless pipe)	(Table 5-5)	$\alpha_{fab} := 1.0$
Elastic capacity pressure:		$p_{el} := \frac{\left[2 \cdot E \cdot \left(\frac{t_1}{D}\right)^3\right]}{(1 - \nu^2)}$ $p_{el} = 127.32\,MPa$
Plastic capasity pressure		$p_p := 2 \cdot f_y \cdot \alpha_{fab} \cdot \left(\frac{t_1}{D}\right)$ $p_p = 59.8\,MPa$

The external collapse pressure are expressed as a 3'rd degre polynom and has the following expression:

$$(p_c - p_{el}) \cdot (p_c^{\,2} - p_p^{\,2}) = p_c \cdot p_{el} \cdot p_p \cdot f_0 \cdot \frac{D}{t_1}$$

The solution is as follows

$$p_c = y - \frac{b}{3}$$ (Section 13 D700)

Where the following parameters apply:

$$b := -p_{el} \qquad c := -\left(p_p^{\,2} + p_p \cdot p_{el} \cdot f_0 \cdot \frac{D}{t_1}\right)$$

$$d := p_{el} \cdot p_p^{\,2} \qquad u := \frac{1}{3} \cdot \left(\frac{-1}{3} \cdot b^2 + c\right) \qquad \nu_2 := \frac{1}{2} \cdot \left(\frac{2}{27} \cdot b^3 - \frac{1}{3} \cdot b \cdot c + d\right)$$

$$\Phi := \operatorname{acos}\left(\left(\frac{-\nu_2}{\sqrt{-u^3}}\right)\right) \qquad y := -2 \cdot \sqrt{-u} \cdot \cos\left(\left(\frac{\Phi}{3} + \frac{60}{180} \cdot \pi\right)\right)$$

$$p_c := y - \frac{b}{3} \qquad p_c = 41\,\mathrm{MPa}$$

The external pressure at any point along the pipeline shall meet the following criteria:

$$p_e = 9\,\mathrm{MPa} \qquad p_e \le \frac{p_c}{1.1 \cdot \gamma_m \cdot \gamma_{SC}} = 1$$

1 =True
0=False

Max Utilisation factor: $UF := \frac{p_e \cdot (1.1 \cdot \gamma_m \cdot \gamma_{SC})}{p_c}$ $UF = 0.33$

$$\text{Check} := \begin{cases} \text{"OK"} & \text{if } UF \le 1 \\ \text{"FAIL"} & \text{otherwise} \end{cases} \qquad \text{Check} = \text{"OK"}$$

Appendix 4 Bently AutoPIPE

A 4.1 AutoPIPE Features

Feature	AutoPIPE	AutoPIPE Plus	Nuclear
Hanger	✓	✓	✓
Static Linear	✓	✓	✓
Static Nonlinear	✓	✓	✓
Modal	✓	✓	✓
Response Spectrum (Uniform & Multiple Support) (SRSS combination method Standard version only)	✓ (Note 3)	✓	✓
Harmonic		✓	✓
Force Spectrum		✓	✓
Time History		✓	✓
SAM		✓	✓
Buried pipe		✓	✓
NUREG combinations and Code case 411 spectrum		✓	✓
Static correction - Missing mass correction and ZPA		✓	✓
50 Response Spectrum load cases		✓	✓
Static earthquake	✓	✓	✓
Wind - ASCE, UBC and User Profile	✓	✓	✓
Thermal Bowing	✓	✓	✓
Wave loading and buoyancy		✓	✓
Fluid Transient Loads		✓	✓
Relief Valve Loads		✓	✓
Thermal Transient Analysis			✓
Fatigue Analysis (class 1)			✓
High Energy Leakage and Crack Criteria (ASME Class 1, 2, 3)			✓
ASME B31.1, B31.3, B31.4, and B31.8	✓ (Note 2)	✓	✓
European piping code EN13480	✓ (Note 2)	✓	✓
B31.4 Offshore, A31.8 Offshore & CSA_Z662 Offshore codes		✓	✓
ASME III Class 2 and Class 3 (multiple years)			✓
ASME III Class 1 (multiple years)			✓
JSME S NC1-PPC			✓
Canadian piping codes		✓	✓
International piping codes		✓	✓
KHK Level 2 piping code		Note 1	✓
Analysis Sets for multiple static analyses	✓	✓	✓
General piping code	✓	✓	✓
Rotating Equipment reports	✓	✓	✓
Large model size	✓	✓	✓
Beam elements for modeling frames and supports	✓	✓	✓
Material and Component Library utilities	✓	✓	✓
STAAD Structural Libraries (17 countries)	✓	✓	✓

Plus features only **Nuclear features only**

Maximum defined static and dynamic load cases:

Load Cases	Standard 9.4	Plus 9.4	Nuclear 9.4
Gravity	1	1	1
Hydrotest	1	1	1
Thermal	5	100	100
Pressure	5	100	100
Static Earthquake	5	10	10
Wind	5	10	10
User	5	140	140
Response Spectrum	5	50	50
Harmonic	Not Available	10	10
Seismic Anchor Movement	Not Available	10	10
Force Spectrum	Not Available	10	10
Time History	Not Available	50	50
Static Analysis Cases	27 [Note 1]	82 [Note 1]	82 [Note 1]

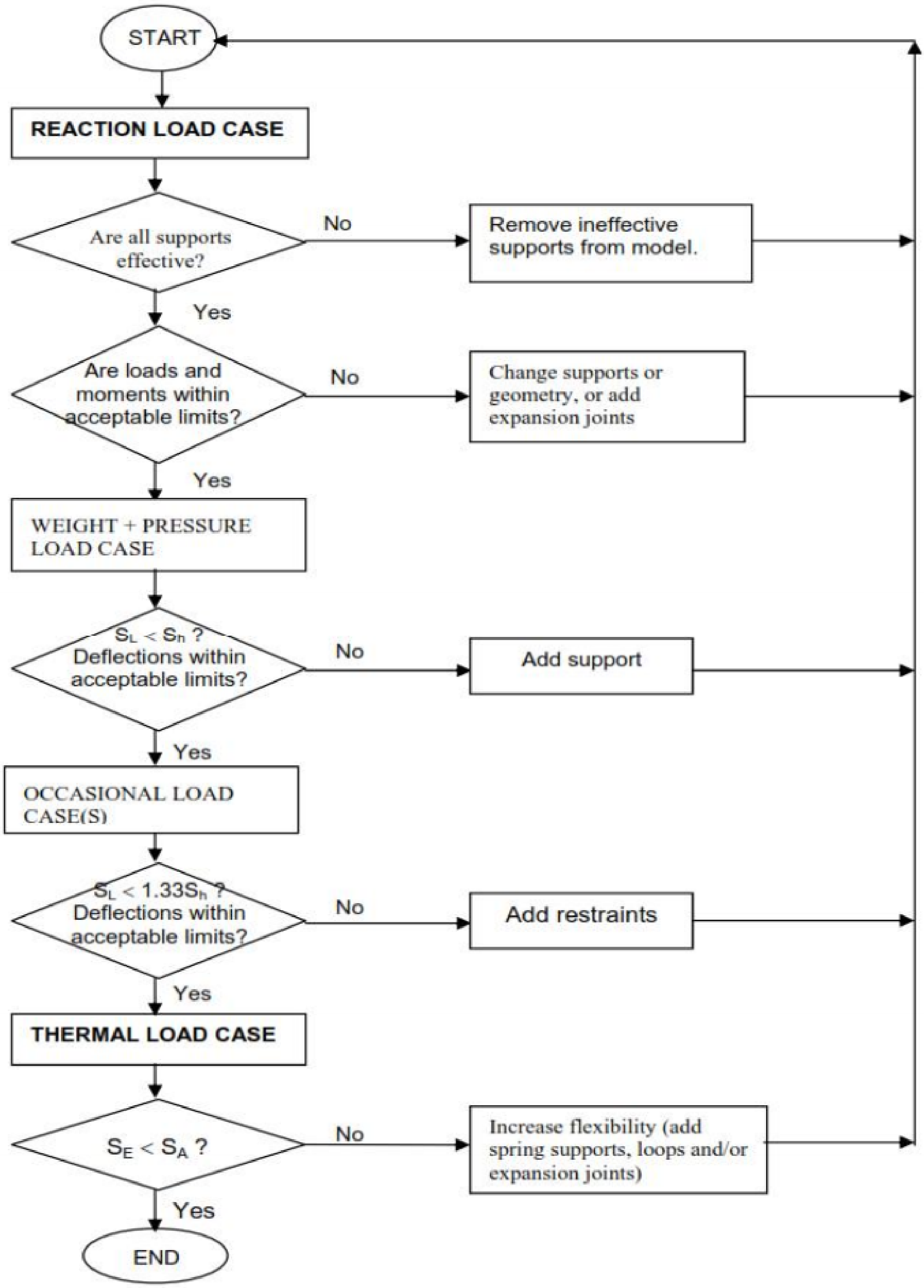
Note 1: Maximum number of load cases that can be analyzed in a single analysis set during a static analysis run in v9.1 and later. However an unlimited number of analysis sets can be run in a single static analysis in v9.1 and later.

= Gravity (1) + Hydrotest (1) + Thermal (20) + Pressure (20) + Static Earthquake (10) + Wind (10) + User (20)

= 82 cases for Plus & Nuclear (27 for Standard)

Up to 100 different thermal loadings can be defined and analyzed in a single static analysis. Only 20 thermal load cases per analysis set e.g. if want to run 50 thermal cases then define across 3 analysis sets. Since each analysis set can have analyze up to 82 static cases, so literally 100's of loads can be analyzed in different scenarios with different options, linear, non-linear, hot or cold modulus etc in the same static analysis run.

GENERAL FLOW CHART FOR PIPING FLEXIBILITY ANALYSIS



A 4.2 AutoPIPE Stress output

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      *                             *******   **  *******   *******
     ***               **           **    **  **  **    **  **
    ** **            ******         **    **  **  **    **  **
   **   **    **   **  **   *****   *******   **  *******   *****
  *********   **   **  **  **   **  **        **  **        **
 **       **  **   **  **  **   **  **        **  **        **
**         **  *****   **   *****   **        **  **        *******
```

```
                     Pipe Stress Analysis and Design Program

                                Version: 09.04.00.19

                                Edition: Standard

                           Developed and Maintained by

                          BENTLEY SYSTEMS, INCORPORATED
                          1600 Riviera Ave., Suite 300
                             Walnut Creek, CA  94596
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          ************************************************************
          **                                                        **
          **              AUTOPIPE SYSTEM INFORMATION               **
          **                                                        **
          ************************************************************


          SYSTEM NAME : 6INCH SPOOL


          PROJECT ID  : 6INCH SPOOL


          PREPARED BY  : ______________________________
                         LKA

          CHECKED BY   : ______________________________


          1ST APPROVER : ______________________________


          2ND APPROVER : ______________________________


          PIPING CODE            : ASME B31.8

          YEAR                   : 2010

          VERTICAL AXIS          : Y

          AMBIENT TEMPERATURE    :    4.0 deg C

          COMPONENT LIBRARY      : AUTOPIPE

          MATERIAL LIBRARY       : AUTOB318

          MODEL REVISION NUMBER  :    31
```

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                    A N A L Y S I S   S U M M A R Y


   Current model revision number : 31

     Static - Analysis set number .................. 1
              Date and Time of analysis ............ May 21, 2015   1:59 PM
              Model Revision Number ................ 31
              Number of load cases ................. 5
              Load cases analyzed .................. GR T1 T2 P1 P2
              Description .......................... Analysis Set No.1
              Gaps/Friction/Soil considered ........ No
              Hanger design run .................... No
              Cut short included ................... No
              Thermal bowing included .............. No
              Include Bourdon rotational effect ..... No
              Pipe radius for Bourdon calculation ... Mean
              Weight of contents included .......... Yes
              Pressure stiffening case ............. None
              Hot modulus case ..................... None
              Water elevation for buoyancy loads .... Not considered
              Use corroded thickness in analysis .... No
              Rigid stiffness factor ............... 1000.0
```

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CODE COMPLIANCE COMBINATIONS

<Description> Combination	Category	Method	Case/Combination	Factor	M/S	K-Factor	Allowable (N/mm2)	D/A/P
GR + Max P{1}	Sustain	Sum	GR[1]	1.00			405.00	Y N Y
			Max Long	1.00				
Max Range	Expansion	Sum	Temp. Range	1.00			Automatic	Y Y Y
Amb to T1{1}	Expansion	Sum	T1[1]	1.00			Automatic	Y Y Y
Amb to T2{1}	Expansion	Sum	T2[1]	1.00			Automatic	Y Y Y
Max P{1}	Hoop	Sum	Max Hoop	1.00			Automatic	Y Y Y
GRTP1{1}	Rest-Fun	Sum	Max Long	1.00			Automatic	Y N Y
			Max Hoop	1.00				
			GR[1]	1.00				
			T1[1]	1.00				
			P1[1]	1.00				
GRTP2{1}	Rest-Fun	Sum	Max Long	1.00			Automatic	Y Y Y
			Max Hoop	1.00				
			GR[1]	1.00				
			T2[1]	1.00				
			P2[1]	1.00				

Notes:

D/A/P: [D]efault/[A]uto-Update/[P]rint options (Y=Yes, N=No)

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                              NON-CODE COMBINATIONS

  <Description>
  Combination               Method       Case/Combination     Factor  D/A/P
  -----------------------   -------- ------------------------ ------ -------
  Gravity{1}                Sum      GR[1]                      1.00  Y Y Y

    <4.00 deg C>
  Thermal 1{1}              Sum      T1[1]                      1.00  Y Y Y

    <100.00 deg C>
  Thermal 2{1}              Sum      T2[1]                      1.00  Y Y Y

  Pressure 1{1}             Sum      P1[1]                      1.00  Y Y Y

  Pressure 2{1}             Sum      P2[1]                      1.00  Y Y Y

  GRTP1{1}                  Sum      GR[1]                      1.00  Y Y Y
                                     T1[1]                      1.00
                                     P1[1]                      1.00

  GRTP2{1}                  Sum      GR[1]                      1.00  Y Y Y
                                     T2[1]                      1.00
                                     P2[1]                      1.00

  Notes:

  D/A/P: [D]efault/[A]uto-Update/[P]rint options (Y=Yes, N=No)
```

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   CODE COMPLIANCE

   Y - Factor ............................ 0.00
   Weld efficiency factor ................ 1.00
   Range reduction factor ................ 1.00
   Design factor ......................... 0.72
   Temperature derating factor ........... 1.00
   Design Pressure Factor ................ 1.00
   Minimum stress ratio used in reports... 0.00
   Number of stress points per span ...... 0
   Include corrosion in stress calcs. .... Y
   Include torsion in code stress ........ N
   Include axial force in code stress .... Y
   Include sustain load margin ........... Only if allowable stress is exceeded
   Set sustained SIF=1 no bends .......... N
   Set sustained/occasional SIF = 1 ...... N
   Set sustained/occasional SIF = 0.75i .. N
   Apply cold/hot modulus ratio .......... N
   Disable auto code combinations ........ N
   Disable auto non-code combinations .... N
   No. of thermal ranges to report ....... 0
   Include Max Range combination ......... Y
   Total stress .......................... Octahedral
   Direct shear .......................... None
   Longitudinal pressure calculation ..... PD/4t
   Inc. Axial Str and Pcase in Sustained.. Y
```

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                                       D I S P L A C E M E N T S

 Point      Load                      TRANSLATIONS (mm  )         ROTATIONS (deg  )
 name    combination                    X       Y       Z          X       Y       Z
 ------  -----------------------     ------  ------  ------     ------  ------  ------

 *** Segment A  begin ***

 A00     Gravity{1}                    0.00 -160.00    0.00       0.93    0.00    0.93
         Thermal 1{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Thermal 2{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Pressure 1{1}                 0.00    0.00    0.00       0.00    0.00    0.00
         Pressure 2{1}                 0.00    0.00    0.00       0.00    0.00    0.00
         GRTP1{1}                      0.00 -160.00    0.00       0.93    0.00    0.93
         GRTP2{1}                      0.00 -160.00    0.00       0.93    0.00    0.93

 A01 N   Gravity{1}                  134.57 -160.06  102.76       0.89    0.18   -1.81
         Thermal 1{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Thermal 2{1}                 -7.41    7.45    0.00       0.00    0.00    0.02
         Pressure 1{1}                -0.72    0.73    0.00       0.00    0.00    0.00
         Pressure 2{1}                -0.72    0.73    0.00       0.00    0.00    0.00
         GRTP1{1}                    133.85 -159.33  102.76       0.89    0.18   -1.81
         GRTP2{1}                    126.43 -151.88  102.76       0.89    0.18   -1.79

 A01 F   Gravity{1}                  148.46 -173.46  108.45       0.92    0.19   -1.58
         Thermal 1{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Thermal 2{1}                 -6.98    7.93    0.00       0.00    0.00   -0.02
         Pressure 1{1}                -0.68    0.78    0.00       0.00    0.00    0.00
         Pressure 2{1}                -0.68    0.78    0.00       0.00    0.00    0.00
         GRTP1{1}                    147.78 -172.68  108.45       0.92    0.19   -1.59
         GRTP2{1}                    140.80 -164.75  108.45       0.92    0.19   -1.60

 A02 N   Gravity{1}                  148.48 -213.85  101.02       1.02    0.21   -0.53
         Thermal 1{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Thermal 2{1}                 -4.69    5.40    0.00       0.00    0.00   -0.12
         Pressure 1{1}                -0.46    0.53    0.00       0.00    0.00   -0.01
         Pressure 2{1}                -0.46    0.53    0.00       0.00    0.00   -0.01
         GRTP1{1}                    148.02 -213.32  101.02       1.02    0.21   -0.54
         GRTP2{1}                    143.33 -207.92  101.02       1.02    0.21   -0.66

 A02 F   Gravity{1}                  146.97 -216.53   91.09       1.05    0.20    0.00
         Thermal 1{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Thermal 2{1}                 -5.40    3.79    0.00       0.00    0.00   -0.17
         Pressure 1{1}                -0.53    0.37    0.00       0.00    0.00   -0.02
         Pressure 2{1}                -0.53    0.37    0.00       0.00    0.00   -0.02
         GRTP1{1}                    146.45 -216.16   91.09       1.05    0.20   -0.02
         GRTP2{1}                    141.05 -212.37   91.09       1.05    0.20   -0.18

 A03 N   Gravity{1}                  228.49 -216.56  -16.52       1.12    0.16    0.73
         Thermal 1{1}                  0.00    0.00    0.00       0.00    0.00    0.00
         Thermal 2{1}                -27.86   -2.34    0.00       0.00    0.00   -0.22
         Pressure 1{1}                -2.72   -0.23    0.00       0.00    0.00   -0.02
         Pressure 2{1}                -2.72   -0.23    0.00       0.00    0.00   -0.02
         GRTP1{1}                    225.77 -216.79  -16.52       1.12    0.16    0.71
         GRTP2{1}                    197.90 -219.12  -16.52       1.12    0.16    0.49
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D I S P L A C E M E N T S

Point name	Load combination	TRANSLATIONS (mm) X	Y	Z	ROTATIONS (deg) X	Y	Z
A03 F	Gravity{1}	233.44	-212.34	-26.69	1.12	0.15	0.41
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	-29.09	-4.51	0.00	0.00	0.00	-0.20
	Pressure 1{1}	-2.84	-0.44	0.00	0.00	0.00	-0.02
	Pressure 2{1}	-2.84	-0.44	0.00	0.00	0.00	-0.02
	GRTP1{1}	230.59	-212.78	-26.69	1.12	0.15	0.39
	GRTP2{1}	201.50	-217.29	-26.69	1.12	0.15	0.19
A07	Gravity{1}	233.49	-250.98	-37.82	1.07	0.03	-0.65
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	-22.20	-16.66	0.00	0.00	0.00	-0.03
	Pressure 1{1}	-2.17	-1.63	0.00	0.00	0.00	0.00
	Pressure 2{1}	-2.17	-1.63	0.00	0.00	0.00	0.00
	GRTP1{1}	231.32	-252.61	-37.82	1.07	0.03	-0.65
	GRTP2{1}	209.12	-269.27	-37.82	1.07	0.03	-0.69
A06	Gravity{1}	233.55	-281.60	-30.76	1.03	-0.15	0.34
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	-14.82	-14.05	0.00	0.00	0.00	0.06
	Pressure 1{1}	-1.45	-1.37	0.00	0.00	0.00	0.01
	Pressure 2{1}	-1.45	-1.37	0.00	0.00	0.00	0.01
	GRTP1{1}	232.10	-282.98	-30.76	1.03	-0.15	0.35
	GRTP2{1}	217.28	-297.02	-30.76	1.03	-0.15	0.41
A08	Gravity{1}	233.61	-159.75	-6.93	0.98	-0.23	1.61
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	-7.44	-4.92	0.00	0.00	0.00	0.08
	Pressure 1{1}	-0.73	-0.48	0.00	0.00	0.00	0.01
	Pressure 2{1}	-0.73	-0.48	0.00	0.00	0.00	0.01
	GRTP1{1}	232.88	-160.23	-6.93	0.98	-0.23	1.61
	GRTP2{1}	225.45	-165.15	-6.93	0.98	-0.23	1.69
A04 N	Gravity{1}	233.67	20.33	11.85	0.93	-0.07	1.29
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	-0.55	0.75	0.00	0.00	0.00	0.01
	Pressure 1{1}	-0.05	0.07	0.00	0.00	0.00	0.00
	Pressure 2{1}	-0.05	0.07	0.00	0.00	0.00	0.00
	GRTP1{1}	233.61	20.40	11.85	0.93	-0.07	1.29
	GRTP2{1}	233.06	21.16	11.85	0.93	-0.07	1.31
A04 F	Gravity{1}	242.81	30.00	4.85	0.93	-0.02	1.05
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	-0.01	0.33	0.00	0.00	0.00	0.00
	Pressure 1{1}	0.00	0.03	0.00	0.00	0.00	0.00
	Pressure 2{1}	0.00	0.03	0.00	0.00	0.00	0.00
	GRTP1{1}	242.81	30.03	4.85	0.93	-0.02	1.05
	GRTP2{1}	242.81	30.36	4.85	0.93	-0.02	1.05

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D I S P L A C E M E N T S

Point name	Load combination	TRANSLATIONS (mm) X	Y	Z	ROTATIONS (deg) X	Y	Z
A05	Gravity{1}	248.00	30.00	0.00	0.93	0.00	0.93
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Thermal 2{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Pressure 1{1}	0.00	0.00	0.00	0.00	0.00	0.00
	Pressure 2{1}	0.00	0.00	0.00	0.00	0.00	0.00
	GRTP1{1}	248.00	30.00	0.00	0.93	0.00	0.93
	GRTP2{1}	248.00	30.00	0.00	0.93	0.00	0.93

*** Segment A end ***

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                                     R E S T R A I N T   R E A C T I O N S

 Point     Load                           FORCES (N     )            MOMENTS (N.m    )
 name   combination                   X      Y      Z    Result    X       Y       Z    Result
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 A00     Anchor    Tag No.: <None>
         Gravity{1}                15645 -17237  -1149  23307   -4018    1748  -88455   88564
         Thermal 1{1}                  0      0      0      0       0       0       0       0
         Thermal 2{1}              -1175    162      0   1186       0       0    4268    4268
         Pressure 1{1}              -115     16      0    116       0       0     417     417
         Pressure 2{1}              -115     16      0    116       0       0     417     417
         GRTP1{1}                  15530 -17221  -1149  23218   -4018    1748  -88038   88147
         GRTP2{1}                  14355 -17059  -1149  22325   -4018    1748  -83770   83885

 A05     Anchor    Tag No.: <None>
         Gravity{1}               -15645  -7395   -833  17325    -138   -4756   37180   37483
         Thermal 1{1}                  0      0      0      0       0       0       0       0
         Thermal 2{1}               1175   -162      0   1186       0       0     573     573
         Pressure 1{1}               115    -16      0    116       0       0      56      56
         Pressure 2{1}               115    -16      0    116       0       0      56      56
         GRTP1{1}                 -15530  -7410   -833  17228    -138   -4756   37236   37539
         GRTP2{1}                 -14355  -7572   -833  16251    -138   -4756   37809   38107
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G L O B A L F O R C E S & M O M E N T S

Point name	Load combination	FORCES (N) X	Y	Z	Result	MOMENTS (N.m) X	Y	Z	Result
*** Segment A begin ***									
A00	Gravity{1}	-15645	17237	1149	23307	4018	-1748	88455	88564
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-4268	4268
	Pressure 1{1}	115	-16	0	116	0	0	-417	417
	Pressure 2{1}	115	-16	0	116	0	0	-417	417
	GRTP1{1}	-15530	17221	1149	23218	4018	-1748	88038	88147
	GRTP2{1}	-14355	17059	1149	22325	4018	-1748	83770	83885
A01 N-	Gravity{1}	-15645	13405	841	20620	-2751	-1748	-18011	18303
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	3731	3731
	Pressure 1{1}	115	-16	0	116	0	0	365	365
	Pressure 2{1}	115	-16	0	116	0	0	365	365
	GRTP1{1}	-15530	13389	841	20522	-2751	-1748	-17646	17944
	GRTP2{1}	-14355	13228	841	19538	-2751	-1748	-13915	14292
A01 N+	Gravity{1}	-15645	13405	841	20620	-2751	-1748	-18011	18303
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	3731	3731
	Pressure 1{1}	115	-16	0	116	0	0	365	365
	Pressure 2{1}	115	-16	0	116	0	0	365	365
	GRTP1{1}	-15530	13389	841	20522	-2751	-1748	-17646	17944
	GRTP2{1}	-14355	13228	841	19538	-2751	-1748	-13915	14292
A01 F-	Gravity{1}	-15645	13001	808	20358	-3130	-1373	-31169	31356
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	4342	4342
	Pressure 1{1}	115	-16	0	116	0	0	425	425
	Pressure 2{1}	115	-16	0	116	0	0	425	425
	GRTP1{1}	-15530	12985	808	20260	-3130	-1373	-30744	30934
	GRTP2{1}	-14355	12823	808	19265	-3130	-1373	-26403	26623
A01 F+	Gravity{1}	-15645	13001	808	20358	-3130	-1373	-31169	31356
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	4342	4342
	Pressure 1{1}	115	-16	0	116	0	0	425	425
	Pressure 2{1}	115	-16	0	116	0	0	425	425
	GRTP1{1}	-15530	12985	808	20260	-3130	-1373	-30744	30934
	GRTP2{1}	-14355	12823	808	19265	-3130	-1373	-26403	26623
A02 N-	Gravity{1}	-15645	11826	714	19625	-3130	214	-57063	57150
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	4679	4679
	Pressure 1{1}	115	-16	0	116	0	0	457	457
	Pressure 2{1}	115	-16	0	116	0	0	457	457
	GRTP1{1}	-15530	11810	714	19524	-3130	214	-56606	56693
	GRTP2{1}	-14355	11649	714	18500	-3130	214	-51927	52022

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G L O B A L   F O R C E S   &   M O M E N T S

Point name	Load combination	FORCES (N) X	Y	Z	Result	MOMENTS (N.m) X	Y	Z	Result
A02 N+	Gravity{1}	-15645	11826	714	19625	-3130	214	-57063	57150
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	4679	4679
	Pressure 1{1}	115	-16	0	116	0	0	457	457
	Pressure 2{1}	115	-16	0	116	0	0	457	457
	GRTP1{1}	-15530	11810	714	19524	-3130	214	-56606	56693
	GRTP2{1}	-14355	11649	714	18500	-3130	214	-51927	52022
A02 F-	Gravity{1}	-15645	11422	681	19383	-2813	535	-55251	55325
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	4215	4215
	Pressure 1{1}	115	-16	0	116	0	0	412	412
	Pressure 2{1}	115	-16	0	116	0	0	412	412
	GRTP1{1}	-15530	11406	681	19281	-2813	535	-54839	54914
	GRTP2{1}	-14355	11245	681	18247	-2813	535	-50623	50704
A02 F+	Gravity{1}	-15645	11422	681	19383	-2813	535	-55251	55325
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	4215	4215
	Pressure 1{1}	115	-16	0	116	0	0	412	412
	Pressure 2{1}	115	-16	0	116	0	0	412	412
	GRTP1{1}	-15530	11406	681	19281	-2813	535	-54839	54914
	GRTP2{1}	-14355	11245	681	18247	-2813	535	-50623	50704
A03 N-	Gravity{1}	-15645	8271	428	17702	289	535	32300	32305
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-2362	2362
	Pressure 1{1}	115	-16	0	116	0	0	-231	231
	Pressure 2{1}	115	-16	0	116	0	0	-231	231
	GRTP1{1}	-15530	8255	428	17593	289	535	32069	32075
	GRTP2{1}	-14355	8094	428	16485	289	535	29707	29713
A03 N+	Gravity{1}	-15645	8271	428	17702	289	535	32300	32305
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-2362	2362
	Pressure 1{1}	115	-16	0	116	0	0	-231	231
	Pressure 2{1}	115	-16	0	116	0	0	-231	231
	GRTP1{1}	-15530	8255	428	17593	289	535	32069	32075
	GRTP2{1}	-14355	8094	428	16485	289	535	29707	29713
A03 F-	Gravity{1}	-15645	7867	395	17516	479	721	35787	35798
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-2825	2825
	Pressure 1{1}	115	-16	0	116	0	0	-276	276
	Pressure 2{1}	115	-16	0	116	0	0	-276	276
	GRTP1{1}	-15530	7851	395	17406	479	721	35511	35522
	GRTP2{1}	-14355	7689	395	16289	479	721	32686	32697

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G L O B A L   F O R C E S   &   M O M E N T S

Point name	Load combination	FORCES (N) X	Y	Z	Result	MOMENTS (N.m) X	Y	Z	Result
A03 F+	Gravity{1}	-15645	7867	395	17516	479	721	35787	35798
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-2825	2825
	Pressure 1{1}	115	-16	0	116	0	0	-276	276
	Pressure 2{1}	115	-16	0	116	0	0	-276	276
	GRTP1{1}	-15530	7851	395	17406	479	721	35511	35522
	GRTP2{1}	-14355	7689	395	16289	479	721	32686	32697
A07 -	Gravity{1}	-15645	4323	110	16232	479	2309	-2569	3488
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-1809	1809
	Pressure 1{1}	115	-16	0	116	0	0	-177	177
	Pressure 2{1}	115	-16	0	116	0	0	-177	177
	GRTP1{1}	-15530	4308	110	16117	479	2309	-2746	3620
	GRTP2{1}	-14355	4146	110	14942	479	2309	-4555	5129
A07 +	Gravity{1}	-15645	4323	110	16232	479	2309	-2569	3488
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-1809	1809
	Pressure 1{1}	115	-16	0	116	0	0	-177	177
	Pressure 2{1}	115	-16	0	116	0	0	-177	177
	GRTP1{1}	-15530	4308	110	16117	479	2309	-2746	3620
	GRTP2{1}	-14355	4146	110	14942	479	2309	-4555	5129
A06 -	Gravity{1}	-15645	523	-196	15655	479	2019	-18925	19038
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-718	718
	Pressure 1{1}	115	-16	0	116	0	0	-70	70
	Pressure 2{1}	115	-16	0	116	0	0	-70	70
	GRTP1{1}	-15530	507	-196	15540	479	2019	-18995	19108
	GRTP2{1}	-14355	345	-196	14360	479	2019	-19713	19822
A06 +	Gravity{1}	-15645	523	-196	15655	479	2019	-18925	19038
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	-718	718
	Pressure 1{1}	115	-16	0	116	0	0	-70	70
	Pressure 2{1}	115	-16	0	116	0	0	-70	70
	GRTP1{1}	-15530	507	-196	15540	479	2019	-18995	19108
	GRTP2{1}	-14355	345	-196	14360	479	2019	-19713	19822
A08 -	Gravity{1}	-15645	-3278	-502	15993	479	-336	-9626	9644
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	372	372
	Pressure 1{1}	115	-16	0	116	0	0	36	36
	Pressure 2{1}	115	-16	0	116	0	0	36	36
	GRTP1{1}	-15530	-3294	-502	15884	479	-336	-9589	9607
	GRTP2{1}	-14355	-3455	-502	14773	479	-336	-9217	9236

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G L O B A L   F O R C E S   &   M O M E N T S

Point name	Load combination	FORCES (N) X	Y	Z	Result	MOMENTS (N.m) X	Y	Z	Result
A08 +	Gravity{1}	-15645	-3278	-502	15993	479	-336	-9626	9644
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	372	372
	Pressure 1{1}	115	-16	0	116	0	0	36	36
	Pressure 2{1}	115	-16	0	116	0	0	36	36
	GRTP1{1}	-15530	-3294	-502	15884	479	-336	-9589	9607
	GRTP2{1}	-14355	-3455	-502	14773	479	-336	-9217	9236
A04 N-	Gravity{1}	-15645	-6821	-787	17086	479	-4391	22152	22588
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	1389	1389
	Pressure 1{1}	115	-16	0	116	0	0	136	136
	Pressure 2{1}	115	-16	0	116	0	0	136	136
	GRTP1{1}	-15530	-6837	-787	16987	479	-4391	22288	22721
	GRTP2{1}	-14355	-6999	-787	15990	479	-4391	23677	24085
A04 N+	Gravity{1}	-15645	-6821	-787	17086	479	-4391	22152	22588
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	1389	1389
	Pressure 1{1}	115	-16	0	116	0	0	136	136
	Pressure 2{1}	115	-16	0	116	0	0	136	136
	GRTP1{1}	-15530	-6837	-787	16987	479	-4391	22288	22721
	GRTP2{1}	-14355	-6999	-787	15990	479	-4391	23677	24085
A04 F-	Gravity{1}	-15645	-7226	-819	17253	110	-4756	32486	32833
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	926	926
	Pressure 1{1}	115	-16	0	116	0	0	91	91
	Pressure 2{1}	115	-16	0	116	0	0	91	91
	GRTP1{1}	-15530	-7241	-819	17155	110	-4756	32577	32922
	GRTP2{1}	-14355	-7403	-819	16172	110	-4756	33503	33839
A04 F+	Gravity{1}	-15645	-7226	-819	17253	110	-4756	32486	32833
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	926	926
	Pressure 1{1}	115	-16	0	116	0	0	91	91
	Pressure 2{1}	115	-16	0	116	0	0	91	91
	GRTP1{1}	-15530	-7241	-819	17155	110	-4756	32577	32922
	GRTP2{1}	-14355	-7403	-819	16172	110	-4756	33503	33839
A05	Gravity{1}	-15645	-7395	-833	17325	-138	-4756	37180	37483
	Thermal 1{1}	0	0	0	0	0	0	0	0
	Thermal 2{1}	1175	-162	0	1186	0	0	573	573
	Pressure 1{1}	115	-16	0	116	0	0	56	56
	Pressure 2{1}	115	-16	0	116	0	0	56	56
	GRTP1{1}	-15530	-7410	-833	17228	-138	-4756	37236	37539
	GRTP2{1}	-14355	-7572	-833	16251	-138	-4756	37809	38107

*** Segment A end ***

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            G E N E R A L   P I P E   S T R E S S   R E P O R T
                                            (Stress in N/mm2  )
 Point     Load                        Hoop   Longitudinal   Shear     Principal    Total
 name   combination                   Stress   Max     Min   Stress   Max    Min   Stress  Loc
 ------ ----------------------       ------ ------ ------  ------ ------ ------  ------  ---

 *** Segment A  begin ***

 A00    SIFI= 1.00   SIFO= 1.00
        Gravity{1}                     0.00  341.20 -345.90    3.39  341.24-345.93  345.95   93
        Thermal 1{1}                   0.00    0.00    0.00    0.00    0.00   0.00    0.00  270
        Thermal 2{1}                   0.00   16.58  -16.54    0.00   16.58 -16.54   16.58   90
        Pressure 1{1}                190.19   71.70   68.46    0.00  190.19  68.46  166.85  270
        Pressure 2{1}                190.19   71.70   68.46    0.00  190.19  68.46  166.85  270
        GRTP1{1}                     190.19  409.67 -274.20    3.39  409.72-274.23  404.40   93
        GRTP2{1}                     190.19  393.15 -257.64    3.39  393.20-257.67  389.34   93

 A01 N- SIFI= 1.00   SIFO= 1.00
        Gravity{1}                     0.00   68.86  -72.52    3.39   69.03 -72.68   72.75  279
        Thermal 1{1}                   0.00    0.00    0.00    0.00    0.00   0.00    0.00  270
        Thermal 2{1}                   0.00   14.50  -14.45    0.00   14.50 -14.45   14.50  270
        Pressure 1{1}                190.19   71.50   68.67    0.00  190.19  68.67  166.81   90
        Pressure 2{1}                190.19   71.50   68.67    0.00  190.19  68.67  166.81   90
        GRTP1{1}                     190.19  137.54   -1.04    3.39  190.40  -1.10  190.80  279
        GRTP2{1}                     190.19  123.31   13.24    3.39  190.36  13.18  184.02  281

 A01 N+ SIFI= 1.00   SIFO= 1.00
        Gravity{1}                     0.00   68.86  -72.52    3.39   69.03 -72.68   72.75  189
        Thermal 1{1}                   0.00    0.00    0.00    0.00    0.00   0.00    0.00  270
        Thermal 2{1}                   0.00   14.50  -14.45    0.00   14.50 -14.45   14.50  180
        Pressure 1{1}                190.19   71.50   68.67    0.00  190.19  68.67  166.81    0
        Pressure 2{1}                190.19   71.50   68.67    0.00  190.19  68.67  166.81    0
        GRTP1{1}                     190.19  137.54   -1.04    3.39  190.40  -1.10  190.80  189
        GRTP2{1}                     190.19  123.31   13.24    3.39  190.36  13.18  184.02  191

 A01 F- SIFI= 1.00   SIFO= 1.00
        Gravity{1}                     0.00  123.18 -118.92    6.07  123.48-119.23  123.63  357
        Thermal 1{1}                   0.00    0.00    0.00    0.00    0.00   0.00    0.00  270
        Thermal 2{1}                   0.00   16.69  -17.01    0.00   16.69 -17.01   17.01    0
        Pressure 1{1}                190.19   71.71   68.42    0.00  190.19  68.42  166.85    0
        Pressure 2{1}                190.19   71.71   68.42    0.00  190.19  68.42  166.85    0
        GRTP1{1}                     190.19  191.60  -47.21    6.07  197.01 -47.36  217.92  177
        GRTP2{1}                     190.19  174.61  -30.54    6.07  192.27 -30.71  207.42  177

 A01 F+ SIFI= 1.00   SIFO= 1.00
        Gravity{1}                     0.00  123.18 -118.92    6.07  123.48-119.23  123.63  177
        Thermal 1{1}                   0.00    0.00    0.00    0.00    0.00   0.00    0.00  270
        Thermal 2{1}                   0.00   16.69  -17.01    0.00   16.69 -17.01   17.01  180
        Pressure 1{1}                190.19   71.71   68.42    0.00  190.19  68.42  166.85  180
        Pressure 2{1}                190.19   71.71   68.42    0.00  190.19  68.42  166.85  180
        GRTP1{1}                     190.19  191.60  -47.21    6.07  197.01 -47.36  217.92  357
        GRTP2{1}                     190.19  174.61  -30.54    6.07  192.27 -30.71  207.42  357

 A02 N- SIFI= 1.00   SIFO= 1.00
        Gravity{1}                     0.00  223.53 -219.27    6.07  223.70-219.44  223.78  180
        Thermal 1{1}                   0.00    0.00    0.00    0.00    0.00   0.00    0.00  270
        Thermal 2{1}                   0.00   17.99  -18.31    0.00   17.99 -18.31   18.31  180
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G E N E R A L   P I P E   S T R E S S   R E P O R T
(Stress in N/mm2)

Point name	Load combination	Hoop Stress	Longitudinal Max	Longitudinal Min	Shear Stress	Principal Max	Principal Min	Total Stress	Loc
	Pressure 1{1}	190.19	71.84	68.29	0.00	190.19	68.29	166.87	180
	Pressure 2{1}	190.19	71.84	68.29	0.00	190.19	68.29	166.87	180
	GRTP1{1}	190.19	291.82	-147.43	6.07	292.18	-147.54	293.36	0
	GRTP2{1}	190.19	273.51	-129.44	6.07	273.95	-129.55	278.66	0
A02 N+ SIFI= 1.00 SIFO= 1.00									
	Gravity{1}	0.00	223.53	-219.27	6.07	223.70	-219.44	223.78	0
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	17.99	-18.31	0.00	17.99	-18.31	18.31	0
	Pressure 1{1}	190.19	71.84	68.29	0.00	190.19	68.29	166.87	0
	Pressure 2{1}	190.19	71.84	68.29	0.00	190.19	68.29	166.87	0
	GRTP1{1}	190.19	291.82	-147.43	6.07	292.18	-147.54	293.36	180
	GRTP2{1}	190.19	273.51	-129.44	6.07	273.95	-129.55	278.66	180
A02 F- SIFI= 1.00 SIFO= 1.00									
	Gravity{1}	0.00	216.20	-213.09	1.04	216.21	-213.09	216.21	357
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	16.33	-16.38	0.00	16.33	-16.38	16.38	0
	Pressure 1{1}	190.19	71.68	68.48	0.00	190.19	68.48	166.84	0
	Pressure 2{1}	190.19	71.68	68.48	0.00	190.19	68.48	166.84	0
	GRTP1{1}	190.19	284.68	-141.42	1.04	284.69	-141.42	288.22	177
	GRTP2{1}	190.19	268.33	-125.11	1.04	268.34	-125.11	274.99	177
A02 F+ SIFI= 1.00 SIFO= 1.00									
	Gravity{1}	0.00	216.20	-213.09	1.04	216.21	-213.09	216.21	87
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	16.33	-16.38	0.00	16.33	-16.38	16.38	90
	Pressure 1{1}	190.19	71.68	68.48	0.00	190.19	68.48	166.84	90
	Pressure 2{1}	190.19	71.68	68.48	0.00	190.19	68.48	166.84	90
	GRTP1{1}	190.19	284.68	-141.42	1.04	284.69	-141.42	288.22	267
	GRTP2{1}	190.19	268.33	-125.11	1.04	268.34	-125.11	274.99	267
A03 N- SIFI= 1.00 SIFO= 1.00									
	Gravity{1}	0.00	126.45	-124.20	1.04	126.46	-124.21	126.46	269
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	9.14	-9.19	0.00	9.14	-9.19	9.19	270
	Pressure 1{1}	190.19	70.97	69.18	0.00	190.19	69.18	166.73	270
	Pressure 2{1}	190.19	70.97	69.18	0.00	190.19	69.18	166.73	270
	GRTP1{1}	190.19	195.63	-53.22	1.04	195.82	-53.23	221.65	89
	GRTP2{1}	190.19	186.45	-44.08	1.04	190.46	-44.09	215.64	89
A03 N+ SIFI= 1.00 SIFO= 1.00									
	Gravity{1}	0.00	126.45	-124.20	1.04	126.46	-124.21	126.46	359
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	9.14	-9.19	0.00	9.14	-9.19	9.19	0
	Pressure 1{1}	190.19	70.97	69.18	0.00	190.19	69.18	166.73	0
	Pressure 2{1}	190.19	70.97	69.18	0.00	190.19	69.18	166.73	0
	GRTP1{1}	190.19	195.63	-53.22	1.04	195.82	-53.23	221.65	179
	GRTP2{1}	190.19	186.45	-44.08	1.04	190.46	-44.09	215.64	179

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G E N E R A L   P I P E   S T R E S S   R E P O R T
(Stress in N/mm2)

Point name	Load combination	Hoop Stress	Longitudinal Max	Longitudinal Min	Shear Stress	Principal Max	Principal Min	Total Stress	Loc
A03 F-	SIFI= 1.00 SIFO= 1.00								
	Gravity{1}	0.00	141.01	-136.75	0.93	141.02	-136.75	141.02	359
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	10.80	-11.12	0.00	10.80	-11.12	11.12	0
	Pressure 1{1}	190.19	71.13	68.99	0.00	190.19	68.99	166.76	0
	Pressure 2{1}	190.19	71.13	68.99	0.00	190.19	68.99	166.76	0
	GRTP1{1}	190.19	210.00	-65.61	0.93	210.05	-65.62	230.12	179
	GRTP2{1}	190.19	198.88	-54.81	0.93	198.98	-54.82	222.72	179
A03 F+	SIFI= 1.00 SIFO= 1.00								
	Gravity{1}	0.00	141.01	-136.75	0.93	141.02	-136.75	141.02	359
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	10.80	-11.12	0.00	10.80	-11.12	11.12	0
	Pressure 1{1}	190.19	71.13	68.99	0.00	190.19	68.99	166.76	0
	Pressure 2{1}	190.19	71.13	68.99	0.00	190.19	68.99	166.76	0
	GRTP1{1}	190.19	210.00	-65.61	0.93	210.05	-65.62	230.12	179
	GRTP2{1}	190.19	198.88	-54.81	0.93	198.98	-54.82	222.72	179
A07	SIFI= 1.00 SIFO= 1.00								
	Gravity{1}	0.00	15.54	-11.27	0.93	15.59	-11.35	15.62	222
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	6.86	-7.18	0.00	6.86	-7.18	7.18	0
	Pressure 1{1}	190.19	70.75	69.38	0.00	190.19	69.38	166.70	0
	Pressure 2{1}	190.19	70.75	69.38	0.00	190.19	69.38	166.70	0
	GRTP1{1}	190.19	86.12	58.27	0.93	190.20	58.27	168.78	40
	GRTP2{1}	190.19	91.85	52.22	0.93	190.20	52.21	170.20	27
A06	SIFI= 1.00 SIFO= 1.00								
	Gravity{1}	0.00	75.98	-71.71	0.93	75.99	-71.72	75.99	186
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	2.63	-2.95	0.00	2.63	-2.95	2.95	0
	Pressure 1{1}	190.19	70.34	69.79	0.00	190.19	69.79	166.64	0
	Pressure 2{1}	190.19	70.34	69.79	0.00	190.19	69.79	166.64	0
	GRTP1{1}	190.19	146.31	-1.92	0.93	190.21	-1.92	191.16	6
	GRTP2{1}	190.19	148.92	-4.85	0.93	190.21	-4.86	192.66	6
A08	SIFI= 1.00 SIFO= 1.00								
	Gravity{1}	0.00	39.50	-35.24	0.93	39.52	-35.26	39.54	178
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	1.28	-1.60	0.00	1.28	-1.60	1.60	180
	Pressure 1{1}	190.19	70.20	69.92	0.00	190.19	69.92	166.62	180
	Pressure 2{1}	190.19	70.20	69.92	0.00	190.19	69.92	166.62	180
	GRTP1{1}	190.19	109.42	34.97	0.93	190.20	34.96	175.35	358
	GRTP2{1}	190.19	107.82	36.25	0.93	190.20	36.24	174.91	358
A04 N-	SIFI= 1.00 SIFO= 1.00								
	Gravity{1}	0.00	89.75	-85.49	0.93	89.76	-85.50	89.77	11
	Thermal 1{1}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	270
	Thermal 2{1}	0.00	5.23	-5.55	0.00	5.23	-5.55	5.55	180
	Pressure 1{1}	190.19	70.59	69.54	0.00	190.19	69.54	166.68	180
	Pressure 2{1}	190.19	70.59	69.54	0.00	190.19	69.54	166.68	180

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            G E N E R A L   P I P E   S T R E S S   R E P O R T
                                        (Stress in N/mm2  )
 Point     Load                        Hoop   Longitudinal   Shear     Principal    Total
 name   combination                   Stress   Max     Min   Stress   Max     Min   Stress  Loc
 ------ ----------------------        ------ ------ ------  ------ ------ ------  ------  ---
        GRTP1{1}                      190.19 160.33 -15.94    0.93 190.22 -15.95  198.64  191
        GRTP2{1}                      190.19 165.47 -21.40    0.93 190.22 -21.40  201.74  191

 A04 N+ SIFI= 1.00   SIFO= 1.00
        Gravity{1}                      0.00  89.75 -85.49    0.93  89.76 -85.50   89.77  191
        Thermal 1{1}                    0.00   0.00   0.00    0.00   0.00   0.00    0.00  270
        Thermal 2{1}                    0.00   5.23  -5.55    0.00   5.23  -5.55    5.55    0
        Pressure 1{1}                 190.19  70.59  69.54    0.00 190.19  69.54  166.68    0
        Pressure 2{1}                 190.19  70.59  69.54    0.00 190.19  69.54  166.68    0
        GRTP1{1}                      190.19 160.33 -15.94    0.93 190.22 -15.95  198.64   11
        GRTP2{1}                      190.19 165.47 -21.40    0.93 190.22 -21.40  201.74   11

 A04 F- SIFI= 1.00   SIFO= 1.00
        Gravity{1}                      0.00 125.06 -127.03   9.23 125.74-127.70  128.03  360
        Thermal 1{1}                    0.00   0.00   0.00    0.00   0.00   0.00    0.00  270
        Thermal 2{1}                    0.00   3.57  -3.61    0.00   3.57  -3.61    3.61    0
        Pressure 1{1}                 190.19  70.43  69.73    0.00 190.19  69.73  166.65    0
        Pressure 2{1}                 190.19  70.43  69.73    0.00 190.19  69.73  166.65    0
        GRTP1{1}                      190.19 195.49 -57.30    9.23 202.44 -57.65  224.96  360
        GRTP2{1}                      190.19 199.06 -60.92    9.23 204.86 -61.26  227.43  360

 A04 F+ SIFI= 1.00   SIFO= 1.00
        Gravity{1}                      0.00 125.06 -127.03   9.23 125.74-127.70  128.03   90
        Thermal 1{1}                    0.00   0.00   0.00    0.00   0.00   0.00    0.00  270
        Thermal 2{1}                    0.00   3.57  -3.61    0.00   3.57  -3.61    3.61   90
        Pressure 1{1}                 190.19  70.43  69.73    0.00 190.19  69.73  166.65   90
        Pressure 2{1}                 190.19  70.43  69.73    0.00 190.19  69.73  166.65   90
        GRTP1{1}                      190.19 195.49 -57.30    9.23 202.44 -57.65  224.96   90
        GRTP2{1}                      190.19 199.06 -60.92    9.23 204.86 -61.26  227.43   90

 A05    SIFI= 1.00   SIFO= 1.00
        Gravity{1}                      0.00 143.25 -145.26   9.23 143.84-145.85  146.14   90
        Thermal 1{1}                    0.00   0.00   0.00    0.00   0.00   0.00    0.00  270
        Thermal 2{1}                    0.00   2.20  -2.25    0.00   2.20  -2.25    2.25   90
        Pressure 1{1}                 190.19  70.29  69.86    0.00 190.19  69.86  166.63   90
        Pressure 2{1}                 190.19  70.29  69.86    0.00 190.19  69.86  166.63   90
        GRTP1{1}                      190.19 213.54 -75.40    9.23 216.75 -75.72  237.60   90
        GRTP2{1}                      190.19 215.74 -77.65    9.23 218.73 -77.97  239.22   90

 *** Segment A  end   ***
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                     BENTLEY
01:59 PM                                                                   AutoPIPE Standard 9.4.0.19 RESULT PAGE  19
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )             (Stress in N/mm2   )
 Point     Load                        In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                    Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------     ------- ------- ------  ------  ------ ----  ------ ------

 A00    SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}              4018.3888037.95   3.39    2.35  341.93 SUST  434.75 405.00**

        TR:Amb to T2{1}               0.00 4267.86   0.00    0.02   16.56 DISP   16.58   3.33**

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00   3.33

        Amb to T2{1}                  0.00 4267.86   0.00    0.02   16.56 DISP   16.58   3.33**

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                   4018.3888037.95   0.00   67.73  341.93 RFun  404.36 405.00
                                                                          LONG  409.67 360.00**

        GRTP2{1}                   4018.3883770.08   0.00   67.75  325.39 RFun  389.29 405.00
                                                                          LONG  393.15 405.00

 A02 N- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             56606.00  214.04   6.07   -2.13  219.63 SUST  317.17 405.00

        TR:Amb to T2{1}            4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 120.90

        Amb to T2{1}               4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                  56606.00  214.04   0.00   72.20  219.63 RFun  293.17 405.00
                                                                          LONG  291.82 360.00

        GRTP2{1}                  51927.22  214.04   0.00   72.04  201.47 RFun  278.47 405.00
                                                                          LONG  273.51 405.00

 A02 N+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             56606.00  214.04   6.07   -2.13  219.63 SUST  317.17 405.00

        TR:Amb to T2{1}            4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 120.90

        Amb to T2{1}               4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                  56606.00  214.04   0.00   72.20  219.63 RFun  293.17 405.00
                                                                          LONG  291.82 360.00

        GRTP2{1}                  51927.22  214.04   0.00   72.04  201.47 RFun  278.47 405.00
                                                                          LONG  273.51 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                 BENTLEY
01:59 PM                                                               AutoPIPE Standard 9.4.0.19 RESULT PAGE  20
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )            (Stress in N/mm2   )
 Point     Load                          In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                      Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A02 N- SIFI= 1.00   SIFO= 1.00
         GR + Max P{1}           56606.00  214.04   6.07   -2.13  219.63 SUST  317.17 405.00

        TR:Amb to T2{1}           4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 120.90

        Amb to T2{1}              4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 56606.00  214.04   0.00   72.20  219.63 RFun  293.17 405.00
                                                                         LONG  291.82 360.00

        GRTP2{1}                 51927.22  214.04   0.00   72.04  201.47 RFun  278.47 405.00
                                                                         LONG  273.51 405.00

 A02 N+ SIFI= 1.00   SIFO= 1.00
         GR + Max P{1}           56606.00  214.04   6.07   -2.13  219.63 SUST  317.17 405.00

        TR:Amb to T2{1}           4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 120.90

        Amb to T2{1}              4678.78    0.00   0.00    0.16   18.15 DISP   18.31 120.90

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 56606.00  214.04   0.00   72.20  219.63 RFun  293.17 405.00
                                                                         LONG  291.82 360.00

        GRTP2{1}                 51927.22  214.04   0.00   72.04  201.47 RFun  278.47 405.00
                                                                         LONG  273.51 405.00

 A02 F- SIFI= 1.00   SIFO= 1.00
         GR + Max P{1}           54838.87 2813.04   1.04   -1.56  213.05 SUST  309.71 405.00

        TR:Amb to T2{1}           4215.45    0.00   0.00    0.02   16.36 DISP   16.38 128.37

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 128.37

        Amb to T2{1}              4215.45    0.00   0.00    0.02   16.36 DISP   16.38 128.37

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 54838.87 2813.04   0.00   71.63  213.05 RFun  288.21 405.00
                                                                         LONG  284.68 360.00

        GRTP2{1}                 50623.42 2813.04   0.00   71.61  196.72 RFun  274.98 405.00
                                                                         LONG  268.33 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                      BENTLEY
01:59 PM                                                                    AutoPIPE Standard 9.4.0.19 RESULT PAGE  21
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )          (Stress in N/mm2   )
 Point     Load                      In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                  Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------     ------- ------- ------  ------  ------ ----  ------ ------

 A02 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}              2813.0454838.87   1.04   -1.56  213.05 SUST  309.71 405.00

        TR:Amb to T2{1}               0.00 4215.45   0.00    0.02   16.36 DISP   16.38 128.37

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 128.37

        Amb to T2{1}                  0.00 4215.45   0.00    0.02   16.36 DISP   16.38 128.37

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                   2813.0454838.87   0.00   71.63  213.05 RFun  288.21 405.00
                                                                          LONG  284.68 360.00

        GRTP2{1}                   2813.0450623.42   0.00   71.61  196.72 RFun  274.98 405.00
                                                                          LONG  268.33 405.00

 A02 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             54838.87 2813.04   1.04   -1.56  213.05 SUST  309.71 405.00

        TR:Amb to T2{1}            4215.45    0.00   0.00    0.02   16.36 DISP   16.38 128.37

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 128.37

        Amb to T2{1}               4215.45    0.00   0.00    0.02   16.36 DISP   16.38 128.37

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                  54838.87 2813.04   0.00   71.63  213.05 RFun  288.21 405.00
                                                                          LONG  284.68 360.00

        GRTP2{1}                  50623.42 2813.04   0.00   71.61  196.72 RFun  274.98 405.00
                                                                          LONG  268.33 405.00

 A02 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}              2813.0454838.87   1.04   -1.56  213.05 SUST  309.71 405.00

        TR:Amb to T2{1}               0.00 4215.45   0.00    0.02   16.36 DISP   16.38 128.37

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 128.37

        Amb to T2{1}                  0.00 4215.45   0.00    0.02   16.36 DISP   16.38 128.37

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                   2813.0454838.87   0.00   71.63  213.05 RFun  288.21 405.00
                                                                          LONG  284.68 360.00

        GRTP2{1}                   2813.0450623.42   0.00   71.61  196.72 RFun  274.98 405.00
                                                                          LONG  268.33 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                      BENTLEY
01:59 PM                                                                    AutoPIPE Standard 9.4.0.19 RESULT PAGE  22
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )           (Stress in N/mm2   )
 Point     Load                          In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                      Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A05    SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}                    138.2337236.07   9.23    1.01  144.47 SUST  239.73 405.00

        TR:Amb to T2{1}                    0.00  573.09   0.00    0.02    2.22 DISP    2.25 198.34

        Amb to T1{1}                       0.00    0.00   0.00    0.00    0.00 DISP    0.00 198.34

        Amb to T2{1}                       0.00  573.09   0.00    0.02    2.22 DISP    2.25 198.34

        Max P{1}                                                               HOOP  158.94 324.00

        GRTP1{1}                         138.2337236.07   0.00   69.07  144.47 RFun  237.06 405.00
                                                                               LONG  213.54 360.00

        GRTP2{1}                         138.2337809.15   0.00   69.05  146.70 RFun  238.68 405.00
                                                                               LONG  215.74 405.00

 A03 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}                  35511.09  720.65   0.93   -2.13  137.81 SUST  235.03 405.00

        TR:Amb to T2{1}                 2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Amb to T1{1}                       0.00    0.00   0.00    0.00    0.00 DISP    0.00 203.04

        Amb to T2{1}                    2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Max P{1}                                                               HOOP  158.94 324.00

        GRTP1{1}                       35511.09  720.65   0.00   72.20  137.81 RFun  230.12 405.00
                                                                               LONG  210.00 360.00

        GRTP2{1}                       32685.64  720.65   0.00   72.04  126.85 RFun  222.71 405.00
                                                                               LONG  198.88 405.00

 A03 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}                  35511.09  720.65   0.93   -2.13  137.81 SUST  235.03 405.00

        TR:Amb to T2{1}                 2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Amb to T1{1}                       0.00    0.00   0.00    0.00    0.00 DISP    0.00 203.04

        Amb to T2{1}                    2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Max P{1}                                                               HOOP  158.94 324.00

        GRTP1{1}                       35511.09  720.65   0.00   72.20  137.81 RFun  230.12 405.00
                                                                               LONG  210.00 360.00

        GRTP2{1}                       32685.64  720.65   0.00   72.04  126.85 RFun  222.71 405.00
                                                                               LONG  198.88 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                 BENTLEY
01:59 PM                                                               AutoPIPE Standard 9.4.0.19 RESULT PAGE  23
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )           (Stress in N/mm2   )
 Point     Load                       In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                   Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A03 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            35511.09  720.65   0.93   -2.13  137.81 SUST  235.03 405.00

        TR:Amb to T2{1}           2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 203.04

        Amb to T2{1}              2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 35511.09  720.65   0.00   72.20  137.81 RFun  230.12 405.00
                                                                         LONG  210.00 360.00

        GRTP2{1}                 32685.64  720.65   0.00   72.04  126.85 RFun  222.71 405.00
                                                                         LONG  198.88 405.00

 A03 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            35511.09  720.65   0.93   -2.13  137.81 SUST  235.03 405.00

        TR:Amb to T2{1}           2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 203.04

        Amb to T2{1}              2825.44    0.00   0.00    0.16   10.96 DISP   11.12 203.04

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 35511.09  720.65   0.00   72.20  137.81 RFun  230.12 405.00
                                                                         LONG  210.00 360.00

        GRTP2{1}                 32685.64  720.65   0.00   72.04  126.85 RFun  222.71 405.00
                                                                         LONG  198.88 405.00

 A04 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            32576.97  109.61   9.23    0.98  126.40 SUST  221.84 405.00

        TR:Amb to T2{1}            925.71    0.00   0.00    0.02    3.59 DISP    3.61 216.23

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 216.23

        Amb to T2{1}               925.71    0.00   0.00    0.02    3.59 DISP    3.61 216.23

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 32576.97  109.61   0.00   69.09  126.40 RFun  224.40 405.00
                                                                         LONG  195.49 360.00

        GRTP2{1}                 33502.68  109.61   0.00   69.07  129.99 RFun  226.87 405.00
                                                                         LONG  199.06 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                      BENTLEY
01:59 PM                                                                    AutoPIPE Standard 9.4.0.19 RESULT PAGE  24
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )             (Stress in N/mm2   )
 Point     Load                            In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                        Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A04 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}              109.6332576.97   9.23    0.98  126.40 SUST  221.84 405.00

        TR:Amb to T2{1}              0.00  925.71   0.00    0.02    3.59 DISP    3.61 216.23

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 216.23

        Amb to T2{1}                 0.00  925.71   0.00    0.02    3.59 DISP    3.61 216.23

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                   109.6332576.97   0.00   69.09  126.40 RFun  224.40 405.00
                                                                         LONG  195.49 360.00

        GRTP2{1}                   109.6333502.68   0.00   69.07  129.99 RFun  226.87 405.00
                                                                         LONG  199.06 405.00

 A04 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            32576.97  109.61   9.23    0.98  126.40 SUST  221.84 405.00

        TR:Amb to T2{1}            925.71    0.00   0.00    0.02    3.59 DISP    3.61 216.23

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 216.23

        Amb to T2{1}               925.71    0.00   0.00    0.02    3.59 DISP    3.61 216.23

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 32576.97  109.61   0.00   69.09  126.40 RFun  224.40 405.00
                                                                         LONG  195.49 360.00

        GRTP2{1}                 33502.68  109.61   0.00   69.07  129.99 RFun  226.87 405.00
                                                                         LONG  199.06 405.00

 A04 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}              109.6332576.97   9.23    0.98  126.40 SUST  221.84 405.00

        TR:Amb to T2{1}              0.00  925.71   0.00    0.02    3.59 DISP    3.61 216.23

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 216.23

        Amb to T2{1}                 0.00  925.71   0.00    0.02    3.59 DISP    3.61 216.23

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                   109.6332576.97   0.00   69.09  126.40 RFun  224.40 405.00
                                                                         LONG  195.49 360.00

        GRTP2{1}                   109.6333502.68   0.00   69.07  129.99 RFun  226.87 405.00
                                                                         LONG  199.06 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL BENTLEY
01:59 PM AutoPIPE Standard 9.4.0.19 RESULT PAGE 25

ASME B31.8 (2010) CODE COMPLIANCE
(Moments in N.m) (Stress in N/mm2)

Point name	Load combination	In-Pl. Moment	Out-Pl. Moment	Shear Stress	Axial Stress	Bending Stress	type	Code Stress	Code Allow.
A03 N-	SIFI= 1.00 SIFO= 1.00								
	GR + Max P{1}	288.71	32068.77	1.04	-1.13	124.43	SUST	220.66	405.00
	TR:Amb to T2{1}	0.00	2362.11	0.00	0.02	9.16	DISP	9.19	217.41
	Amb to T1{1}	0.00	0.00	0.00	0.00	0.00	DISP	0.00	217.41
	Amb to T2{1}	0.00	2362.11	0.00	0.02	9.16	DISP	9.19	217.41
	Max P{1}						HOOP	158.94	324.00
	GRTP1{1}	288.71	32068.77	0.00	71.20	124.43	RFun	221.65	405.00
							LONG	195.63	360.00
	GRTP2{1}	288.71	29706.66	0.00	71.18	115.26	RFun	215.63	405.00
							LONG	186.45	405.00
A03 N+	SIFI= 1.00 SIFO= 1.00								
	GR + Max P{1}	32068.77	288.71	1.04	-1.13	124.43	SUST	220.66	405.00
	TR:Amb to T2{1}	2362.11	0.00	0.00	0.02	9.16	DISP	9.19	217.41
	Amb to T1{1}	0.00	0.00	0.00	0.00	0.00	DISP	0.00	217.41
	Amb to T2{1}	2362.11	0.00	0.00	0.02	9.16	DISP	9.19	217.41
	Max P{1}						HOOP	158.94	324.00
	GRTP1{1}	32068.77	288.71	0.00	71.20	124.43	RFun	221.65	405.00
							LONG	195.63	360.00
	GRTP2{1}	29706.66	288.71	0.00	71.18	115.26	RFun	215.63	405.00
							LONG	186.45	405.00
A03 N-	SIFI= 1.00 SIFO= 1.00								
	GR + Max P{1}	288.71	32068.77	1.04	-1.13	124.43	SUST	220.66	405.00
	TR:Amb to T2{1}	0.00	2362.11	0.00	0.02	9.16	DISP	9.19	217.41
	Amb to T1{1}	0.00	0.00	0.00	0.00	0.00	DISP	0.00	217.41
	Amb to T2{1}	0.00	2362.11	0.00	0.02	9.16	DISP	9.19	217.41
	Max P{1}						HOOP	158.94	324.00
	GRTP1{1}	288.71	32068.77	0.00	71.20	124.43	RFun	221.65	405.00
							LONG	195.63	360.00
	GRTP2{1}	288.71	29706.66	0.00	71.18	115.26	RFun	215.63	405.00
							LONG	186.45	405.00

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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                    BENTLEY
01:59 PM                                                                  AutoPIPE Standard 9.4.0.19 RESULT PAGE  26
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )           (Stress in N/mm2   )
 Point     Load                          In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                      Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A03 N+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}           32068.77  288.71   1.04   -1.13  124.43 SUST  220.66 405.00

        TR:Amb to T2{1}          2362.11    0.00   0.00    0.02    9.16 DISP    9.19 217.41

        Amb to T1{1}                0.00    0.00   0.00    0.00    0.00 DISP    0.00 217.41

        Amb to T2{1}             2362.11    0.00   0.00    0.02    9.16 DISP    9.19 217.41

        Max P{1}                                                        HOOP  158.94 324.00

        GRTP1{1}                32068.77  288.71   0.00   71.20  124.43 RFun  221.65 405.00
                                                                        LONG  195.63 360.00

        GRTP2{1}                29706.66  288.71   0.00   71.18  115.26 RFun  215.63 405.00
                                                                        LONG  186.45 405.00

 A01 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}           30744.46 1373.04   6.07   -2.13  119.40 SUST  217.23 405.00

        TR:Amb to T2{1}          4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Amb to T1{1}                0.00    0.00   0.00    0.00    0.00 DISP    0.00 220.84

        Amb to T2{1}             4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Max P{1}                                                        HOOP  158.94 324.00

        GRTP1{1}                30744.46 1373.04   0.00   72.20  119.40 RFun  217.67 405.00
                                                                        LONG  191.60 360.00

        GRTP2{1}                26402.69 1373.04   0.00   72.04  102.58 RFun  207.15 405.00
                                                                        LONG  174.61 405.00

 A01 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}           30744.46 1373.04   6.07   -2.13  119.40 SUST  217.23 405.00

        TR:Amb to T2{1}          4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Amb to T1{1}                0.00    0.00   0.00    0.00    0.00 DISP    0.00 220.84

        Amb to T2{1}             4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Max P{1}                                                        HOOP  158.94 324.00

        GRTP1{1}                30744.46 1373.04   0.00   72.20  119.40 RFun  217.67 405.00
                                                                        LONG  191.60 360.00

        GRTP2{1}                26402.69 1373.04   0.00   72.04  102.58 RFun  207.15 405.00
                                                                        LONG  174.61 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                BENTLEY
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )            (Stress in N/mm2   )
 Point     Load                        In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                    Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------     ------- ------- ------  ------  ------ ----  ------ ------

 A01 F- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             30744.46 1373.04   6.07   -2.13  119.40 SUST  217.23 405.00

        TR:Amb to T2{1}            4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 220.84

        Amb to T2{1}               4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                  30744.46 1373.04   0.00   72.20  119.40 RFun  217.67 405.00
                                                                          LONG  191.60 360.00

        GRTP2{1}                  26402.69 1373.04   0.00   72.04  102.58 RFun  207.15 405.00
                                                                          LONG  174.61 405.00

 A01 F+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             30744.46 1373.04   6.07   -2.13  119.40 SUST  217.23 405.00

        TR:Amb to T2{1}            4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 220.84

        Amb to T2{1}               4341.76    0.00   0.00    0.16   16.85 DISP   17.01 220.84

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                  30744.46 1373.04   0.00   72.20  119.40 RFun  217.67 405.00
                                                                          LONG  191.60 360.00

        GRTP2{1}                  26402.69 1373.04   0.00   72.04  102.58 RFun  207.15 405.00
                                                                          LONG  174.61 405.00

 A04 N- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             22287.91 4390.52   0.93   -2.13   88.14 SUST  185.37 405.00

        TR:Amb to T2{1}            1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 252.71

        Amb to T2{1}               1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Max P{1}                                                          HOOP  158.94 324.00

        GRTP1{1}                  22287.91 4390.52   0.00   72.20   88.14 RFun  198.64 405.00
                                                                          LONG  160.33 360.00

        GRTP2{1}                  23676.94 4390.52   0.00   72.04   93.43 RFun  201.74 405.00
                                                                          LONG  165.47 405.00
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6INCH SPOOL
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )             (Stress in N/mm2   )
 Point     Load                         In-Pl. Out-Pl.  Shear  Axial  Bending         Code   Code
 name   combination                     Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A04 N+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            22287.91 4390.52   0.93   -2.13   88.14 SUST  185.37 405.00

        TR:Amb to T2{1}           1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 252.71

        Amb to T2{1}              1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 22287.91 4390.52   0.00   72.20   88.14 RFun  198.64 405.00
                                                                         LONG  160.33 360.00

        GRTP2{1}                 23676.94 4390.52   0.00   72.04   93.43 RFun  201.74 405.00
                                                                         LONG  165.47 405.00

 A04 N- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            22287.91 4390.52   0.93   -2.13   88.14 SUST  185.37 405.00

        TR:Amb to T2{1}           1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 252.71

        Amb to T2{1}              1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 22287.91 4390.52   0.00   72.20   88.14 RFun  198.64 405.00
                                                                         LONG  160.33 360.00

        GRTP2{1}                 23676.94 4390.52   0.00   72.04   93.43 RFun  201.74 405.00
                                                                         LONG  165.47 405.00

 A04 N+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            22287.91 4390.52   0.93   -2.13   88.14 SUST  185.37 405.00

        TR:Amb to T2{1}           1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 252.71

        Amb to T2{1}              1389.03    0.00   0.00    0.16    5.39 DISP    5.55 252.71

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 22287.91 4390.52   0.00   72.20   88.14 RFun  198.64 405.00
                                                                         LONG  160.33 360.00

        GRTP2{1}                 23676.94 4390.52   0.00   72.04   93.43 RFun  201.74 405.00
                                                                         LONG  165.47 405.00
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6INCH SPOOL
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ASME B31.8 (2010) CODE COMPLIANCE
(Moments in N.m) (Stress in N/mm2)

Point name	Load combination	In-Pl. Moment	Out-Pl. Moment	Shear Stress	Axial Stress	Bending Stress	type	Code Stress	Code Allow.
A06	SIFI= 1.00 SIFO= 1.00								
	GR + Max P{1}	18995.27	2018.79	0.93	-2.13	74.11	SUST	171.35	405.00
	TR:Amb to T2{1}	718.21	0.00	0.00	0.16	2.79	DISP	2.95	266.73
	Amb to T1{1}	0.00	0.00	0.00	0.00	0.00	DISP	0.00	266.73
	Amb to T2{1}	718.21	0.00	0.00	0.16	2.79	DISP	2.95	266.73
	Max P{1}						HOOP	158.94	324.00
	GRTP1{1}	18995.27	2018.79	0.00	72.20	74.11	RFun	191.15	405.00
							LONG	146.31	360.00
	GRTP2{1}	19713.47	2018.79	0.00	72.04	76.89	RFun	192.66	405.00
							LONG	148.92	405.00
A06	SIFI= 1.00 SIFO= 1.00								
	GR + Max P{1}	18995.27	2018.79	0.93	-2.13	74.11	SUST	171.35	405.00
	TR:Amb to T2{1}	718.21	0.00	0.00	0.16	2.79	DISP	2.95	266.73
	Amb to T1{1}	0.00	0.00	0.00	0.00	0.00	DISP	0.00	266.73
	Amb to T2{1}	718.21	0.00	0.00	0.16	2.79	DISP	2.95	266.73
	Max P{1}						HOOP	158.94	324.00
	GRTP1{1}	18995.27	2018.79	0.00	72.20	74.11	RFun	191.15	405.00
							LONG	146.31	360.00
	GRTP2{1}	19713.47	2018.79	0.00	72.04	76.89	RFun	192.66	405.00
							LONG	148.92	405.00
A01 N-	SIFI= 1.00 SIFO= 1.00								
	GR + Max P{1}	2750.93	17645.86	3.39	1.83	69.29	SUST	162.89	405.00
	TR:Amb to T2{1}	0.00	3730.77	0.00	0.02	14.48	DISP	14.50	275.18
	Amb to T1{1}	0.00	0.00	0.00	0.00	0.00	DISP	0.00	275.18
	Amb to T2{1}	0.00	3730.77	0.00	0.02	14.48	DISP	14.50	275.18
	Max P{1}						HOOP	158.94	324.00
	GRTP1{1}	2750.93	17645.86	0.00	68.25	69.29	RFun	190.71	405.00
							LONG	137.54	360.00
	GRTP2{1}	2750.93	13915.08	0.00	68.28	55.03	RFun	183.92	405.00
							LONG	123.31	405.00

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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                   BENTLEY
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )           (Stress in N/mm2   )
 Point     Load                       In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                   Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A01 N+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            17645.86 2750.93   3.39    1.83   69.29 SUST  162.89 405.00

        TR:Amb to T2{1}           3730.77    0.00   0.00    0.02   14.48 DISP   14.50 275.18

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 275.18

        Amb to T2{1}              3730.77    0.00   0.00    0.02   14.48 DISP   14.50 275.18

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 17645.86 2750.93   0.00   68.25   69.29 RFun  190.71 405.00
                                                                         LONG  137.54 360.00

        GRTP2{1}                 13915.08 2750.93   0.00   68.28   55.03 RFun  183.92 405.00
                                                                         LONG  123.31 405.00

 A01 N- SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             2750.9317645.86   3.39    1.83   69.29 SUST  162.89 405.00

        TR:Amb to T2{1}              0.00 3730.77   0.00    0.02   14.48 DISP   14.50 275.18

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 275.18

        Amb to T2{1}                 0.00 3730.77   0.00    0.02   14.48 DISP   14.50 275.18

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                  2750.9317645.86   0.00   68.25   69.29 RFun  190.71 405.00
                                                                         LONG  137.54 360.00

        GRTP2{1}                  2750.9313915.08   0.00   68.28   55.03 RFun  183.92 405.00
                                                                         LONG  123.31 405.00

 A01 N+ SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}            17645.86 2750.93   3.39    1.83   69.29 SUST  162.89 405.00

        TR:Amb to T2{1}           3730.77    0.00   0.00    0.02   14.48 DISP   14.50 275.18

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 275.18

        Amb to T2{1}              3730.77    0.00   0.00    0.02   14.48 DISP   14.50 275.18

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                 17645.86 2750.93   0.00   68.25   69.29 RFun  190.71 405.00
                                                                         LONG  137.54 360.00

        GRTP2{1}                 13915.08 2750.93   0.00   68.28   55.03 RFun  183.92 405.00
                                                                         LONG  123.31 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                   BENTLEY
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                             ASME B31.8 (2010)  CODE COMPLIANCE
                       (Moments in N.m    )            (Stress in N/mm2   )
Point     Load                      In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
name   combination                  Moment  Moment Stress  Stress  Stress type  Stress Allow.
------ ----------------------     ------- ------- ------  ------  ------ ----  ------ ------

A08    SIFI= 1.00   SIFO= 1.00
       GR + Max P{1}              9589.44  335.90   0.93   -2.13   37.23 SUST  134.49 405.00

       TR:Amb to T2{1}             372.33    0.00   0.00    0.16    1.44 DISP    1.60 303.59

       Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 303.59

       Amb to T2{1}                372.33    0.00   0.00    0.16    1.44 DISP    1.60 303.59

       Max P{1}                                                          HOOP  158.94 324.00

       GRTP1{1}                   9589.44  335.90   0.00   72.20   37.23 RFun  175.34 405.00
                                                                         LONG  109.42 360.00

       GRTP2{1}                   9217.11  335.90   0.00   72.04   35.79 RFun  174.90 405.00
                                                                         LONG  107.82 405.00

A08    SIFI= 1.00   SIFO= 1.00
       GR + Max P{1}              9589.44  335.90   0.93   -2.13   37.23 SUST  134.49 405.00

       TR:Amb to T2{1}             372.33    0.00   0.00    0.16    1.44 DISP    1.60 303.59

       Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 303.59

       Amb to T2{1}                372.33    0.00   0.00    0.16    1.44 DISP    1.60 303.59

       Max P{1}                                                          HOOP  158.94 324.00

       GRTP1{1}                   9589.44  335.90   0.00   72.20   37.23 RFun  175.34 405.00
                                                                         LONG  109.42 360.00

       GRTP2{1}                   9217.11  335.90   0.00   72.04   35.79 RFun  174.90 405.00
                                                                         LONG  107.82 405.00

A07    SIFI= 1.00   SIFO= 1.00
       GR + Max P{1}              2746.19 2309.23   0.93   -2.13   13.92 SUST  111.25 405.00

       TR:Amb to T2{1}            1808.74    0.00   0.00    0.16    7.02 DISP    7.18 326.82

       Amb to T1{1}                  0.00    0.00   0.00    0.00    0.00 DISP    0.00 326.82

       Amb to T2{1}               1808.74    0.00   0.00    0.16    7.02 DISP    7.18 326.82

       Max P{1}                                                          HOOP  158.94 324.00

       GRTP1{1}                   2746.19 2309.23   0.00   72.20   13.92 RFun  168.77 405.00
                                                                         LONG   86.12 360.00

       GRTP2{1}                   4554.93 2309.23   0.00   72.04   19.81 RFun  170.19 405.00
                                                                         LONG   91.85 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                 BENTLEY
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                              ASME B31.8 (2010)  CODE COMPLIANCE
                        (Moments in N.m    )            (Stress in N/mm2   )
 Point     Load                      In-Pl. Out-Pl.  Shear  Axial  Bending        Code   Code
 name   combination                  Moment  Moment Stress  Stress  Stress type  Stress Allow.
 ------ ----------------------    ------- ------- ------  ------  ------ ----  ------ ------

 A07    SIFI= 1.00   SIFO= 1.00
        GR + Max P{1}             2746.19 2309.23   0.93   -2.13   13.92 SUST  111.25 405.00

        TR:Amb to T2{1}           1808.74    0.00   0.00    0.16    7.02 DISP    7.18 326.82

        Amb to T1{1}                 0.00    0.00   0.00    0.00    0.00 DISP    0.00 326.82

        Amb to T2{1}              1808.74    0.00   0.00    0.16    7.02 DISP    7.18 326.82

        Max P{1}                                                         HOOP  158.94 324.00

        GRTP1{1}                  2746.19 2309.23   0.00   72.20   13.92 RFun  168.77 405.00
                                                                         LONG   86.12 360.00

        GRTP2{1}                  4554.93 2309.23   0.00   72.04   19.81 RFun  170.19 405.00
                                                                         LONG   91.85 405.00
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                    BENTLEY
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                            R E S U L T    S U M M A R Y
                            ----------------------------


 Maximum displacements (mm)
 ---------------------------

         Maximum X :    248.00      Point : A05        Load Comb.: Gravity{1}
         Maximum Y :   -297.02      Point : A06        Load Comb.: GRTP2{1}
         Maximum Z :    108.45      Point : A01 F      Load Comb.: Gravity{1}
         Max. total:    369.30      Point : A06        Load Comb.: GRTP2{1}

 Maximum rotations (deg)
 -----------------------

         Maximum X :      1.12      Point : A03 N      Load Comb.: Gravity{1}
         Maximum Y :     -0.23      Point : A08        Load Comb.: Gravity{1}
         Maximum Z :     -1.81      Point : A01 N      Load Comb.: Gravity{1}
         Max. total:      2.02      Point : A01 N      Load Comb.: Gravity{1}



 Maximum restraint forces (N )
 ------------------------------

         Maximum X :     15645      Point : A00        Load Comb.: Gravity{1}
         Maximum Y :    -17237      Point : A00        Load Comb.: Gravity{1}
         Maximum Z :     -1149      Point : A00        Load Comb.: Gravity{1}
         Max. total:     23307      Point : A00        Load Comb.: Gravity{1}

 Maximum restraint moments (N.m  )
 ----------------------------------

         Maximum X :     -4018      Point : A00        Load Comb.: Gravity{1}
         Maximum Y :     -4756      Point : A05        Load Comb.: Gravity{1}
         Maximum Z :    -88455      Point : A00        Load Comb.: Gravity{1}
         Max. total:     88564      Point : A00        Load Comb.: Gravity{1}
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6INCH SPOOL
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                            R E S U L T    S U M M A R Y
                            ----------------------------



 Maximum pipe forces (N )
 -------------------------

          Maximum X :    -15645      Point : A00        Load Comb.: Gravity{1}
          Maximum Y :     17237      Point : A00        Load Comb.: Gravity{1}
          Maximum Z :      1149      Point : A00        Load Comb.: Gravity{1}
          Max. total:     23307      Point : A00        Load Comb.: Gravity{1}

 Maximum pipe moments (N.m  )
 -----------------------------

          Maximum X :      4018      Point : A00        Load Comb.: Gravity{1}
          Maximum Y :     -4756      Point : A04 F      Load Comb.: Gravity{1}
          Maximum Z :     88455      Point : A00        Load Comb.: Gravity{1}
          Max. total:     88564      Point : A00        Load Comb.: Gravity{1}
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6INCH SPOOL
05/21/2015 6INCH SPOOL                                                     BENTLEY
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                            R E S U L T   S U M M A R Y
                            ---------------------------



     Maximum sustained stress

                              Point            : A00
                              Stress    N/mm2  : 434.75
                              Allowable N/mm2  : 405.00
                              Ratio            : 1.07
                              Load combination : GR + Max P{1}

     Maximum displacement stress

                              Point            : A02 N
                              Stress    N/mm2  : 18.31
                              Allowable N/mm2  : 120.90
                              Ratio            : 0.15
                              Load combination : Max Range

     Maximum hoop stress

                              Point            : A00
                              Stress    N/mm2  : 158.94
                              Allowable N/mm2  : 324.00
                              Ratio            : 0.49
                              Load combination : Max P{1}

     Maximum Longitudinal stress

                              Point            : A00
                              Stress    N/mm2  : 409.67
                              Allowable N/mm2  : 360.00
                              Ratio            : 1.14
                              Load combination : GRTP1{1}

     Maximum Combined stress

                              Point            : A00
                              Stress    N/mm2  : 404.36
                              Allowable N/mm2  : 405.00
                              Ratio            : 1.00
                              Load combination : GRTP1{1}
```

```
----------------------------------------------------------------------------------------------------------------
6INCH SPOOL
05/21/2015 6INCH SPOOL                                                   BENTLEY
01:59 PM                                                                 AutoPIPE Standard 9.4.0.19 RESULT PAGE  36
----------------------------------------------------------------------------------------------------------------


                            R E S U L T   S U M M A R Y
                            ---------------------------



     Maximum sustained stress ratio

                              Point            : A00
                              Stress    N/mm2  : 434.75
                              Allowable N/mm2  : 405.00
                              Ratio            : 1.07
                              Load combination : GR + Max P{1}

     Maximum displacement stress ratio

                              Point            : A00
                              Stress    N/mm2  : 16.58
                              Allowable N/mm2  : 3.33
                              Ratio            : 4.98
                              Load combination : Max Range

     Maximum hoop stress ratio

                              Point            : A00
                              Stress    N/mm2  : 158.94
                              Allowable N/mm2  : 324.00
                              Ratio            : 0.49
                              Load combination : Max P{1}

     Maximum Longitudinal stress ratio

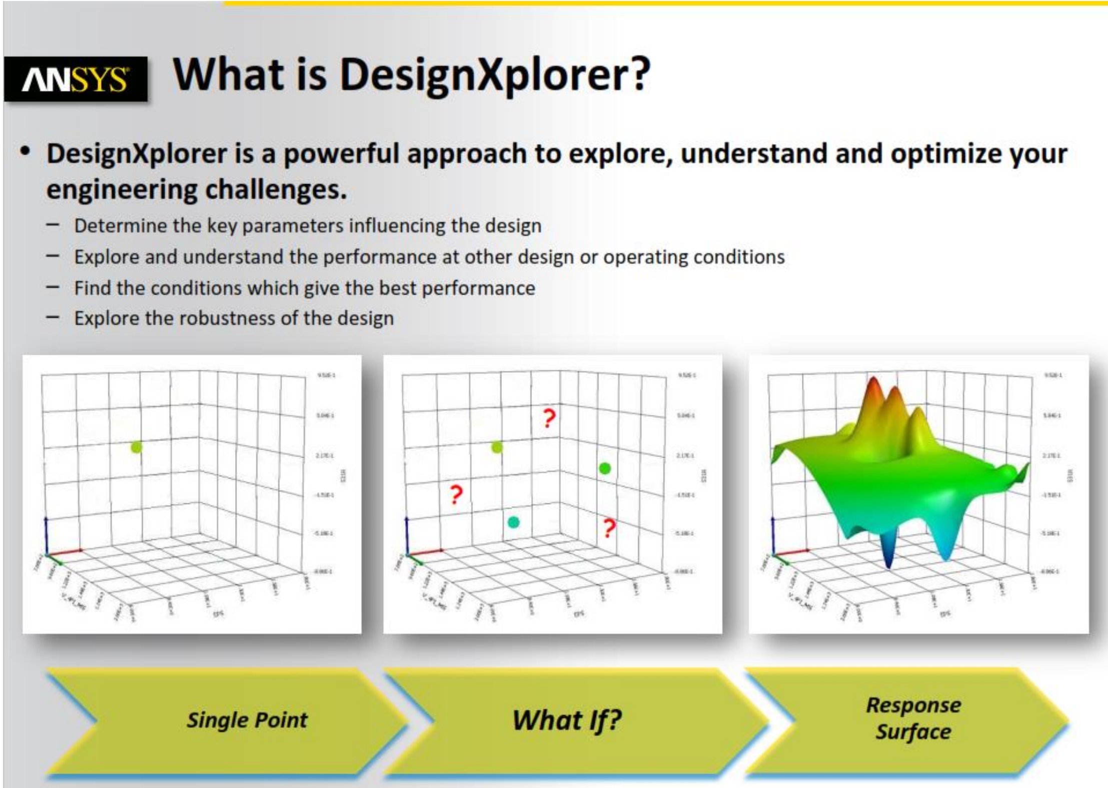
                              Point            : A00
                              Stress    N/mm2  : 409.67
                              Allowable N/mm2  : 360.00
                              Ratio            : 1.14
                              Load combination : GRTP1{1}

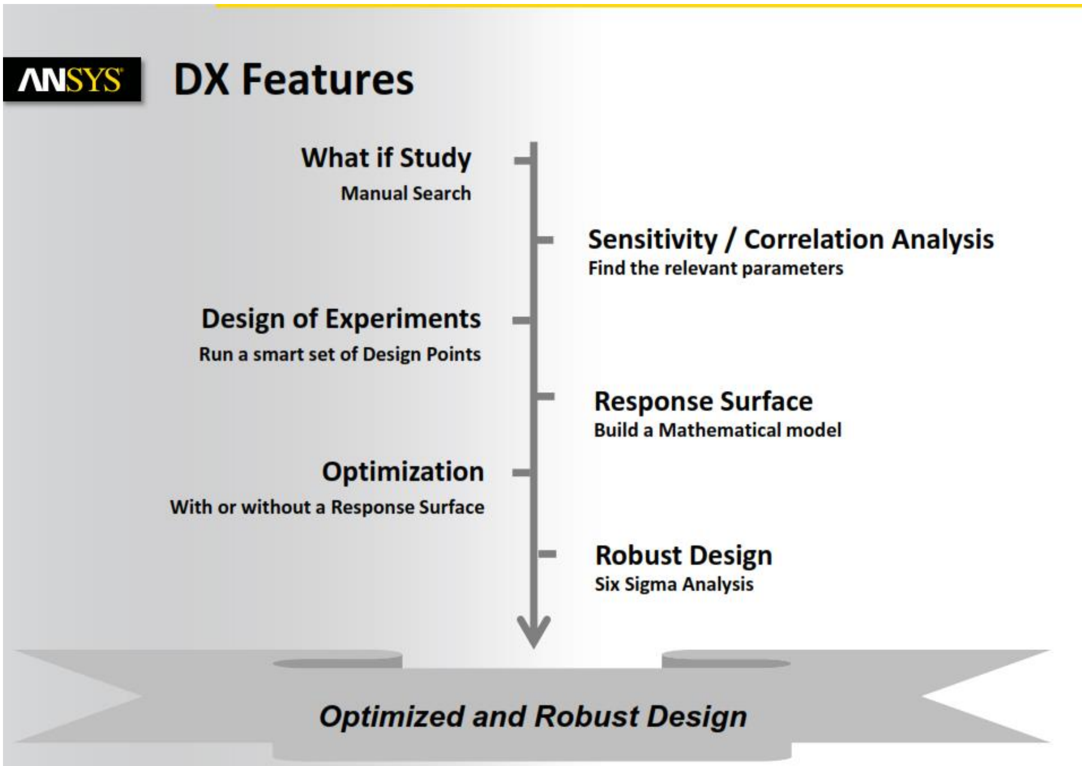
     Maximum Combined stress ratio

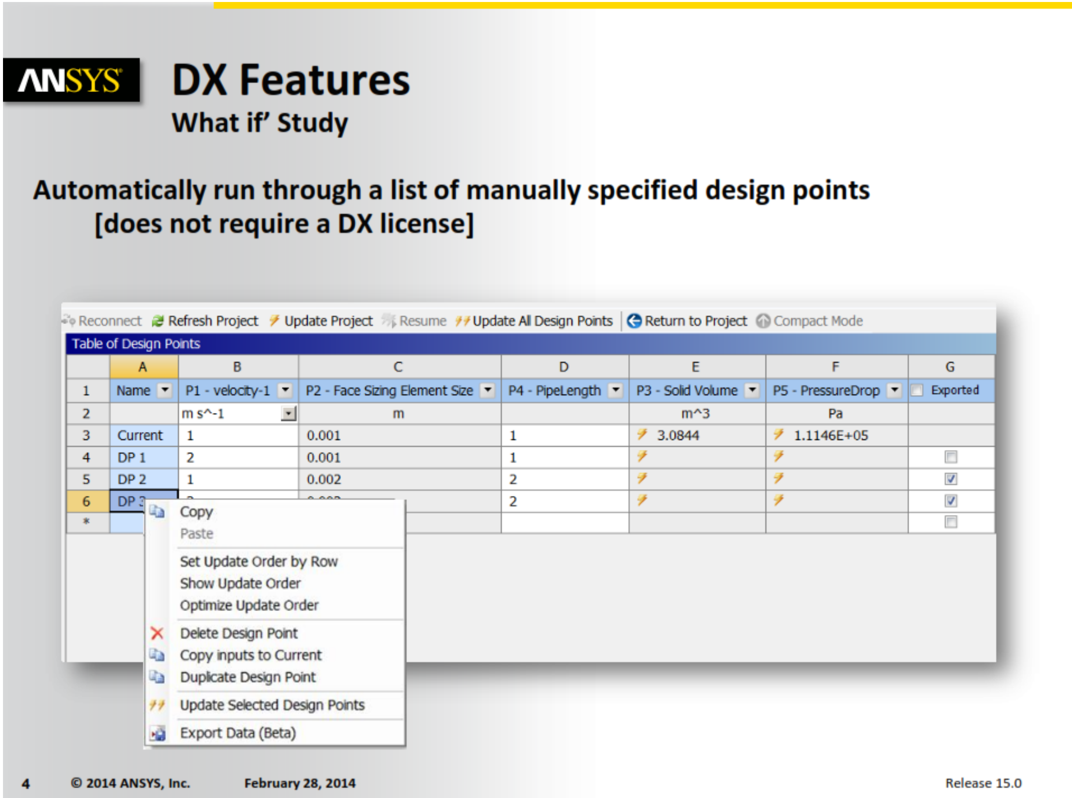
                              Point            : A00
                              Stress    N/mm2  : 404.36
                              Allowable N/mm2  : 405.00
                              Ratio            : 1.00
                              Load combination : GRTP1{1}

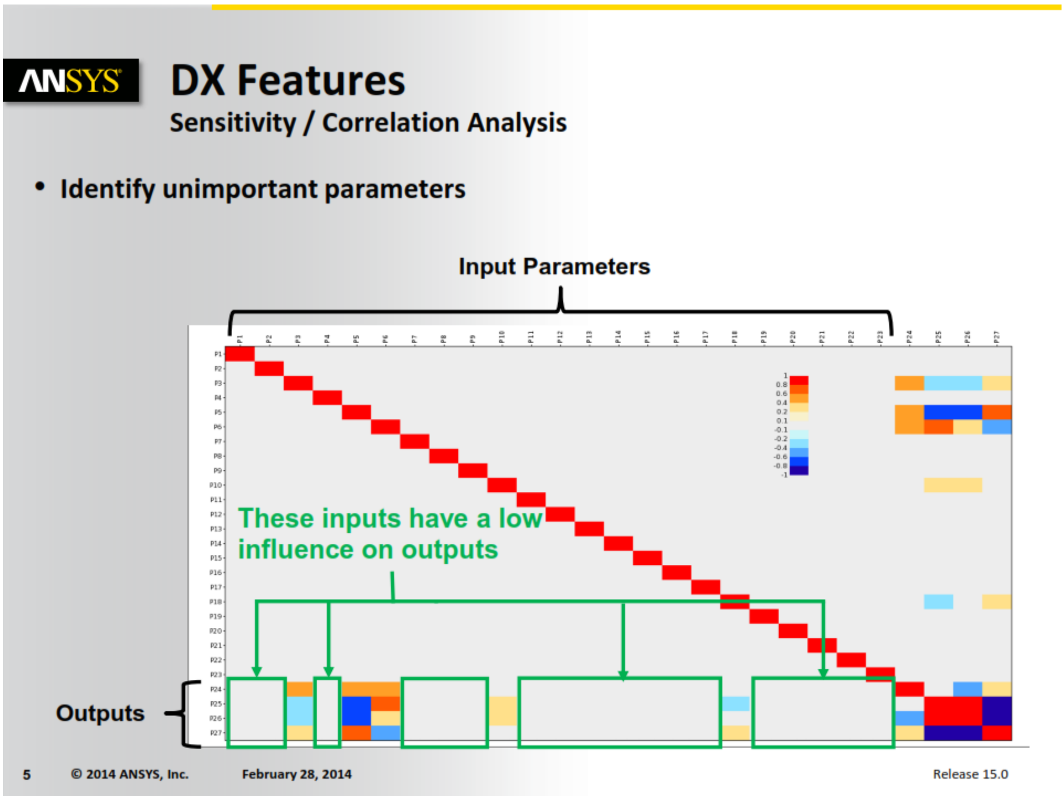

     * * * The system does not satisfy ASME B31.8 (2010) code requirements * * *
     * * * for the selected options                                        * * *
```

Appendix 5 Ansys Design Explorer Features


ANSYS
What is DesignXplorer?
• DesignXplorer is a powerful approach to explore, understand and optimize your engineering challenges.
– Determine the key parameters influencing the design
– Explore and understand the performance at other design or operating conditions
– Find the conditions which give the best performance
– Explore the robustness of the design
?
?
?
Single Point
What If?
Response Surface


ANSYS
DX Features
What if Study
Manual Search
Sensitivity / Correlation Analysis
Find the relevant parameters
Design of Experiments
Run a smart set of Design Points
Response Surface
Build a Mathematical model
Optimization
With or without a Response Surface
Robust Design
Six Sigma Analysis
Optimized and Robust Design

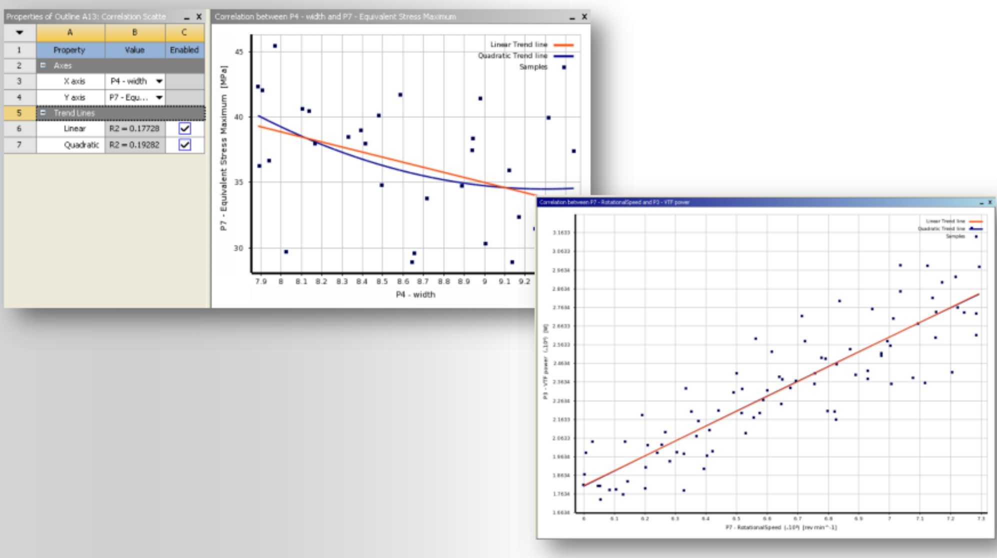

DX Features
What if' Study
Automatically run through a list of manually specified design points [does not require a DX license]
Reconnect Refresh Project Update Project Resume Update All Design Points Return to Project Compact Mode
Table of Design Points
A B C D E F G
1 Name P1 - velocity-1 P2 - Face Sizing Element Size P4 - PipeLength P3 - Solid Volume P5 - PressureDrop Exported
2 m s^-1 m m^3 Pa
3 Current 1 0.001 1 3.0844 1.1146E+05
4 DP 1 2 0.001 1
5 DP 2 1 0.002 2
6 DP 3 2
Copy
Paste
Set Update Order by Row
Show Update Order
Optimize Update Order
Delete Design Point
Copy inputs to Current
Duplicate Design Point
Update Selected Design Points
Export Data (Beta)
4 © 2014 ANSYS, Inc. February 28, 2014 Release 15.0


DX Features
Sensitivity / Correlation Analysis
• Identify unimportant parameters
Input Parameters
These inputs have a low influence on outputs
Outputs
5 © 2014 ANSYS, Inc. February 28, 2014 Release 15.0

DX Features

Sensitivity / Correlation Analysis

- **Identify the degree to which the relationship is linear/quadratic**

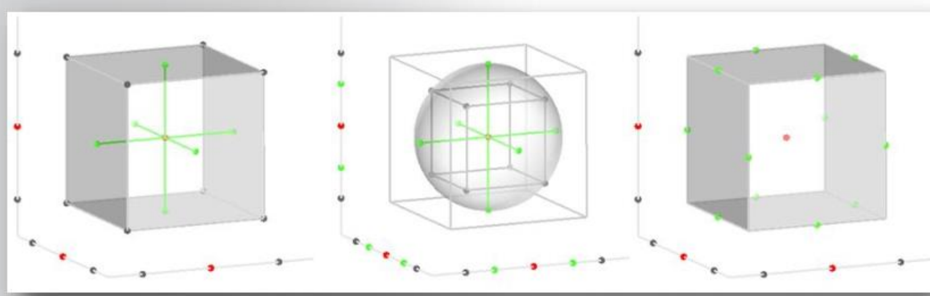


  February 28, 2014 Release 15.0

DX Features

Design of Experiments

- **Specify the DOE Type**
- **Specify each parameter range and type (Continuous, Discrete, Manufacturable Values)**
- **Design Points are automatically chosen to explore the parametric space efficiently**

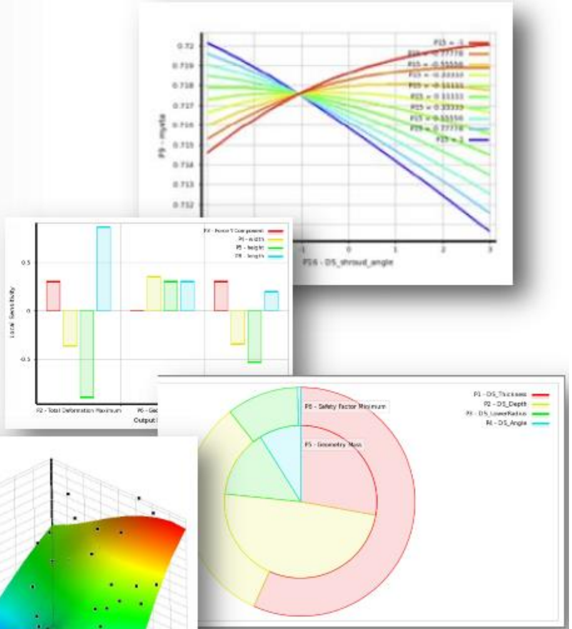


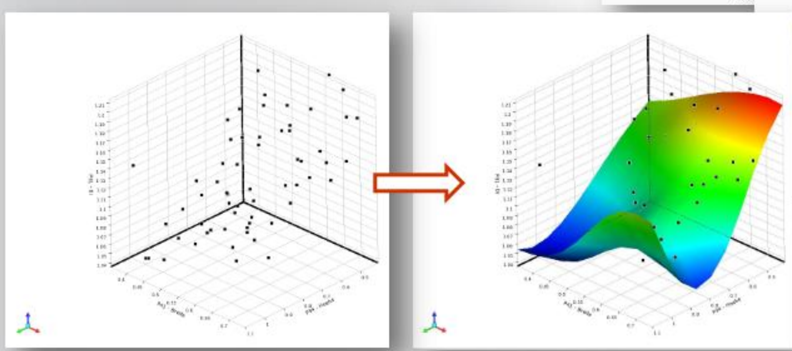
  February 28, 2014 Release 15.0

DX Features

Response Surface

- Generate a Surrogate Model
- Min/Max Search
- 2D/3D plots
- Local Sensitivities
- RS Quality Assessment

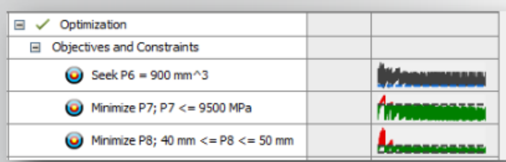




DX Features

Optimization

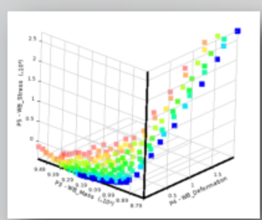
- Define Objectives, Constraints and Input Parameter Relationships
- Based on a Response surface
  - Explore thousands of configurations in a few seconds

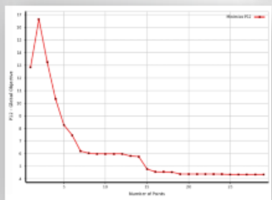


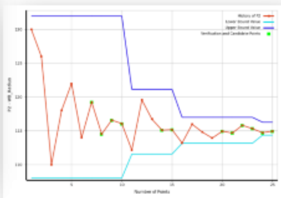
Find the Best Design

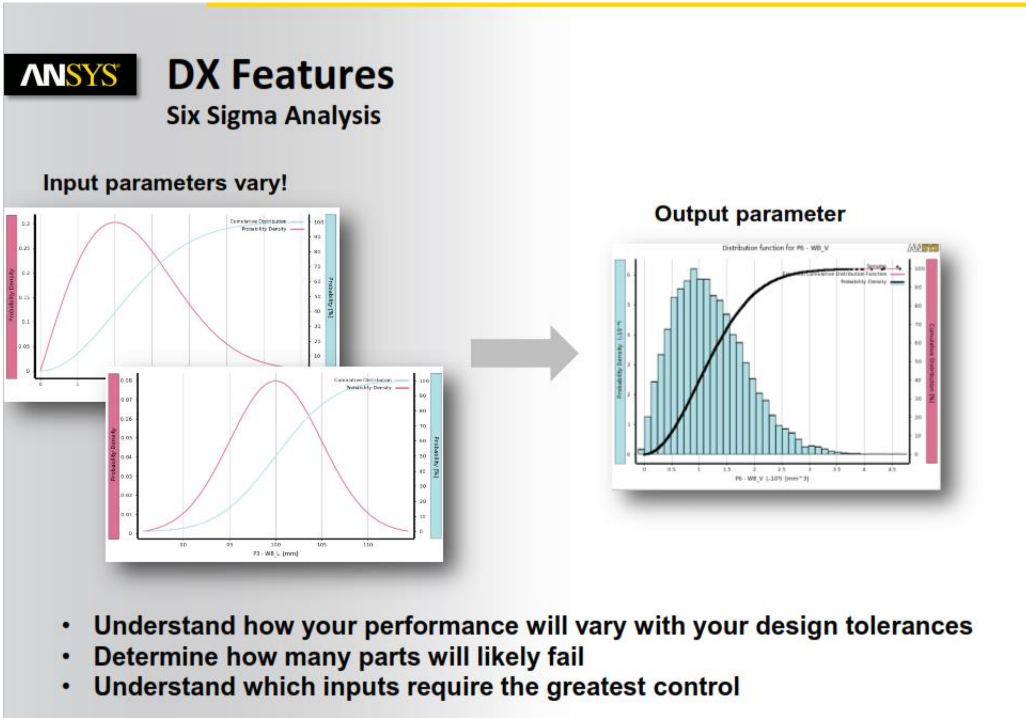
Reference	Name	P8 - WB_DIS (mm) Parameter Value	Variation from Reference
◉	---- Initial Design ----	79.999	0.00 %
○	---- Initial Design ---- (verified) (DP 0)	80	0.00 %
○	Candidate Point 1	41.449	-48.19 %
○	Candidate Point 1 (verified)	41.632	-47.96 %
○	With Rounded Values	41.186	-48.52 %
○	With Rounded Values (verified)	41.373	-48.28 %
	New Custom Candidate Point		

- Based on Direct Solves
  - Follow algorithm convergence








ANSYS
DX Features
Six Sigma Analysis
Input parameters vary!
Output parameter
• Understand how your performance will vary with your design tolerances
• Determine how many parts will likely fail
• Understand which inputs require the greatest control

Appendix 6 Spool Type Comparison

Table A6-1 Spool type comparison

Analysis Criteria	Vertical spool	Horizontal spool
Ease of Installation	Vertical connectors are gravity deployed directly over the hub, connection tooling is simpler, lighter & cheaper, allows short hub-to-hub spacing, vertical connections can allow compact receiver assemblies, large rotation of hubs is not allowed.	Horizontal spools are gravity positioned and then controlled “make-up” is achieved by tooling, docking of hub on porch is vertical but stroking is horizontal, stroking generates deflection, so short jumpers difficult, thermal expansion can be accommodated via known pull-in loads and relative hub twist can be accommodated
Controls Multibore and umbilical installation (e.g. tree jumpers)	Difficult / industry very little experience	Simple / lots industry experience – Horizontal connectors can accommodate multibore designs easier than vertical connectors due to better-controlled alignment systems. The vertical connector does not generally have a specific orientation whereas the receptacle and keys on the connector orientate the horizontal systems. If a multibore system is utilised there would be advantages in using it throughout on main connections and process connections alike. There has been very little utilisation of Multibore on vertical systems whereas multibore Horizontals have been used on the block 18, Girassol and Dalia projects amongst others.
Landing and locking loads.	The vertical connector running tool has to control the landing loads and ensure they are not transferred into the locking function.	The horizontal system has distinctly different landing and locking operations thus giving a high level of control over these functions
Seal replacement	Vertical connector has to be completely removed in order to replace a seal. With the vertical connector greater care is required to ensure the connector is not separated too far. *Most of the vertical jumpers can accommodate flexible end stroking by up to 0.5m in vertical direction and access to seal can be achieved dependent upon connector type and flexibility of spool. Hence the requirement for total re-movement of spool is not necessarily required.*	Seals (or seal plate, gasket) may be replaced by stroking back a horizontal connector.
Torsional Load Capacity	Vertical connectors are generally exposed to higher torsional loads as a result of the connector orientation.	The torsional loads on horizontal connectors can be minimized by optimizing the spool geometry
Turning moment	Because the vertical vonnector is fitted to taller structure there is an increase in the turning moment on the structure.	Simple
Forces and moments on hub and connector	*High bending moment on connectors due to jumpers geometry tolerances and make up of connection. Buoyancy elements may be required in order to keep within connector and pipe material capacity.*	*As for vertical spools, bending moments caused by weight and connection makeup limits the capacity of the system.*
Impact on structure design	Vertical connector spool configurations can result in significant loading of seabed structures.	Horizontal connectors require a greater degree of receiver structure and hub support than the more compact arrangement possible with vertical connectors,
Hydrate avoidance	Vertical configured spool hampers free drainage of water	The horizontal connector more easily accommodates the retrofit of insulation in the form of “doghouses”, these are more difficult to effectively design and deploy on vertical

Analysis Criteria	Vertical spool	Horizontal spool
		connectors.
Complexity	Simpler connection on trees and manifold *Free span of vertical jumper may require buoyancy element and VIV strakes in order to avoid Vortex induced vibration and fatigue problems. complexity and cost increases*	Requires more subsea complexity in connection system
Maintenance	No difference *Increased survey may be required in order to monitor vibrations, and system components*	No difference
Flow Assurance	Gas is more likely to collect within the jumper enabling hydrate to form in the jumper if hydrate mitigation procedure fail. Hydrate formation may become an issue at the top of the hairpin 'U' bends in vertical connectors, which is less of an issue with horizontal connection system	Pipework is less likely to collect gas pockets that causes hydrates
Size and Weight	Large, Heavy	Larger, Heavier
Anti-Snagging Capability	Pipe runs vertically out of the connector *Higher risk of snagging*	Pipe runs horizontal out of the structure *Medium risk often protected by GRP covers*
Proven Technology	Yes	Yes
Emergency Disconnection Feature	Yes	Yes
Soft Landing System	Landing and locking loads, the vertical connector running tool has to control the landing loads and ensure they are not transferred into the locking function. System has soft landing system or controlled descent during final alignment of critical components	Landing and locking loads, The horizontal system has distinctly different landing and locking operations thus giving a high level of control over these functions. Soft landings system not required as hubs stroked into contact as separate operation
Tolerance to Hydrodynamics induce Loads	Low	High
Controls Multibore and umbilical installation (e.g. tree jumpers)	Difficult I industry very little experience	Simple I lots industry experience
Controlled connector landing and makeup	Greater risk of seal damage or problems with connector makeup	Lesser risk of seal damage or problems with connector makeup
Decouple Schedule for spool handling and makeup	Difficult	Simple
Retrieval of tree/manifold	Difficult	Simple
Connector stroking distance	Neutral on Spool design	Can be used to advantage or neutralised (U spool)
Pigging	The complexity (andrisk) is increased in the vertical connection system because of the extra 5D bends that have to be fitted to the Pigging loop. The Pig launcher receiver has to have a 90 degree bend fitted so that it does not interfere with the connector installation tooling.	Less risk of pig getting stuck due to horizontal orientation of spool.
Loads on Horizontal vs. Vertical axis connections	Advantage for riser base	Advantage for FTA-manifold- tree
insulation	For the vertical system there is a limit on the thickness of insulation so that the tool can still be placed on and taken off the connector. if additional insulation is required this would make the insulation doghouse large and difficult to install. A further consideration is plane of deploying the insulation doghouse, for verticals it has to be wrapped around the connector whereas for Horizontals it is lowered onto the connector and hence is easier.	Horizontal connectors can incorporate insulation requirements easier than vertical connectors due to the potential clash of the vertical tooling system.
Metrology	Higher requirement Vertical connectors require more accurate metrology in order to accurately install both ends of a flowline spool. This is because the vertical connector is placed directly on the final alignment	Lower requirement

Analysis Criteria	Vertical spool	Horizontal spool
	structure whereas the horizontal connector is lowered into a receptacle that gives both coarse and final alignment as well as allowing an additional tolerance during the connector final make up.	
Multibore design	The vertical connector does not generally have a specific orientation whereas the receptacle and keyson the connector orientate the horizontal systems. if a multibore system is utilised there would be advantages in using it throughout on main connections and process connections alike. There has been very little utilisation of Multibore on vertical systems whereas multibore Horizontals have been used on Greater Plutonio, Girassol and Dalia projects amongst others. *IKM has participated in the ICHTHYS field in the NW of Australia. A multibore vertical connector design with piggyback was used here.*	Horizontal connectors can accommodate multibore designs easier than vertical connectors due to better-controlled alignment systems.
Equipment Retrieval	Difficult The requirement to recover flowline / umbilical jumper in order to retrieve subsea production equipment, such as tree or manifold – horizontal connectors may be disconnected and stroked away from the equipment and left in the receptacle. Vertical connectors require to be lifted away from the equipment and either wet parked or retrieved to surface increasing total vessel time. Horizontal connectors only require one end of a flowline spool to be disconnected in order to retrieve an item of subsea equipment, whereas vertical connectors require both ends to be disconnected. Requirement for additional structure – vertical connectors require a secondary receptacle in order to wet park the flowline spool after retrieval of the subsea equipment. Alternatively a secondary connection system such as a flowbase could be utilised. A horizontal connector does not require any secondary equipment for wet parking but does require some form of structure to accommodate the guidance and/or pull in system.	Simple
Deploy to place system	Not affected by seabed condition	
Buoyancy application	Buoyancy in some cases is required in order to reduce connector and spool stresses where nominal spools are particularly long, loads are particularly high due to structure movements and equipment are installed out with installation tolerances necessitating the design of special jumpers. *Ref. IKM Comments on requirement for buoyancy*	
Structural Requirements	The vertical structure has a slightly smaller footprint than the horizontal connector and this may result in a slightly reduced weight and footprint for structures using a vertical connector but they are slightly taller. Taller Structures however will mean higher moments that are acting to "turn" over the structures. *IKM experience: vertical Jumpers requires large structural test arrangement onshore and supporting equipment when deploying on barge as they need to be in an upraised position with support for lifting equipment hence required deck area increases and more labour for seafastening must be carried out.*	*Requires pecial installation lifting spreader which is costly, the spool requires often more deck space.*
Seabed space requirement	*Requires larger seabed area due to bends and flexibility requirements. Can interfer with other structures and subsea equipment*	*Less seabed area required as this spool is in the vertical position*