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ABSTRACT

Rigid horizontal spools provide the connection between flowlines and subsea structures. A typical subsea development may consist of a number of wells and subsea structures, which each need to be tie-in with the help of subsea spools. The spool design consequently need to be highly reliable as they also serve an important function of accommodating displacements caused by pipeline expansion to avoid damage to the connecting structures, in addition to forming the connection between pipelines and subsea structures such as manifolds and templates. Spools also have to accommodate tolerances for metrology, fabrication and installation. Loads imposed on the spool connecting hubs due to misalignments during tie-in as well as the pipe expansion set the limitations for the spool design. Different spool shapes, provide different levels of flexibility. The main objective of this thesis is to design horizontal subsea spools at a water depth of more than 1000m that is able to accommodate a 1m pipeline expansion whilst complying with the limitations set by the hub capacities. An analysis was carried out for different spool shapes in order to judge their ability to accommodate the imposed loads.

The minimum spool size for the three spool configurations was determined by the use of the finite element program ANSYS 15.0. The limiting design criteria were found to be the hub capacities and the spools were optimized based on this limitation. The wall thickness of the spools complies with the limit states described in DNV-OS-F101. The spools were analysed through a series of six load steps. In the first load step, the spool self-weight was applied, followed by a tie-in sequence in the second load step. An evaluation was made to investigate which combination of metrology and fabrication tolerances were governing. Operating and design conditions were subsequently applied.

The workings of DNV standard for pipeline design for wall thickness design of spools along with other design considerations such as installation, fabrication and operational issues is presented.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

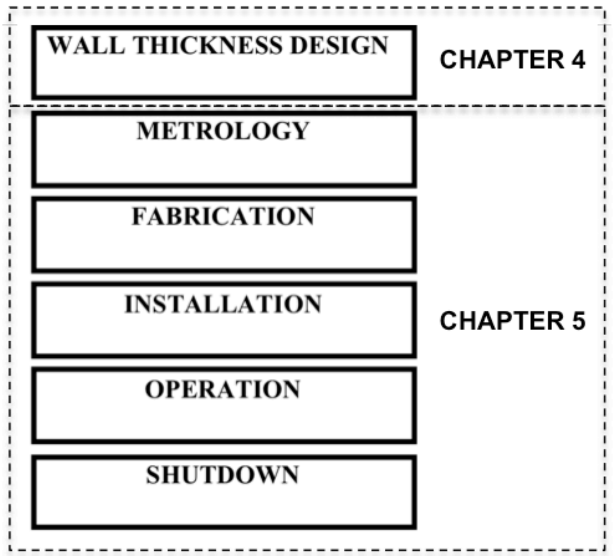
Today the oil and gas industry focuses largely on developing subsea installations for processing and transportation of oil and gas. In the early 1970s the concept of subsea field development was conceptualized by placing wellheads and production equipment on the seabed and in the past 40 years subsea systems have moved from manually operated systems at shallow depths to remote controlled systems at water depths up to 3000m, assisted by complex design and installation developments (Golan & Sangesland, 1992).

Marginal fields or fields at deep water that were previously thought to be either technically unfeasible or uneconomical can now be produced due to developments in the technology of subsea production systems (Yong & Qiang , 2012). This has resulted in a need for subsea connection arrangements such as spool pieces that can be installed and operated without the assistance of divers and that can provide safe and reliable production of hydrocarbons.

Pipelines are the blood vessels of the oil and gas production system. They transport hydrocarbons to and from both offshore and onshore facilities and connect the different subsea components. Pipelines subject to high temperatures from the internal contents will expand and may lead to issues such as pipeline buckling and pipeline walking in addition to the risk of damage to the structures connected to the pipeline that cannot accommodate such displacements. To make up the final connection between the pipelines and the connecting structures such as manifolds and templates, spool pieces are used. Spools are essentially a short piece of pipe, designed to make up for installation misalignments between the connecting structures but also serves a very important function by absorbing pipeline expansion. End expansion has become a significant issue in a number of deepwater developments and reliable tie-ins are vital for a subsea system (Bruton, Carr, Crawford, & Poiate, 2005). A typical deepwater development may consist of several wells, and structures connected by numerous pipelines where each connection needs to be made by the use of tie-in spools.

Spools can be shaped into a large range of geometries and shapes. The flexibility of the spool to be able to accommodate the applied displacement caused by the pipe expansion depends on the length of the individual legs and the overall shape of the spool. The overall size and shape of the spools is therefore an important factor when it comes to deck space for the individual installation vessels and the ease of installation. This thesis aims to design spools and assess different geometries and sizes which are all subject to the same pipeline expansion in order to see if altering the geometry can lead to smaller and more compact spools.

The design life of a spool is normally 25 years and in that time it will experience varying loading conditions both in terms of design loads such as the working pressures and the operational temperatures but also loads imposed by pipeline expansion and hydrodynamic forces. In addition, the designed must also consider loads applied to the spool during processes such as installation, tie-in and shutdown. The life-cycle diagram below shows the different stages in the spool life cycle, which need to be addressed in the design of spools.



1.2 SCOPE AND OBJECTIVES.

The main objective of this thesis was to design horizontal subsea spools at water depths of more than 1000m that is able to accommodate a 1m pipeline expansion whilst complying with the limitations set by the connection hub capacities. An analysis was carried out for different spool shapes in order to judge their ability to accommodate the imposed loads. The analysis is limited to rigid horizontal spools.

The spools are designed and studied with the help of finite element analysis. All relevant steps in the spool life cycle are considered and discussed. An ANSYS code was generated in order to run the analysis with all the required load steps. The relevant load steps included in the analysis of the spools such as external and internal loads as well as the imposed expansion are applied at different load steps. Due to misalignments between the connecting hubs, spools need to be tested with the multiple geometrical configurations associated with the tolerances. The main tasks covered in the thesis are:

1. Presentation of pipelines and the workings of pipeline expansion.
2. The purpose of subsea spools is explained along with descriptions of different types of spools.
3. Present the relevant equations and methodology for spool design according to DNV-OS-F101
4. Discussion on the relevant design considerations that apply for spools and the important design issues related to fabrication and installation.
5. Modeling and FE analysis of three different spool geometries. Tie-in analysis, including fabrication and metrology tolerances were considered as well as operating and design conditions including a 1 meter pipeline expansion. System test and shutdown conditions were also included.
6. Verify that the loads at the spool ends are within the limiting hub capacities and that the assigned wall thickness comply with the criteria set by DNV-OS-F101.

The analysis does not include:

1. Stroking of the spools is not included in the tie-in analysis
2. Hydrodynamic loads
3. Fatigue analysis of the spool
4. Structure settlement

1.3 STRUCTURE OF REPORT

The report structure is as described below:

Chapter 1 provides background information for the thesis topic along with objectives and scope.

Chapter 2 gives relevant theory on pipelines and pipe components as well as relevant equations and discussion on the stresses and strain experiences by the pipe during operation. The concept of pipeline expansion and expansion analysis is presented.

Chapter 3 discusses tie-in spools and its main functions. This includes a presentation of different types of spools.

Chapter 4 presents the principles and workings of wall thickness design for spools based on DNV-OS-F101: Submarine Pipeline Systems

Chapter 5 presents the relevant design considerations based on the different stages in the spool design life. This section describes the workings and importance of accurate subsea metrology, as well as important design considerations during installation and operating stages.

Chapter 6 provides the methodology of the thesis including a description of the workings of finite elements analysis. The design procedure and load steps are presented along with a description of the finite elements used in the analysis. The different spool geometries and layout are also presented.

Chapter 7 gives a presentation of the results and a discussion on the validity of the model as well as a sensitivity analysis. The different spool geometries are discussed and compared.

Chapter 8 concludes on the main findings and gives a summary of the results based on the set objective

CHAPTER 2

PIPELINES AND PIPE EXPANSION

A subsea assembly for oil and gas production can range from complicated arrangements with several wells linked to a template or tied-back to a manifold as shown to single satellite wells linked to a fixed platform (American Petroleum Institute, 2014). Pipelines transport the oil and gas and other production fluids between facilities and forms the backbone of any subsea production system.

The following section gives a description of the different components of a subsea pipeline and explains the concept of pipeline expansion and expansion analysis.

2.1 SUBSEA PIPELINES

Pipelines have a wide range of applications in offshore developments and they vary significantly in size and length depending on their application, location and interface with other facilities (Yong & Qiang, 2005). Subsea pipelines are used for transportation of crude oil and gas from subsea wells and offshore process facilities but also for re-injection of water and gas into the reservoirs. Also, pipelines are operating at greater and greater depths meaning they are exposed to high levels of external pressure, which needs to be addressed in the design. (Mørk, Collberg, Levold, & Bruschi, 1999).

Depending on their applications pipelines is subject to a range of types and levels of loading which give rise to several design issues and challenges. For pipelines operating at a high internal pressure, effects of temperature change are important. High stresses may arise if the pipe is prevented from expanding causing the pipe to break, buckle or bend excessively (Kishawy & Gabbar, 2010). Expansion of the pipeline due to the temperature changes may be severe enough to destroy supports. Is such case, spool pieces are installed at the pipe ends to provide flexibility.

Subsea pipelines operate under challenging conditions and in order for them be able to maintain safe and serviceable, a number of components are in place to protect the pipe steel. A simple pipeline consists of sections of steel tube welded together using arc welding (Palmer & King, 2008). The inner diameter (ID) of the pipe is determined by requirements set by the flow assurance and the wall thickness of the pipe is determined based on the imposed design loads, such as the pressure difference between the inside and the outside of the pipe.

2.2 PIPE MATERIAL

The pipe steel sections are made up of carbon-manganese steel and must have high strength while retaining ductility, facture toughness and weldability. High strength steels are achieved by using low carbon steels which are micro-alloyed to achieve greater resistance against crack growth (Palmer & King, 2008).

Sections of pipe are welded together on a lay barge to form a pipeline. High costs are associated with the fabrication of the pipeline and the steels weldability is crucial. As a rule of thumb, high strengths steels are more difficult to weld and therefore increase the lay barge cost (Yong & Quang , 2014). Higher strength steels are more expensive, but some saving may be made as the yield strength increases and the wall thickness requirement is reduced (Yong & Quang , 2014).

On top of the nominal wall thickness, which is determined based on the design code standards, the wall thickness is normally increased slightly to account for corrosion and fabrication allowances. For sections of a pipeline that is bent, the bend radius also has limitations set by processes such as pigging and must comply with the applied stresses.

2.2 PIPE COMPONENTS

When pipelines are operating in challenging environments or are required to transport hydrocarbons under high temperature and pressure, coatings and insulation may be required for mechanical protection and insulation of the pipe. Coating is applied in order to protect the pipe from corrosive environments as well as providing protection against damage caused by abrasion and general wear and tear (Davis, 2001). Insulation may also be necessary in order to protect the pipe from rapid temperature fall, which may cause hydrate formation. Hydrate formations may eventually lead to blocked pipes and preventing the flow thought the pipelines (Mokhatab, Wilkens, & Leontaritis, 2007)

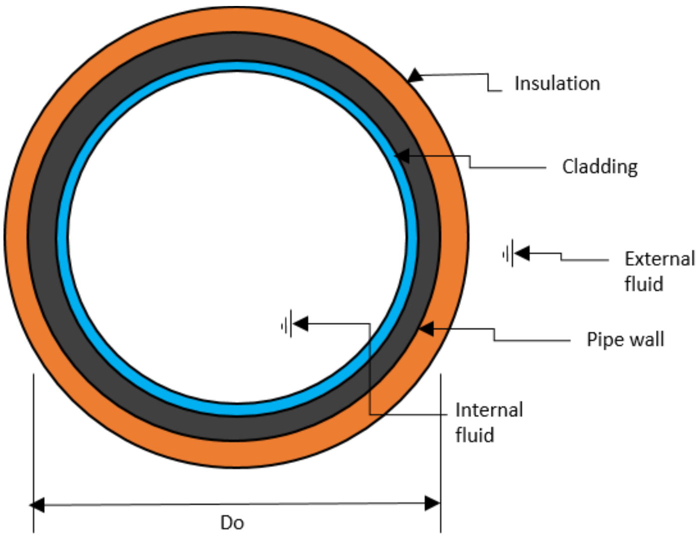


Figure 1: Pipe components

In cases where the pipe is exposed to severe corrosive conditions from the operating fluid, cladding may be used to provide protection. Cladding is a high-cost corrosion resistant alloy that is applied the inner wall of the pipe (Smith, 2012). Cladding is costly, but may provide additional benefits in terms of lowering requirements for processing of the well fluids so that they can be transported over greater distances without the need of drying.

2.3 PIPELINE EXPANSION

Pipelines exposed to constant operating pressure and temperature will expand. As a simplified approach, we can think of a pipeline as a metal bar where the change in length due to change in temperature can be defines as:

$$\Delta L = \alpha\, L_0 \Delta T \qquad (2.1)$$

where α is the coefficient of thermal expansion, L_0 is the original length and ΔT is the temperature change. The coefficient of thermal expansion varies for different materials.

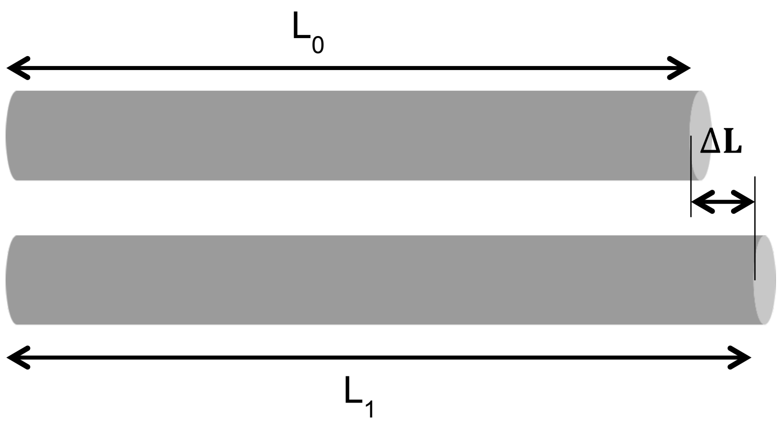


Figure 2: Change in length of pipe due to changes in temperature

In other words, changing temperature and pressure causes the pipeline material to expand and contract axially and radially. If the support conditions prevent the pipeline from expanding, axial stresses build up in the pipeline. The expansion of a pipeline can cause several issues, which need to be addressed in the design. Expansion of a pipeline may cause the pipe to buckle, either laterally if it is exposed to the seabed, or cause upheaval buckling if the pipeline is buried (Yong & Qiang , 2012)

As the pipe expands, friction between the pipe surface and the seabed acts to try and resist the motion. Frictional resistance in the seabed prevents the pipe from expanding freely resulting in a build-up of axial compressive force in the pipeline (Fyrileiv & Collberg, 2005). If this compressive axial force is large enough, the natural tendency is for the pipeline to buckle to relieve the stress (Bruton, White, Cheuk, Bolton, & Carr, 2006). This uncontrolled buckling can have serious consequences for the integrity of the pipeline. Lateral buckling is not a failure mode but the stresses may exceed yield on the first load cycle and involve significant plasticity or it may lead to local buckling (Harrison, Brunner, & Bruton , 2003). In addition, regular shutdowns in normal operation may lead to very high stress cycles, eventually causing fatigue damage to the structure (Bruton, Carr, Crawford, & Poiate, 2005), and there is also a risk that the repeated contraction and expansion may lead to pipeline walking, causing additional stresses on the system (Yong & Quang , 2014).

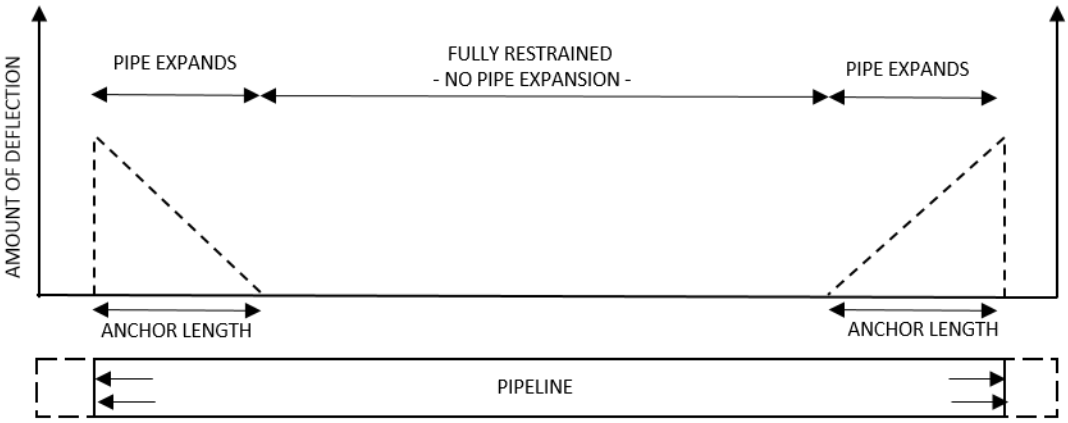


Figure 3: Pipeline end expansion

The response of a pipeline with free ends lied on a flat seabed is not statically determinate but depends on the how the pipe is restrained from expanding (Palmer & King, 2008). During pipeline expansion, the frictional resistance in the seabed prevents the pipe from expanding freely, which results in a build-up of axial compressive force in the pipeline. The concept of a effective axial force, in contrast to the “true” axial force which is given by integrating the stresses over the pipe cross section, is often used to avoid the need for examining effects of external and internal pressures in detail (Fyrileiv & Collberg, 2005). The effective axial force replaces the integration of the pressure field by considering the forces acting on a closed section of the pipe, much like the Archimedes law. The effective axial force influences several structural responses and is considered when designing for lateral- and upheaval buckling, natural frequencies of spans and anchors forces as well as being expansion analysis (Fyrileiv & Collberg, 2005).

The frictional resistance increases from the free ends towards the middle of the pipe. At some point along the pipe, this compressive force equals the expansion force and the pipeline is restrained from further expansion (Fyrileiv & Collberg, 2005). This point is referred to as the virtual anchor point (VAP) (Yong & Quang , 2014)

At the free ends where the pipe where the frictional resistance is not fully developed, the pipe longitudinal strain develops in the pipe and end expansion occurs. Lower friction between the pipe and the soil results in a smaller section of the pipe being restrained from expanding, thus increasing the end expansion. The maximum pipeline expansion is a result of the longitudinal strain caused by the temperature and pressure effects and the frictional resistance caused by the seabed.

2.3.1 PIPELINE STRAINS

Both pressure and temperature effects cause longitudinal stresses and strains in the pipe. For sections where the pipe is restrained from expanding due to the frictional resistance of the seabed, stresses build up in the pipe as mentioned. If the pipe section is free to move longitudinal strains at the ends contributes to the pipeline end expansion. The effects of pressure, temperature and soil-pipe interaction on pipeline expansion are discussed in the following section (Guo, Ghalambor, Lin Ran, & Song, 2014).

Temperature

As the temperature increase during operational conditions, thermal strains and stresses will arise in the pipeline. If the pipeline is unrestrained, it is free to expand and thermal strain will develop in the pipe.

$$\varepsilon_T = \propto \Delta T \tag{2.2}$$

where

$\propto$ is the linear thermal expansion coefficient

ΔT is the change in temperature between installation and operation

PRESSURE

Pressure differences along the pipe give rise to two contributions to the longitudinal stress in a pipeline: stresses due to the Poisson effect and stresses due to the end cap effect.

Poisson's effect

The Poisson's effect causes the pipeline to contract axially as the pipe expands radially. This means that for a pipe subject to internal pressure, the pipe will increase its diameter slightly and at the same time this circumferential expansion will cause the pipeline to contract slightly longitudinally as shown in Figure 4. If the pipe is restrained from contracting axially, longitudinal tensile stress develops (Guo, Ghalambor, Lin Ran, & Song, 2014).

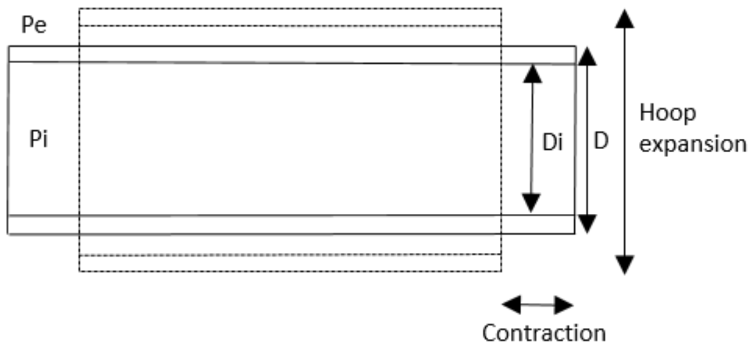


Figure 4: Contraction of pipe due to Poisson's effect

For an unrestrained pipe the strain due to Poisson's effect is:

$$\varepsilon_{Poisson} = -\upsilon\frac{\sigma_{hoop}}{E} \tag{2.3}$$

where

σ_{hoop}	is the hoop stress
υ	is the Poisson's ratio

End Cap effect

Another contribution to the longitudinal stresses in a pipe is caused by the end cap effect, which arises from the pressure effect in the pipe axial direction. The end cap strain is caused by the internal pressure acting on any curvature in the pipeline as well as at closed pipe ends (Fyrileiv & Collberg, 2005). The strain induced by the end cap effect is given by:

$$\varepsilon_{ec} = \frac{\frac{\pi}{4}(D_i^2 P_i - D_0^2 P_0)}{A_{st}} \tag{2.4}$$

where

P_0/P_i	is the external/internal pressure

D_0/D_i	is the outer/internal diameter
A_{st}	is the area of steel

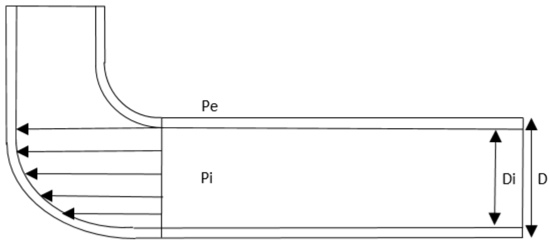


Figure 5: End cap effect

The total longitudinal strain due to temperature and pressure effects is given by:

$$\varepsilon_{longitudinal} = \varepsilon_T + \varepsilon_{ec} + \varepsilon_{Poisson} \tag{2.5}$$

2.3.2 FRICTIONAL STRAIN AND SEABED INTERACTION

The friction acting between the pipeline and the seabed is complex and difficult to determine and a great deal of uncertainty is related to predicting the resistance as the pipe moves (Bruton, White, Cheuk, Bolton, & Carr, 2006). Due to the complexity, the interaction has traditionally been modelled using Coulomb friction models (White & Randolph, 2007).

The frictional strain is linearly dependent on the submerged weight of the unrestrained part of the pipe, which acts as a force on the seabed floor. The frictional strain is zero at the pipe end and varies linearly to the anchor point (Yong & Quang , 2014). The equation for the frictional strain for an unrestrained pipeline is:

$$\varepsilon_{friction} = \frac{\mu w_s L}{A_{steel} E} \tag{2.6}$$

where

μ	is the friction factor
w_s	is the submerged weight of the pipe
L	is the anchor length
A_{steel}	is the cross sectional area of the pipe steel

2.4 EXPANSION ANALYSIS

In order to determine the maximum pipeline expansion, an expansion analysis is performed. A finite element model is often the preferred analytical model and parameters such as temperature and pressure profiles along the pipeline, the pipe submerged weight and the axial friction force is included in the model (Yong & Quang , 2014).

An expansion analysis yields the expansions at either end of the pipeline as well as the maximum axial load in the pipeline. The axial load determines if the pipeline is susceptible to buckling and the end expansion dictates the expansion that the tie-in spools have to accommodate. Both results are important for pipeline design. The maximum pipeline end expansion is calculated using the lower bound friction coefficient and the highest pipeline axial stresses (Yong & Quang , 2014).

One way of determining the total pipeline expansion is by integrating the net strain along the pipe (Yong & Quang , 2014). The net strain is the difference between the applied longitudinal strain and the frictional strain and is integrated between the free end and the anchor point. For a pipe with constant cross section and zero initial strain, and subject to constant pressures and temperatures the pressure- and thermal strain are constant and positive while the frictional strain has a linear variation (Yong & Quang , 2014). Figure 6 shows a schematic representation of the individual strain components.

$$\int_{0}^{L_{anchor}} \varepsilon_{net}\, dL = \int_{0}^{L_{anchor}} \left(\varepsilon_T + \varepsilon_{ec} + \varepsilon_{Poisson} - \varepsilon_{friction}\right) dL \tag{2.7}$$

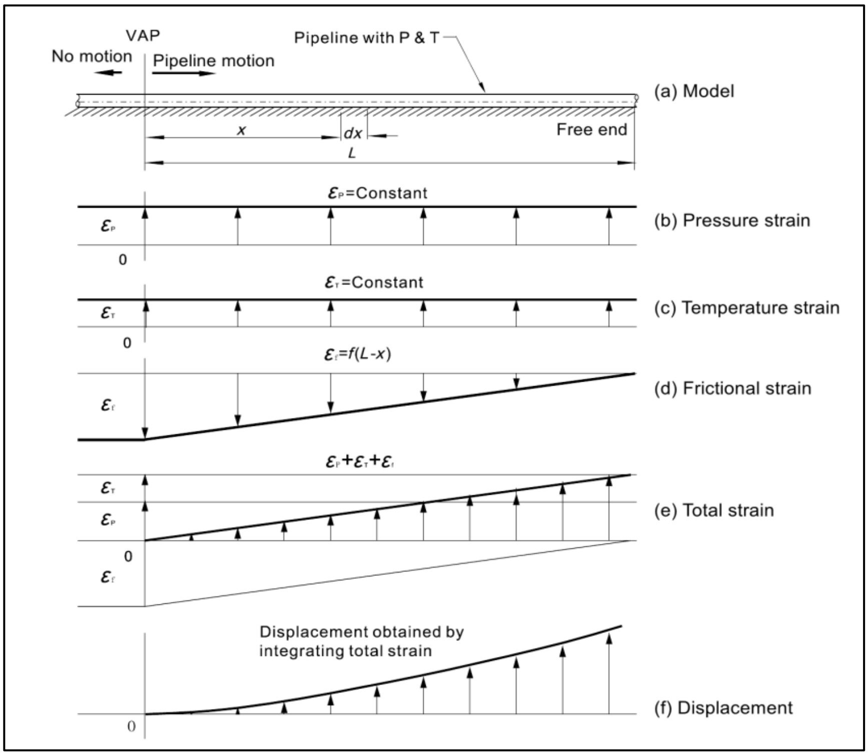


Figure 6: Distributions of strains and displacements along a pipeline (Yong & Quang , 2014)

The end expansion is important for the spool design as it dictates the expansion that the tie-in spools have to accommodate. If the axial movement due by pipeline expansion is not mitigated, the expansion may impose loads and damage adjacent structures such as manifolds and wellheads. An expansion spool is essentially a short section of pipe with bends designed to accommodate the expansion, much like a spring.

CHAPTER 3

TIE IN SPOOLS

Subsea spools are important components of a subsea assembly. Subsea technology is a highly specialized field and with developments in subsea technology, oil and gas production is becoming possible at increasingly deeper water depths (Yong & Qiang , 2012). These deepwater systems must be highly reliable and safe to avoid damage witch potentially can lead to disastrous accidents and to ensure a steady and reliable production of hydrocarbons. The following section describes the main functions of a spool-piece and the important role it plays in the subsea production system.

A tie-in spool is a short pipe section used to connect and transport production fluid between different subsea components. Spools are often tied to different types of structures at either end and may run between a pipeline and a manifold/template or wellhead or even between two pipelines.

A spool mainly serves two functions (Yong & Quang , 2014):

1) Complete the connection between pipelines and subsea structures and compensate for installation misalignments

2) Mitigate axial expansion of flowlines. In order to avoid expansion propagating to adjacent structures, spool-pieces with bends are installed to accommodate the expansions and prevent transmitting high loads into adjacent structures

A thermal expansion analysis is performed to determine the maximum pipeline expansion of the pipeline. The spool absorbs the expansion of the spool by bending and takes advantage of the spools natural flexibility. The longer the spool is the easier it is to bend. If the pipe is bent within its elastic limit, it will return to the preloaded shaped once the load is removed and it will behave much like a spring.

The structural response of the spool can be complicated to assess as it depends on a wide range of parameters, such as soil-pipe frictions, sleeper-pipe frictions, bend stiffness's and variable internal pressure and temperature during operational conditions. (Wang, Bannevake, Xu, & Jukes, 2010). Economically speaking, the most critical part of spool design, is the limited time for fabrication. When the pipeline and the connecting elements are installed, there will always be installation misalignments and it will not be possible to precisely determine the relative position of the structures prior to installation. Hence the final spool dimensions cannot be determined until the pipelines and structures have been installed and the relative distance and position has been measured.

Historically spool pieces have been the primary tie-in method pipeline tie-ins in shallow water depths (McKeehan, 1993). Traditionally, divers would measure the relative position of the connecting hubs so that the spool could be fabricated to size. Today diverless applications are made possible for moderate- and deep water installation with the use of specialist measuring techniques. These techniques are discussed further in Chapter 5.

Spools have large flexibility in design and a range of different geometrical configurations and shapes are possible in order to optimize design for any given field layout. There are a number of different systems developed for connection of subsea flowlines. The use of rigid spools is the most common tie-in method, but flexible pipelines are also used (Lewis, 2014). We can distinguish between two main types of rigid spools, vertical and horizontal. Vertical spools are generally referred to as jumpers and are vertical pipe sections that are elevated off the seabed. Vertical jumpers are susceptible to damage by trawling and are not widely used in the North Sea and are therefore not covered in this thesis.

3.1 HORIZONTAL TIE-IN

A typical spool consists of a steel pipe with two end connecting hubs. If the spool is required to accommodate pipeline expansion, bends or offsets are typically incorporated. A spool can be shaped into almost every configuration but are typically L, Z shaped or U shaped and offer a great deal of flexibility when it comes to geometrical shapes in order to optimize the design for a given field layout.

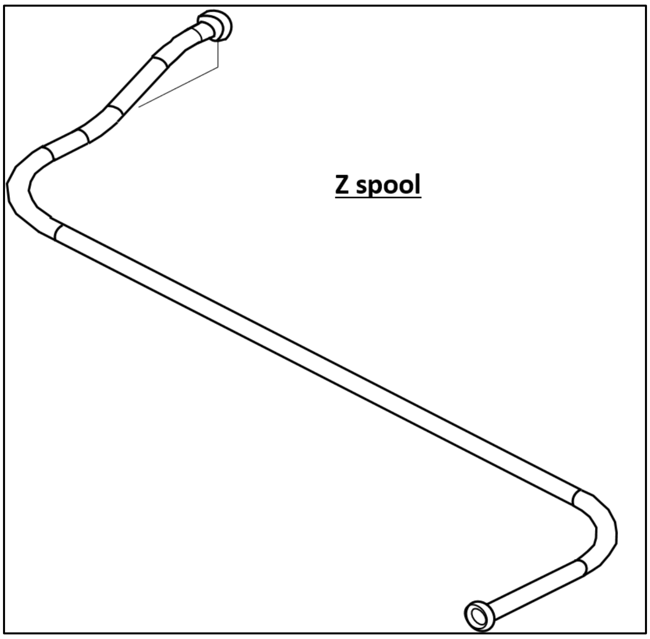


Figure 7: Z-shaped spool piece

The spool legs provide flexibility and the longer the spool legs are, the less the expansion forces are transferred to the connecting hubs (McKeehan, 1993).

The final connection between the spool-piece and the connecting structures is made with a connector. The purpose of the connector is to join and produce a pressure tight seal between the connecting structures. This is achieved by joining and sealing the two connector hubs that are welded to each of the pipe sections that need to be joined. The final tie-in is performed using a connecting tool usually operated from an ROV. The connecting tool clamps together the two mating hubs (Chan, Mylonas, & McKinnon, 2008). The connecting hubs of the tie-in structures are typically placed above the seabed. Goosenecks are therefore necessary to rise the last part of the pool above the seabed to connect with the raised hubs. Clamp connectors are the most common type of connectors used in the North Sea for diverless interventions (Corbetta G., 1997). A clamp connector consists of a gasket that is placed between two flanges that are forced together. They can have two bolts, but for ROV operated clamp connectors, one of the bolts replaced with a hinge (Corbetta & Cox, 2001).

The tie-in process is generally more complicated and more time consuming, which may increase costs. For deepwater applications and for tie-ins where large flowline movement is expected, spools are considered advantageous as larger hub movements can be expected (Corbetta G., 1997). Except for deployment of the spool on the seafloor the operations have very low weather dependence since the tie-in operations is independent of vessel motion (Yong & Qiang , 2012).

CHAPTER 4

WALL THICKNESS DESIGN

As mentioned subsea pipelines are exposed to both internal pressure from the production and external pressure from the surrounding water. Additionally a pipeline under operation is usually exposed to temperatures above the ambient temperature of the surrounding seawater. Differences between internal and external pressure along with temperature effects causes pipelines to expand or contract both in the radial and longitudinal directions.

The pipe wall thickness determination is one of the most important and fundamental tasks in pipeline design. (Qiang & Yong, 2014). The wall thickness is determined on the basis of the maximum design pressures as well as being a function of material grade, diameter, water-depth and installation methods (Americal Bureau of Shipping ABS, 2006).

A spool has both bends and straight sections and according to Bruschi et al. (2006) the standards does not provide consistent design rules for pipe bends. There is wide experience in design of spools for moderate depths and the limiting conditions are well known (Lui, Hooper, & Mashner, 2014). However, most of the research into pipe bends has been focused on problems arising in industries such as process industry where effects of internal pressure are of great importance (Bjerkås, Alsos, Hval, Lange, & Holden, 2010). In subsea applications however, both external and internal pressure are acting, and limited attention has been given to this.

The following section gives a description of the design philosophy behind the DNV-OS-F101: Submarine Pipeline Systems and the liming criteria for wall thickness design of spools. The following section is largely taken form the design code itself, unless other references are stated.

4.1 LOAD AND RESISTANCE FACTOR DESIGN (LRFD)

The design of spools according to DNV-OS-F101 is based on Load and Resistance Factor Design (LRFD). In addition, bends are checked using Allowable Stress Design (ASD) in accordance with DNV OS-F101, section 5 F200.

The principle of LRFD is to verify that a set of factored characteristic design loads (L_d) is smaller than the factored design resistance effects (Rd) for any failure mode.

$$\boldsymbol{L_{Sd} \leq R_d} \tag{4.1}$$

LRFD incorporates uncertainties in the design by the use if partial load and material factors. The design load effect can be given as

$$\boldsymbol{L_{Sd} = L_F\gamma_F\gamma_C + L_E\gamma_E + L_I\gamma_F\gamma_C + L_A\gamma_A\gamma_C} \tag{4.2}$$

where L_F, L_E, L_I and L_A are the functional, environmental, interference and accidental loads respectively.

Functional loads are load imposed during installation, testing operations and general use. The loads are divided into live loads that change during operation (due to flow, temperature, pressure) and dead loads that do not change with time (hydrostatic pressure, buoyance etc.).

Environmental loads consider loads imposed by environmental phenomena such as current and waves.

Interference loads relate to loads induced by dropped objects or fishing tools etc.

Accidental loads are abnormally large loads caused by accidental events.

The load factors for pipelines are given below. They have been determined by structural reliability methods to a pre-defined failure probability.

Table 1: Partial safety factors for LRFD

Load effect factor combinations						
Limit state	**Load effect combination**		**Functional loads**	**Environmental load**	**Interference load**	**Accidental load**
			γ_F	γ_E	γ_I	γ_A
ULS	a	System check	1.2	0.7		
	b	Local check	1.1	1.3	1.1	
FLS	c		1.0	1.0	1.0	
ALS	d		1.0	1.0	1.0	1.0

The design resistance is given by:

$$R_{Rd} = \frac{R_C}{\gamma_m \gamma_{SC}} \tag{4.3}$$

R_C is the characteristic resistance, which depends on the material strength, thickness and initial out of roundness. The characteristic resistance is divided by the material and safety class factors given below.

In limit state design, all foreseeable failure scenarios are considered and the system is designed and checked against all possible failure modes. The partial safety factors are explicit in the different limit states. The relevant limits states are:

ULS – associated with single load or overload situation

SLS – not associated with catastrophic failure but reduced operational capability

FLS – ULS condition accounting for accumulated cycling load effects

ALS – implies loss of structural integrity due to accidental load

For each of the limit states a set of partial safety factors are defined using structural reliability methods. A target safety level or a maximum acceptable failure probability is given as an annual probability of failure (Mørk, Bjørnsen, & Collberg, 1998)**.**

Pipelines are designed with respect to potential failure consequence. This is achieved by introducing safety classes that describe the extent of damage to human health and the environmental and economic consequences of failure. A low safety class implies that the failure of the pipeline will not cause significant damage to human health, the environment or assets.

Table 2: Safety classes

Classification of safety classes	
Safety class	*Definition*
Low	Where failure implies insignificant risk of human injury and minor environmental and economic consequences
Medium	Where failure implies low risk of human injury, minor environmental pollution or high economic or political consequences.
High	Classification for operating conditions where failure implies risk of human injury, significant environmental pollution or very high economic or political consequences.

4.2 FAILURE MECHANISMS

In deep water, spool design is more complicated due to the high external pressure. Subsea pipes are subject to both internal pressure from the operating pressure as well as external pressure from the surrounding water (Junaidi & Koto, 2014). In addition, axial loads and bending stresses are induced due to the pipe expansion and other loads such as tie-in loads.

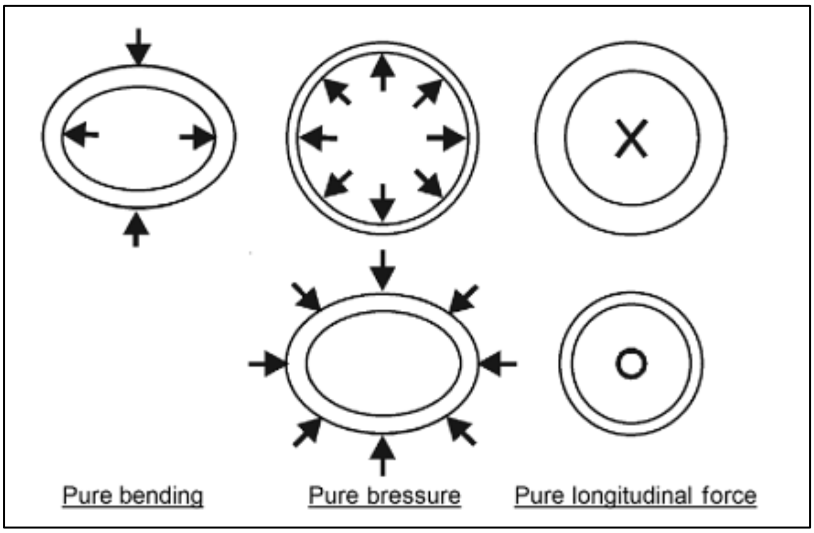


Figure 8: Cross sectional deformation of pipes subject to bending, pressure and axial load (Qiang & Yong, 2014)

Different loads induce different failure mechanisms. The wall thickness of spools is therefore checked against the following failure modes and is performed according to the requirements set by DNV-OS-F101.

1. Burst
2. Local buckling (collapse)
3. Local buckling (combined loading)

5.2.1 PRESSURE CONTAINMENT (BURSTING)

Internal pressure will cause the pipeline to stretch in all directions and the primary requirement of the pipe is to sustain the stresses from the internal pressure (Yong & Qiang, 2005). Stresses in the circumferential direction are referred to as hoop stresses.

For pipelines with high internal pressure, the pipe may fail due to bursting of the cross section. Due to the internal pressure, the pipe cross section expands and the pipe wall thickness decreases. As the wall thickness decrease the hoop stress increase and bursting or rapture occurs when a certain pressure is reached and the hoop stress is higher than the ultimate tensile strength of the material. (Qiang & Yong, 2014)

Hoop stresses are given by:

$$\sigma_h = \frac{(P_i - P_o) \cdot D}{2 \cdot t} \tag{4.4}$$

where

D	is the outer diameter
P_i	is the inner pressure
P_O	is the outer pressure
t	is the wall thickness

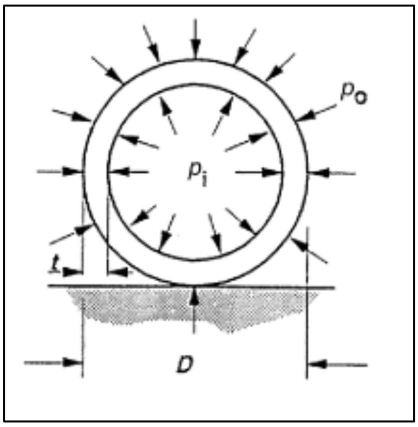


Figure 9: Hoop stress in a pipe subject to internal and external pressure (Qiang & Yong, 2014)

According to DNV-OS-F101 the burst pressure $p_b(t)$ is given by:

$$p_b(t) = \frac{2 \cdot t}{D - t} \cdot f_{cb} \cdot \frac{2}{\sqrt{3}} \tag{4.5}$$

where

$$f_{cb} = Min\left[f_y; \frac{f_u}{1.15}\right] \tag{4.6}$$

The tensile hoop stress is due to the difference between internal and external pressure and should fulfil the following criteria.

$$p_{li} - p_e \leq Min\left(\frac{p_b(t)}{\gamma_m \cdot \gamma_{SC}}; \frac{p_{lt}}{\alpha_{spt}} - p_e; \frac{p_h}{\alpha_{mpt} \cdot \alpha_U}\right) \tag{4.7}$$

$$p_{lt} - p_e \leq Min\left(\frac{p_b(t)}{\gamma_m \cdot \gamma_{SC}}; p_h\right) \tag{4.8}$$

where:

p_{li} is the local incidental pressure

p_{lt} is the local test pressure

p_h is the mill test pressure

γ_{SC} is the safety class resistance factors

$\propto_U$ is the material strength factor as given in the code

5.2.2 LOCAL BUCKLING - COLLAPSE

For subsea pipelines the external pressure from the surrounding water will help counteract the forces imposed on the pipe by the internal pressure, and reducing the risk of bursting failure. As the pipelines are installed at deeper depths, the external pressure increase and become the dominating loading condition. Theoretically a circular pipe without any imperfections will continue to hold its shape when it is exposed to uniform external pressure. However, pipes will always have some material and geometrical imperfections. If the external pressure becomes too high section of the pipe may collapse (Junaidi & Koto, 2014). The failure mode may either be yielding of the cross section or buckling on the compressive side of the pipe. For small diameter/thickness ratios failure is governed by yielding of the cross section while for larger D/t ratios it is governed by elastic buckling (Qiang & Yong, 2014).

The collapse pressure predicted by the formulas for a given wall thickness should then be checked against the hydrostatic pressure at the seabed (Junaidi & Koto, 2014).

The characteristic resistance for external pressure (p_c)(collapse) is given as:

$$(p_c - p_{ei}) \cdot (p_c{}^2 - p_p{}^2) = p_c \cdot p_{ei} \cdot p_p \cdot f_0 \frac{D}{t} \tag{4.9}$$

where

p_c	is the characteristic collapse pressure
$p_{ei} = \frac{2E\left(\frac{t}{D}\right)^3}{1-\upsilon^2}$	is the elastic buckling pressure
$p_p = f_y \cdot \alpha_{fab} \frac{2t}{D}$	is the yield pressure at collapse
$f_0 = \frac{D_{max}-D_{min}}{D}$	is the initial out-of-roundness

The pipeline is not considered to collapse if the minimum differential pressure satisfies the following:

$$p_e - p_{min} \leq \frac{p_c}{\gamma_m \cdot \gamma_{SC}} \tag{4.10}$$

p_{min} is the maximum internal pressure that can be sustained. This is normally taken as zero for as-laid pipeline.

5.2.3 LOCAL BUCKLING - COMBINED LOADING

Spools in installation and operating phases are not only subject to pressure loading but also need to withstand high bending moments due to pipeline expansion, connection loads, and structure settlement as well as axial loads. For the combination of pressure, longitudinal forces and bending the stress level at failure is an interaction between the longitudinal and hoop stresses induced by the different load combinations. A combined loading check is provided by the code. For the purpose of the check, the spool is divided into straight sections and pipe bends.

STRAIGHT PIPE

The straight sections of the pipe are designed according to LRFD design. As the pipeline expands it forces the spool to displace meaning that the response of the pipeline is primarily displacement controlled. At the same time, the seabed imposes a load on the pipe by resisting it from displacing freely often making the response somewhere between load controlled and displacement controlled. According to DNV-OS-F101 the load condition can always be used and is therefore presented below. (Det Norske Veritas, 2007).

The combined loading scenario takes into consideration the effects of design moment, effective axial force and pressure. The moment and axial force are divided by the plastic capacities for moment and force respectively. If the pipe is subject to internal overpressure, there is a risk of bursting and the pressure differential is thus divided by the burst pressure. For external overpressure the pipe may collapse and the pressure differential is divided by the characteristic collapse pressure. Refer to DNV-OS-F101 section 5 D505 for further detail.

Both the equations for internal and external overpressure are presented below:

$$\left\{\gamma_m \cdot \gamma_{SC} \cdot \frac{|M_{Sd}|}{\propto_c M_p} + \left\{\frac{\gamma_m \cdot \gamma_{SC} \cdot S_{Sd}}{\propto_c S_p}\right\}^2\right\}^2 + \left(\propto_p \frac{p_i - p_e}{\propto_c p_b}\right)^2 \leq 1 \tag{4.11}$$

$$\left\{\gamma_m \cdot \gamma_{SC} \cdot \frac{|M_{Sd}|}{\propto_c M_p} + \left\{\frac{\gamma_m \cdot \gamma_{SC} \cdot S_{Sd}}{\propto_c S_p}\right\}^2\right\}^2 + \left(\gamma_m \cdot \gamma_{SC} \cdot \frac{p_e - p_{min}}{p_c}\right)^2 \leq 1 \tag{4.12}$$

where:

M_{Sd} is the design moment

S_{Sd} is the design effective axial force

p_i is the internal pressure

p_e is the external pressure

p_b is the burst pressure

$\propto_c$ is the flow stress parameter

$\propto_p$ accounts for effect of D/t_2 ratio

p_c is the characteristic collapse pressure

p_{min} is the minimum internal pressure. Normally taken as zero except when water filled.

M_p, S_p= plastic capacities for the pipe

$$M_p = f_y \cdot (D-t)^2 \cdot t$$

$$S_p = f_y \cdot \pi \cdot (D-t) \cdot t$$

$$\propto_c = (1-\beta) + \beta \cdot \frac{f_u}{f_y}$$

$$\propto_p = \begin{cases} 1-\beta & \dfrac{p_i - p_e}{p_b} < \dfrac{2}{3} \\ 1 - 3\beta\left(1 - \dfrac{p_i - p_e}{p_b}\right) & \dfrac{p_i - p_e}{p_b} \geq \dfrac{2}{3} \end{cases}$$

$$\beta = \frac{60 - {}^{D}/_{t_2}}{90}$$

SPOOL BENDS

In the case of thin-walled pipes, bends are significantly vulnerable to ovalization and local buckling compared to straight pipes. In such cases, the highest stress is observed in the intrados bend wall, which makes it vulnerable to experience cracking leading to failure (Wang, Bannevake, Xu, & Jukes, 2010).

The Standard does not provide any limit state criteria for pipeline bends. As an alternative to LRFD design the standard provides the following simplified Allowable Stress Design, ASD, check that may be applied provided that:

- The bursting criterion is fulfilled
- The applied moment and axial load is considered displacement controlled
- The ovalistation is acceptable

- The bend is exposed to internal overpressure or that the bend has no potential for collapse.

The last criterion is fulfilled if the system collapse design capacity is three times the actual external overpressure. This recommendation usually leads to significantly thicker wall thickness for the bends than what is required for the straight sections. The increased wall thickness causes the pipe stiffness to increase, causing the spool leg lengths to increase in order to obtain the necessary flexibility needed. (Bjerkås, Alsos, Hval, Lange, & Holden, 2010)

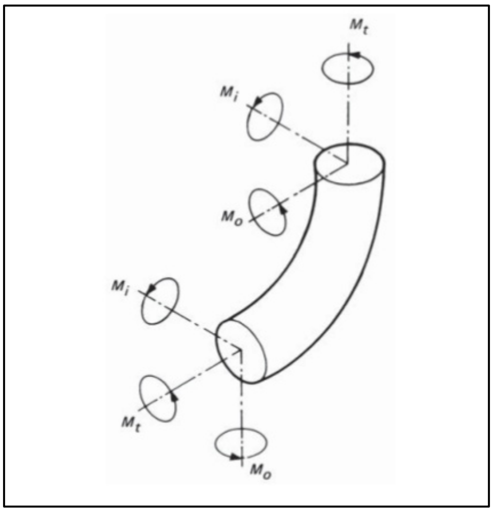


Figure 10: Moments in bends (ASME, 2010)

Allowable stress design

The stress criterion according to Allowable Stress Design (ASD) is described below. The yield stress, $\boldsymbol{f_y}$ is multiplied with a utilization factor, η. Both the equivalent stress, $\boldsymbol{\sigma_e}$ and the longitudingal stress, $\boldsymbol{\sigma_l}$ is checked.

$$\sigma_e \leq \eta \cdot (SMYS - f_{y,temp}) \propto_u \tag{4.13}$$

$$|\sigma_1| \leq \eta \cdot (SMYS - f_{y,temp}) \propto_u \tag{4.14}$$

where

SMYS	is the specified minimum yield stress
$f_{y,temp}$	is the temperature de-rating value
$\propto_u$	is the material strength factor.

The equivalent stress is given by the Von Mises Combined Stress:

$$\sigma_e \leq \sqrt{\sigma_h{}^2 + \sigma_l{}^2 - \sigma_h \cdot \sigma_l + 3\tau_{hl}{}^2} \tag{4.15}$$

The hoop stress is given by:

$$\sigma_h = (p_i - p_e)\frac{D - t_2}{2 \cdot t_2} \tag{4.16}$$

And the longitudinal stress:

$$\sigma_l = \frac{N}{\pi \cdot (D - t_2) \cdot t_2} + \frac{M}{\frac{\pi \cdot (D^4 - (D - 2 \cdot t_2)^4)}{32 \cdot D}} \tag{4.17}$$

where

N is the pipe wall force

M is the bending moment

The usage factors depend on the safety class are given below

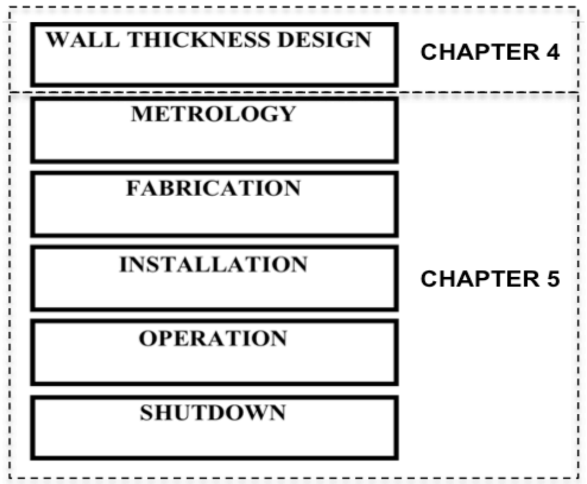
Table 3: Usage factors for ASD check for bends

Usage factors for equivalent stress check			
	Safety class		
	Low	*Medium*	*High*
η	1.00	0.90	0.80

CHAPTER 5

SPOOL DESIGN CONSIDERATIONS

After the wall thickness of the spool has been determined based on the working pressures, the spool dimensions need to be determined for the fabrication stage. The spool dimensions are determined based on subsea measurements taken once the connecting structures has been laid. The spools are then fabricated and sent offshore for deployment. Once they are installed, they are tied-in with the connecting structures and finally tested and approved. Since a spool is subject to a range of different loadings related to the different stage in its life cycle, a full FE analysis should considered both direct loads and also through accumulated loads by performing a load step analysis. The following section discusses the process from subsea measurements, to fabrication as well as loads associated with installation, testing, operation and shutdown.



5.1 METROLOGY

The tie-in structures are often installed within a predefined area, or target box, which is positioned relative to the pipeline. The positioning of the structure is normally not precise enough however, for the spool piece to be prefabricated (McKeehan, 1993). Consequently, the final spool assembly cannot be performed until the pipelines and connecting structures have been installed and the relative orientation and distance between the hubs of the pipeline end and the hub of the tie-in structure need has been determined.

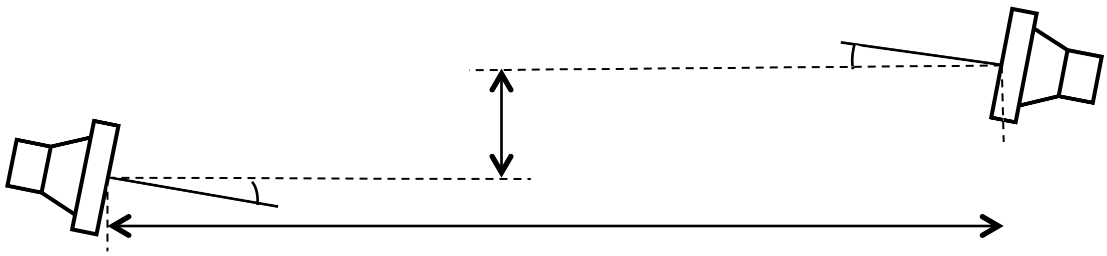


Figure 11: Distance and angular orientation of connecting hubs

A specialist metrology survey is performed and several measuring techniques are available. Taut wire metrology is mainly used by divers at shallow water depths, while acoustic positioning systems such as acoustic measuring techniques can be used at deeper water depths.

5.1.1 TAUT WIRE METROLOGY

Taut wire is a basic measuring technique first employed by divers. The technique involves a wire being tightened between the two hubs. The length and the angles between the wire and the pipe are measured with tape a protractor or by the use of sensors (Corbetta & Cruden, 2000). Traditionally the taut wire system is used by divers, making the technique limited for use at shallow water depths. Newer developments allow for measurements to be performed by wire-length sensors and angle sensors and for data to be transmitted acoustically or by electric wire to an ROV (Alliot, 2006).

5.1.2 ACOUSTIC POSITIONING

Acoustic positioning is widely used for subsea measurements. The acoustic systems take advantage of sound propagation and reflection and use a set of calculations to determine the position of transducers positioned on the sea bottom (Milne, 1983). An array or framework needs to be established prior to the measurements and the required positions are measured relative to the array (Christ & Wernli Sr., 2013).

In the case of **Long Baseline (LBL) systems** the array is made up of transponders deployed at the sea floor, often around the perimeter of the site. Target transducers are placed on the pipeline flanges and hubs and whilst emitting an acoustic pulse, which is detected by each of the transponders in the array. The LBL system provides very high position accuracy independent of water depth, however, the installation of the system is time consuming and the system requires precise calibration (Christ & Wernli Sr., 2013).

Instead of being mounted on the seabed, the **Short Baseline (SBL)** and **Ultra Short Baseline (USBL)** acoustic systems use arrays of acoustic transducers deployed on the side or at the bottom of a surface vessel. The transceiver on the vessel detects acoustic signals from the pre-installed targets on the spool hubs and the distance is determined by knowing the precise time and speed it took for the acoustic signal to travel between the two (IMCA, 2012). In the SBL system, one transceiver transmits the signal but receive from all transducers. The USBL systems are similar to SBL systems except that the transducers are all built in to a single transceiver. USBL are also used to dynamic position DP vessels in relation to a subsea reference transponder (Christ & Wernli Sr., 2013).

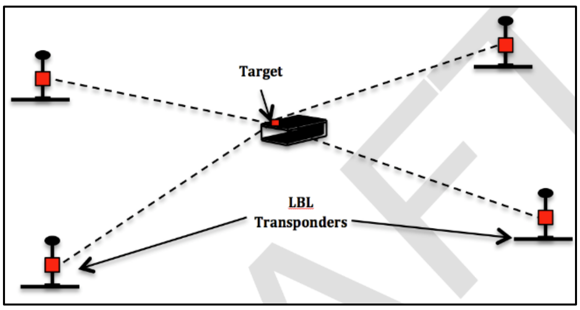


Figure 12: Long Baseline Acoustic Metrology

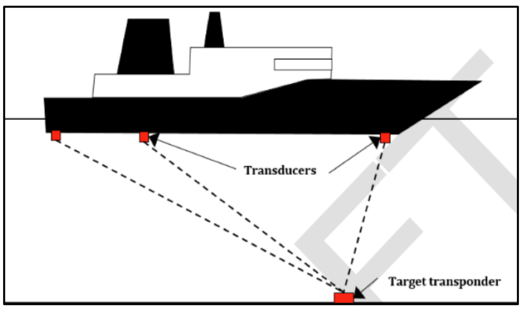


Figure 15: Short Baseline Acoustic Metrology

5.2 FABRICATION

Spools are designed and fabricated according to the measurements provided by the metrology survey. During the fabrication, all the different components such as line pipe, bends and connector hubs are assembled ready for installation. Welding qualification tests are often performed using non-destructive testing (DNT), followed by a hydrotest in order to assess the structural integrity (Antaki, 2003).

Since the survey cannot be performed until the connecting structures are installed there is limited time for design optimization and fabrication.

5.2.1 METROLOGY AND FABRICATION TOLERANCES

Without an accurately constructed spool, the spool and pipe flanges will not fit together. Any inaccuracies in the methodology or the fabrication of the spool, will lead to the connector hub faces not aligning perfectly when installed. Residual loads will then arise when the spools are tied-in, as the spools will need to deform to make up the misalignment (Juluri, Dib, el-Gebaly, & Cooper, 2013). Accurate methodology and fabrication is important in order to ensure that the spool hubs align well with the connecting structures in order for the tie-in loads to remain as low as possible. Requirements are therefore made regarding the metrology and fabrication tolerances, in order to keep alignment loads to a minimum during tie-in.

A set of translational and angular misalignments are set to account for the tolerances. Translational misalignments usually depend on the spool geometry whilst the angular misalignments at the hub face is usually given (Jacobsen, Norland, & Tharigopula, 2015).

The spool design requires careful consideration into all the possible combinations of angular and linear tolerances and misalignments to find the combination of loads that gives the maximum allowable misalignment tolerances (Chan, Mylonas, & McKinnon, 2008).

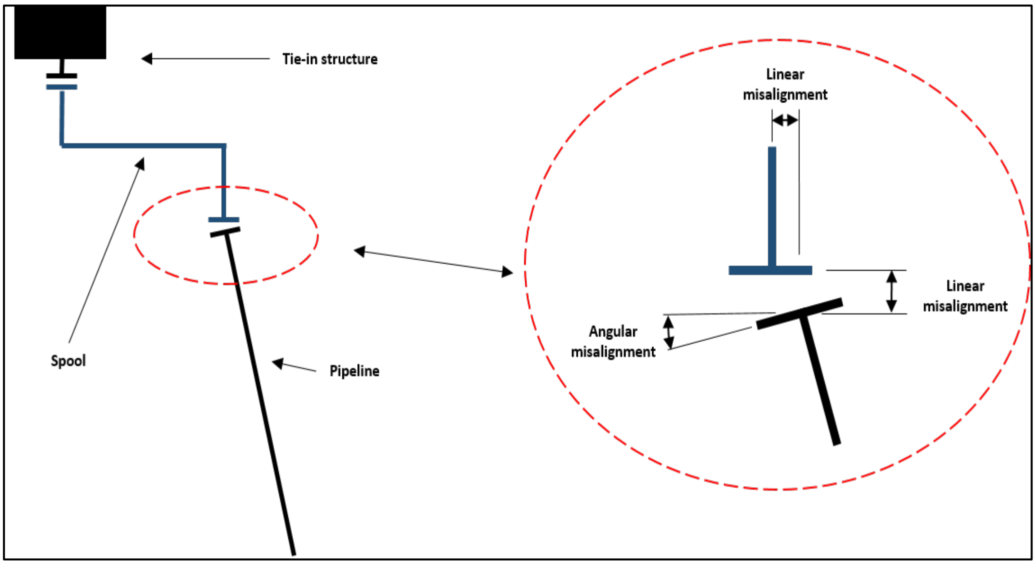


Figure 13: Linear and angular flange misalignment

5.3 INSTALLATION AND TIE-IN

During installation, the spools are transferred from the fabrication yard to the final position on the seabed. Prior to the installation, a lifting analysis should be performed to define the rigging arrangement. Spools are often installed using framework or spreader bars structure as seen in Figure 16. The spool is fastened the framework to support the spool during the lift and avoid collapse as the spool is lifted and lowered through the water column. As spools vary greatly in size and shape, the framework structure or spreader bars often need to be made specifically for each lift. Chan et al (2008) suggest that the width of the spool should be kept to a minimum and recommend that the centre of gravity is close to the spool main axis.

Figure 14: Spreader bars used for spool installation (Chan, Mylonas, & McKinnon, 2008)

The lifting vessel is set in motion by the wave motion, which again is transmitted to the crane tip. As the spool is lowered into the splash zone waves will lead to impact forces on the spool and buoyancy forces will start to apply as it is submerged. Dynamic lifting analysis is usually performed using finite element programs such as SIMO or OrcaFlex to determine the forces on the spool being lifted. Often the main limitation for installation is the overboarding of the spool into the splash zone (Reinholdtsen, Sandvik, & Hansen, 2002). The spool piece can be lifted through the splash zone horizontally or it can be tilted in order to control flooding and to reduce wave loads.

5.3.1 INSTALLATION TOLERANCES

In order to be able to perform the design prior to installation of the connecting structures, target boxes may be used. The purpose of a target box is to give a predefined area for structures such as PLET's and manifolds to be installed in. The dimensions of the target box will be slightly larger than the structure itself and the higher the accuracy of the installation equipment, the smaller the target box can be. The target box enables the design to be completed prior to installation by assigning installation tolerances based on the target boxed, and thus designing the spool based on minimum potential leg lengths as a conservative approach.

The design must ensure that the leg length and rotational capacity are sufficient to accommodate independent tolerances at extreme limits in all directions within the predetermined installation envelope and according to the dimensions of the target boxes.

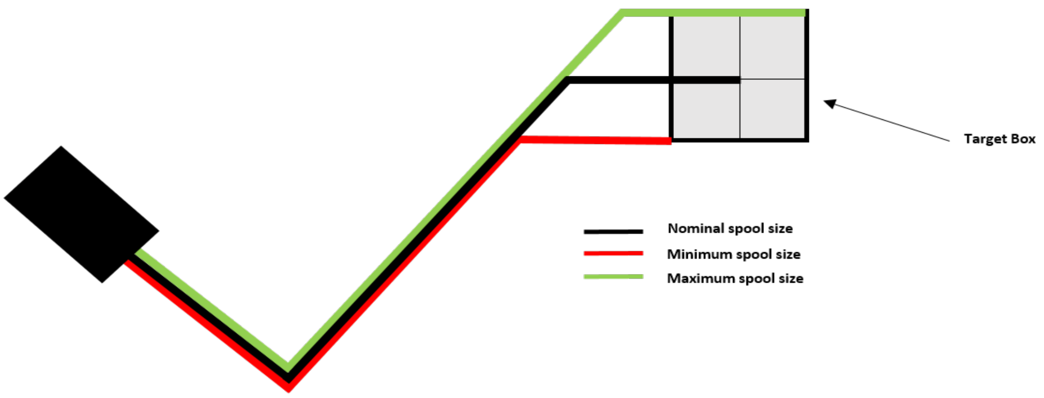


Figure 15: Minimum and minimum spool sizes

5.3.2 WIRE RESONANCE

For deep-water installation of spool-pieces, the length of the lifting wire becomes very long. As the length increased the natural period of oscillation for the vertical motion of the load will increase. If the natural period becomes long enough it may come into the same range as the wave period in the area and resonance motion may set in. Resonance will increase the dynamic load on the crane and may damage the equipment (Nam, Hong, & Kim, 2013).

Active Heave Compensation may mitigate some of these effects but resonance motion is an important issue and may be critical for the operations.

5.3.3 TIE-IN

Once the spool is deployed at the sea bottom, an ROV is used to manoeuvre the spool into its final position. The final tie-in is performed using a connecting tool usually operated from an ROV. The connecting tool, clamp together the spool piece hubs with the tie-in structures hubs (Chan, Mylonas, & McKinnon, 2008).

HUB capacities

The connector is a highly critical component and need to be able to withstand high pressure, bending and torsional stresses. When the pipeline expands, the spool-piece provides flexibility to make up the deformation by bending. These movements impose axial and bending stresses on the connector hubs (Duckworth, Supple, & Neilson, 1986). In addition the misalignments in angular and linear directions between the hubs of the structures need to be corrected for during tie-in, which will impose additional loads on the connectors.

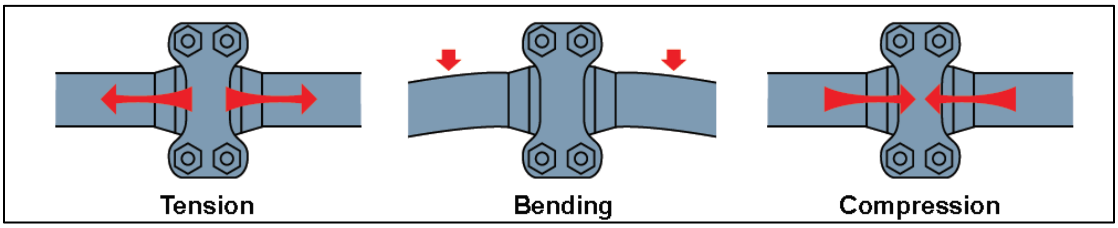


Figure 16: Forces applied to hub during tie-in and operation (Sletteboe, 2012)

Several different connector systems are available, but the main tie-in methods are flanged connectors, mechanical connectors or hyperbaric welding. For each connector system a set of allowable loads are given. These loads depend on the connector ability to make up misalignments and to handle the imposed forces. Table 4 presents an example of load capacities for a typical connector.

Table 4: Allowable loads for a typical connector

		Forces [kN]			Moments [kNm]			
		F_x	F_y	F_z	M_x	M_y	M_z	$M_{bending} = \sqrt{{M_y}^2 + {M_z}^2}$
12”	Tie-in	±100	±100	±30	±50	±225	±200	300
	Operation	±100	±100	±30	±50	±225	±200	300

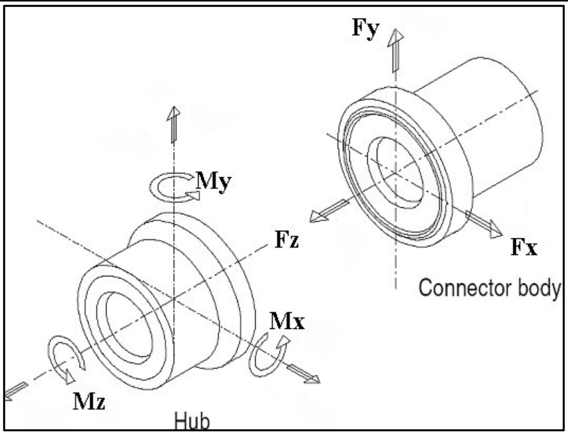


Figure 17: Hub reaction forces (Chan, Mylonas, & McKinnon, 2008)

4.4 OPERATION

During operation, the spool piece is subject to the operating pressure and temperature as well as external pressures and temperatures as discussed in Chapter 4. In addition, several factors such as the imposed hydrodynamic forces from waves and currents should form part of the analysis.

5.4.1 HYDRODYNAMIC FORCES

For pipelines, the main concern for design purposes are current imposed loadings. Wave loading normally does not apply as the effect of the wave action does not penetrate deeply enough into the water column and is mainly a concern for structures at the sea surface.

Current loadings however will impose loadings even on deepwater applications and should be included in a complete FE analysis. Other than direct loads from the current, vortex-induced vibrations (VIV) may occur which causes the spool to oscillate (Guo, Ghalambor, Lin Ran, & Song, 2014). As the current approaches a pipe span, the flow may separate downstream and shed vortices. For certain flow conditions; the vortices may shed periodically from either side of the pipe. In that case the shedding induces pressure differentials causing the pipe to oscillate in the cross flow direction (Guo, Ghalambor, Lin Ran, & Song, 2014). If the frequency of the shedding matches the natural frequency of the spool, resonance occurs, causing the oscillations to amplify and may thus reduce the fatigue life of the structure. Other than vortex shedding other loads that may influence fatigue are, start-up and shutdown cycles and for shallow water, wave action (Det Norske Veritas, 2007).

5.4.2 SOIL INTERACTION AND STRUCTURE SETTLEMENT

Geotechnical surveys are carried out at the site to extract soil data needed for analysis. The sea bottom conditions can range from rock and silt to sand and very soft clay Qiang & Yong, 2014. Friction factors are needed for any analysis where the interaction between the pipe and the soil is important. Often, both lateral and axial friction factors are needed. Typical friction factors for North Sea conditions are presented in Table 5.

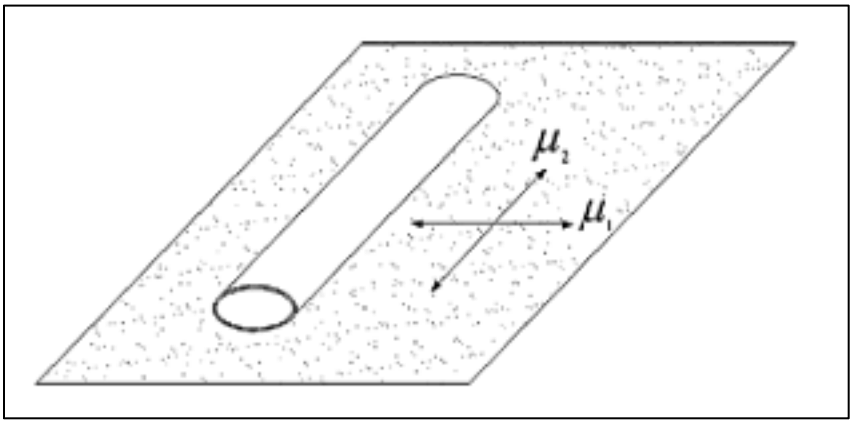


Figure 18: Lateral and Axial Friction Factors (Qiang & Yong, 2014)

Due to the uncertainty in determining friction factors, a range of factors is used and depends highly on the purpose of the analysis. In an expansion analysis a low friction is used as this leads to increased end movement in the analysis and a more conservative result. The pipe-soil interaction is vital for the expansion analysis and the friction factors are usually incorporated into the finite element analysis. (Qiang & Yong, 2014)

Table 5: Friction factors used in the North Sea (Qiang & Yong, 2014)

	Lateral Friction Coefficient		**Axial Friction Coefficient**	
Soil type	Minimum	Maximum	Minimum	Maximum
Sand (non-cohesive)	0.5	0.9	0.55	1.2
Clay (cohesive)	0.3	0.75	0.3	1.0

Other than determining friction factors another important aspect of a geotechnical analysis is to determine parameters for assessing potential structure settlement. Many subsea structures experience settlement during their field life. A pipeline will sink into soft soil until the bearing capacity of the soil is increased sufficiently to support the load. The settlement may be immediate after installation, or the soil may consolidate gradually. Pipe movement may also cause the pipe to be further embedded (Chan, Mylonas, & McKinnon, 2008). It is important to be able to predict the settlement to ensure the structure remains stable and serviceable.

Settlement also affects the friction factors. Whilst the axial friction factor remains more or less the same as the pipeline is embedded, the lateral friction increases significantly was the pipe sinks into the soil. The axial and lateral soil friction and resistance against movement are affected by pipe weight, penetration, load history and consolidation. Modelling of soil behaviour is often based on empirical relations as the physical phenomena influencing the soil resistance are poorly understood (Guan & Nystrøm, 2008).

5.5 SHUTDOWN

When the production in the subsea production system is shut down, and the pipe is empty, the internal pressure is very low compared to the external pressure for deepwater installations. The wall thickness design accounts for this condition by checking for collapse of the spool. Shutdown and start-up cycles are also important for fatigue considerations.

CHAPTER 6

METHODOLOGY

This following chapter presents the design methodology for the spool analysis. A FE analysis of the expansion spools is used to investigate the response and deformation capacity as well as end reactions. Nonlinear effects arising from large deformations and spool-seabed interaction make analytical solutions inadequate at determining the response of spool pieces under installation and operating conditions. FE modeling is therefore necessary for better analysis. The finite element program ANSYS was used to analyse the spool response.

An ANSYS script was written containing a code to run the analysis. The basic principles of finite element modeling are explained along with nonlinear analysis. The section also presents a detailed description of the FE model created for the analysis.

6.1 FINITE ELEMENT MODELING

Finite element analysis is a powerful tool when used to estimate the capacity of pipelines and spool subject to combined loads. A finite element model allows the designer to model aspects such as the geometry, material properties and defects under different loading scenarios. Today's commercial software's are very powerful and complex and have the ability to solve many different types of problems ranging from thermo- mechanics to electro mechanics and structural mechanics. (Moaveni, 1999)

A finite element analysis is a numerical technique characterized by discretizing the structure into a large number of nodes that are connected to form finite elements. Loads and other constraints are applied at the nodes, which generally have three translational and three rotational degrees of freedom. Each node is defined locally in matrix form as well assembled in the global matrix. Values of known displacements are determined through a series of matrix operations performed by the program (Young & Budynas, 2002). Different types of elements, such as solid elements or shell elements can be assigned, depending on the problem.

The FE analysis in this thesis was performed using ANSYS mechanical APDL 15.0. The following section provides a brief description of FE modeling and a description on creating a suitable model for the spool design.

6.1.1 PROGRAM STRUCTURE

Much like an engineering problem, the program divides the process into a pre-processor where a model of the physical problem is created, a solution processor where the problem is solved and a post processor, where the results are analysed.

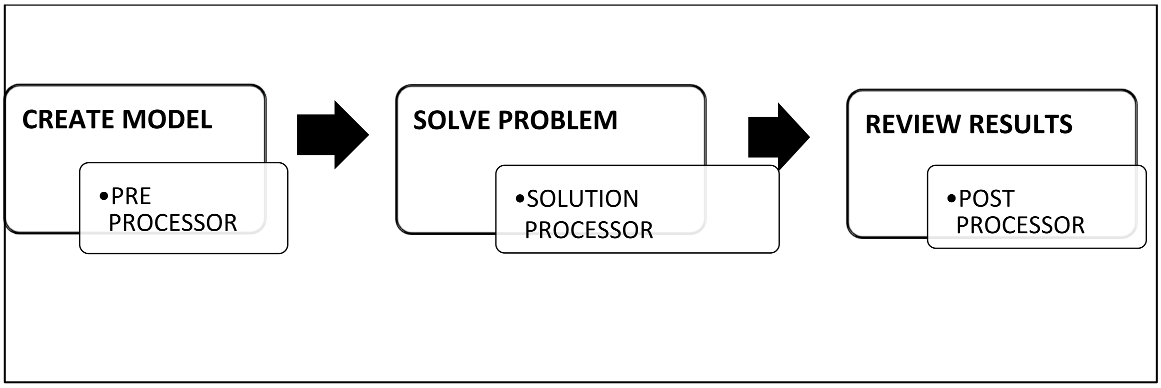


Figure 19: FE program structure

In the preprocessor, the model is built. The geometry of the spool is defined through creating nodes. Elements are assigned to the geometry and the area is meshed to form a large number of finite elements. The smaller the mesh, the more elements are created and the more accurate the analysis is. At the same time, the analysis becomes less economical and more time consuming (ANSYS, 2009).

Properties such as wall thickness and material properties of the spool are associated and assigned to the pipe elements. The physical boundaries, such as the seabed can be modeled and properties such as seabed friction can be assigned to the seabed elements.

In the solution processor the type of analysis is assigned and the program will solve the problem when prompted. The solution is obtained and the post processor is then entered to extract the results and produce plots and tables.

6.1.3. ANALYSIS MODE

A spool is a complicated structure designed to accommodate large displacements. The large displacements cause difficulties when trying to perform traditional static linear analysis. In a non-linear static analysis there are up to three sources of non-linearity, all of which may apply for a spool analysis (Yong & Qiang, 2005).

- Material non-linearity
- Geometric non-linearity
- Boundary non-linearity

By using a FE program, all of these sources non-linearity can be considered. Geometric nonlinearity is characterized by large displacement and/or rotations causing the structure to change geometry as it deflects. Some elements allow for geometric nonlinearities being accounted for in the model. The stiffness contribution calculated due to strains caused by the change in geometry is then added to the overall stiffness matrix (Yong & Qiang, Subsea Pipelines and Risers, 2005). Material nonlinearities may also apply if the resulting loads cause stresses beyond the yield strength (ANSYS, 2009).

Contact or sliding between two structures is also a source of linearity and can be quite tricky to model. The pipe and seabed are modeled using contact and target elements that try to model the frictional resistance and penetration caused when the pipe comes into contact with the seabed.

ANSYS uses the Newton-Raphson approach to solve nonlinear problems. The approach divides the load into a series of increments and the load is then applied over several substeps instead of being applied instantaneous.

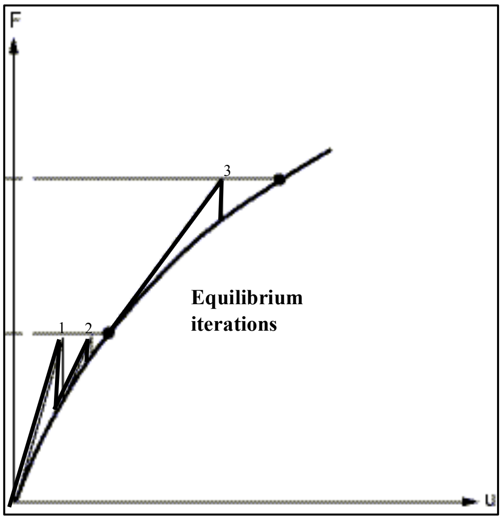


Figure 20: Newton-Raphson interactive analysis (ANSYS, 2009)

At each load step, the program evaluates the difference between the restoring forces and the applied loads and performs a nonlinear solution. To do this the program checks for convergence and if the convergence criterion is not satisfied, the stiffness matrix is reevaluated and updated in an iterative process until the difference between the external and internal forces become acceptably small (ANSYS, 2009). By using load steps the load history can be divided in to a series of different loading scenarios. Different loads and boundary conditions can be applied at the different load steps and a solution for each step can be obtained. For static analysis the loads are assumed to vary linearly within each substep.

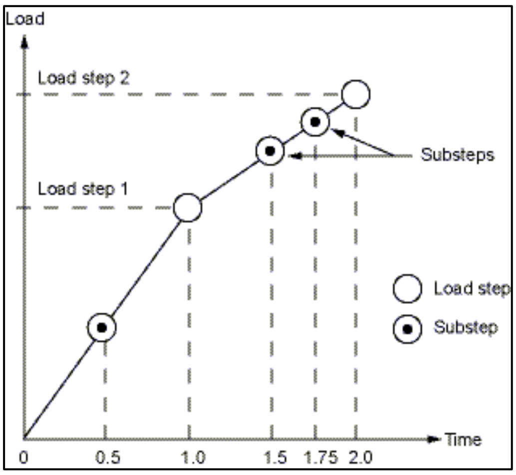


Figure 21: Load history divided into load steps and substeps (ANSYS, 2009)

Convergence can sometimes be hard to obtain. If the load is applied too rapidly the force balance may be too difficult to obtain and the load should be applied over a larger number of substeps. Mesh refinement may also help correct the force imbalance.

Nonlinearity in an analysis may also lead to convergence issues. Frictional contact with large friction factors and structures subject to large may cause convergence difficulties, both of which commonly apply in a spool analysis (Higgins, 2012).

Another issue, which may cause convergence issues in a spool analysis, is if the two contact surfaces are modeled with an initial gap between the pipe and the seabed. This causes the contact pair to be initially displaced and there is a risk of convergence issues caused by rigid body motion (ANSYS, 2009).

6.2 DESCRIPTION OF THE FINITE ELEMENT MODEL

The following section gives a description of the finite element model used to analyse the spools. Three different spools were analysed and a FE model was created to model the response of each spool. The spools are assumed to be tied-in between a PLET and a manifold/template and are designed to take a 1m pipeline expansion applied at the PLET end, whilst keeping the end moments within the limits of the hub capacities.

The spools are also designed to apply with requirements such as metrology and fabrication tolerances and pressure tests as well as design and operating conditions, which are presented in the design basis. Each loading condition was applied to the spool in a series of load steps.

All spools were designed to comply with the following:

- Minimum bend radius = 5*ID
- The pipe will approach the manifold at 45 degrees.
- The spool approach is normal to the connecting structure
- Gooseneck geometry is same for all spools
- All spools have the same pipe material properties, wall thickness and diameter.

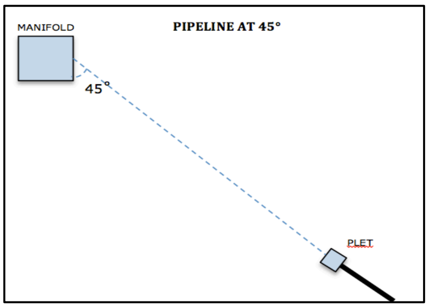


Figure 22: Pipeline approach

6.2.1 ASSUMPTIONS

The following assumptions were made for the spool design.

- Flat seabed
- The pipe elements are modeled with nominal wall thickness and thinning of bends is not implemented.
- The effect of hydrodynamic forces is not considered
- Pipeline walking is not included in the analysis
- Structure settlement has not been included

6.2.2 SPOOL CONFIGURATIONS

Three different spool geometries were analysed. A 3-legged spool was designed first and then the spool legs and bends were subsequently increased for the two subsequent spools. An ANSYS pre-processor script was made for each spool configuration, and the different load steps were applied using the solution processor.

3-LEGGED SPOOL

The first spool to be designed is a 3-legged L shaped spool, with a third leg at 45 degrees as shown in Figure 27. It only has 2 bends, which means little welding is required. Legs A and B are equally long and separated with a 90 degree bend.

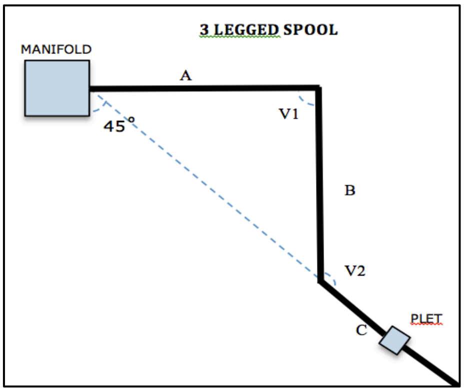


Figure 23: 3-legged spool layout

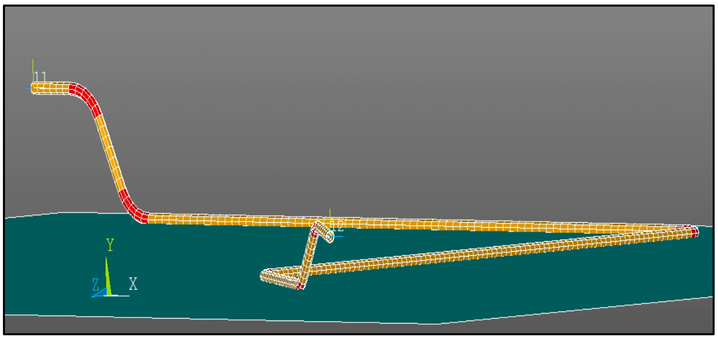


Figure 24:ANSYS model of 3-legged spool

4-LEGGED SPOOL

Figure 29 shows the layout for the second spool design. It consists of a spool with 3 bends and four legs. It is shaped as a Z-spool with a fourth leg at 45 degrees.

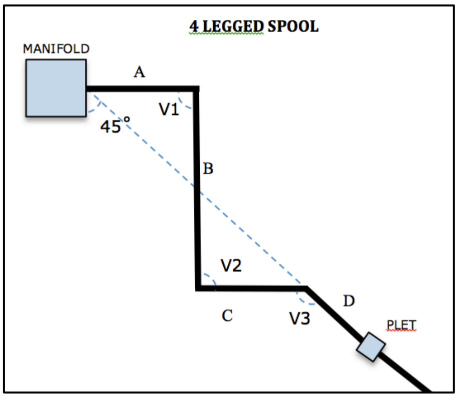


Figure 25: 4-legged spool layout

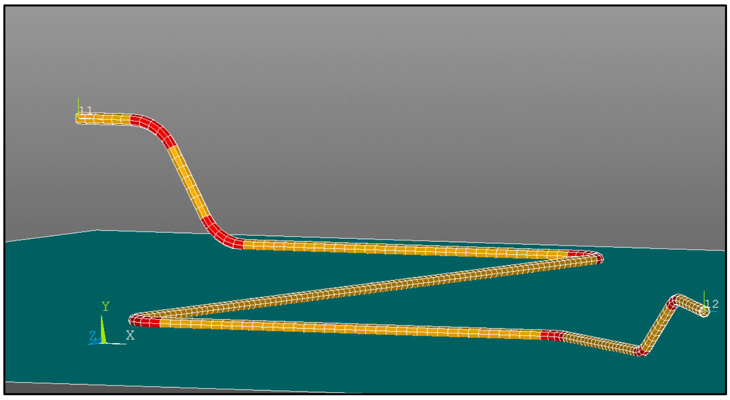


Figure 26:ANSYS model of 4-legged spool

5-LEGGED SPOOL

The final spool analysed was a 5-legged spool designed with a U-shaped bend as shown in Figure 31-32.

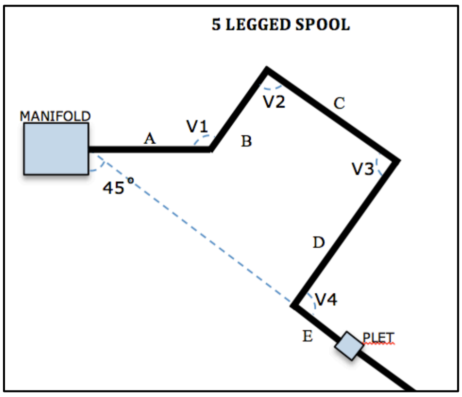


Figure 27: 5-legged spool layout

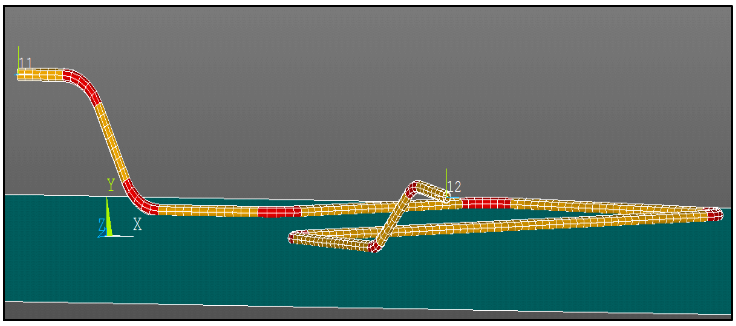


Figure 28: ANSYS model of 5-legged spool

6.2.3 ELEMENTS TYPES

ANSYS provides a large range of element types and different elements process different characteristics and should be chosen based on the analysis type and problem. The following elements were used in the analysis:

1	PIPE288	Pipe straight section
2	ELBOW290	Pipe bend
3	TARGE170	Seabed
4	CONTA177	Contact element on pipe in contact with seabed

A presentation of the elements with a brief discussion on the main characteristics of the elements is given below. The information is gathered from the ANSYS Theory Reference (ANSYS, 2009)

PIPE288

The straight sections of the pipe are modeled using a PIPE288 element. The PIPE288 element is a two-node 3D pipe element which has six degrees of freedom at each node. The element can exhibit linear, quadratic or cubic shape functions and is based on Timoshenko beam theory, which means that shear deformations are included. The element is suitable for analyzing slender pipes and is well suited for linear, large rotation and/or large strain nonlinearity applications (ANSYS, 2009)

The PIPE288 element allows for change in cross sectional area in the case of large deflection analysis. By selecting appropriate key options, the element supports thin-pipe options where a plain stress state in the pipe is assumed.

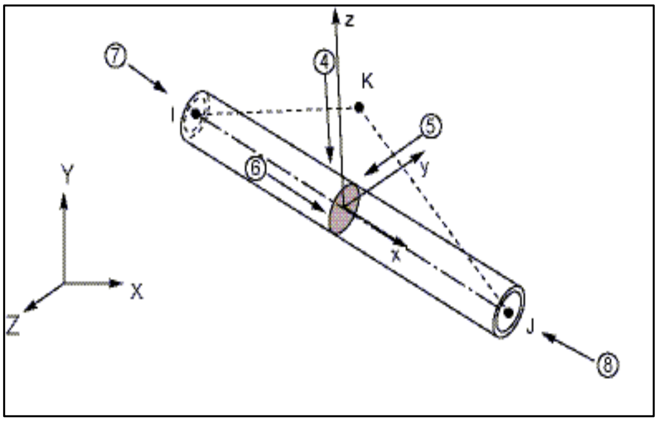


Figure 29: Pipe288 element (ANSYS, 2009)

ELBOW290

The pipe bends are modeled using ELBOW290 elements. The element is suitable for analyzing pipe structures with thin to moderately thick pipe walls. The elements has three nodes, each with six degrees of freedom and is well suited for large rotation /large strain nonlinear applications and also allow for modeling of nonlinear response of initially circular pipes distorted by ovalization and warping (Wang, Bannevake, Xu, & Jukes, 2010)

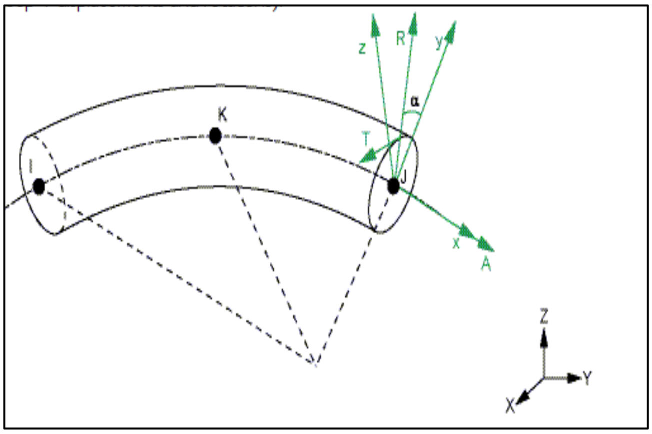


Figure 30: Elbow290 element (ANSYS, 2009)

For thin walled pipes, some ovalization and cross sectional deformation may be necessary as they are more susceptible to complex cross-section deformation which may be allowed for by selecting the appropriate key options.

CONTA175 and TARGE170

The pipe soil interaction is one of the most difficult processes in pipeline design as it poses large uncertainties (Bruton, Bolton, Carr, & White, 2008). This requires careful modeling as interaction between different structures can have great influence on results.

In ANSYS contact between to elements are often solved using contact and target elements and the contact can either be node-node, node-surface or surface-surface depending on the problem. The contact-target pair CONTA175 and TARGE170 represents a 3D node to surface contact, which can be used for modelling contact between a line and a surface and is commonly used to represent contact and sliding between two surfaces. Contact between the two surfaces occurs when the contact elements penetrate the target elements (ANSYS, 2009).

The seabed is modeled as a completely flat and rigid surface using TARGE170 elements. The element is a single quadrilateral element as shown in Figure 35.

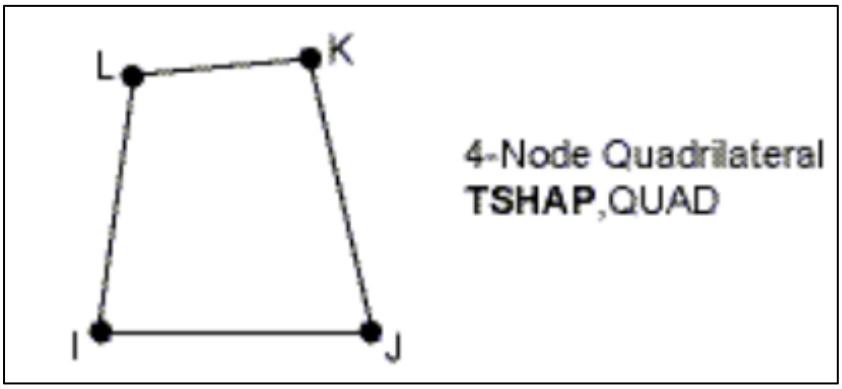


Figure 31: Targe170 elements (ANSYS, 2009)

In order to generate contact between the pipe and the soil, a set of contact elements are assigned to the sections of the spool in contact with the soil. CONTA175 are one-node elements which are assigned to the surface of the pipe sections which are in risk of coming in contact with the seabed during the analysis. The contact elements are created with the ESURF command, where the model reselects the exterior nodes and creates contact nodes on the pipe surface (ANSYS, 2009).

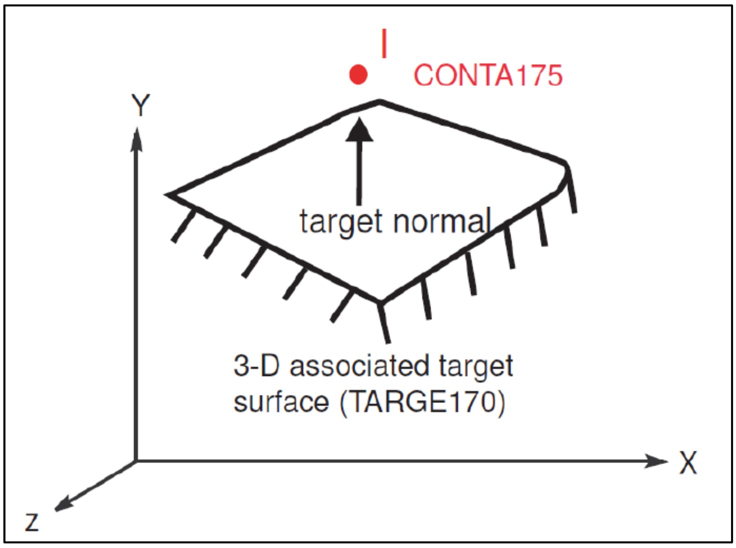


Figure 32: Conta177 elements (ANSYS, 2009)

The contact between the elements is force based meaning that forces develop in the direction normal to the contact surface in order to reduce penetration. The degree of penetration is controlled by use of a penetration tolerance factor assigned to the contact elements. Stiffness factors range from 0.1 (soft) to 1.0 (stiff) (ANSYS, 2009). A balance is needed in order to make the spoil stiff enough to limit the amount of penetration but still soft enough to avoid convergence issues which may arise when the contact is too stiff. The pipe is modeled with an initial gap of 100mm, which disappears once the submerged weight of the spool is added in the first load step

I addition to the penetration resistance, frictional forces develops tangential to the target plane in order to simulate sliding resistance. Pipe-soil frictional data is assigned based on the orthotropic frictional model.

6.2.4 MATERIAL MODELING

For pure elastic analysis it is normally adequate to describe the material characteristics by use of Poisson's ration and Young's Modulus, but for nonlinear analysis a description of the plastic behavior of the material is required. (Americal Bureau of Shipping ABS, 2006)

In order for the pipe steel to represent the complete stress/strain relationship of the pipe material, including non-linear plastic behavior, the pipe steel is modeled using an elastic-plastic model. Once yielding of the pipe material occurs, the pipe steel will deform plastically. The plastic behavior of the material is defined by specifying the stress/strain curve for the steel.

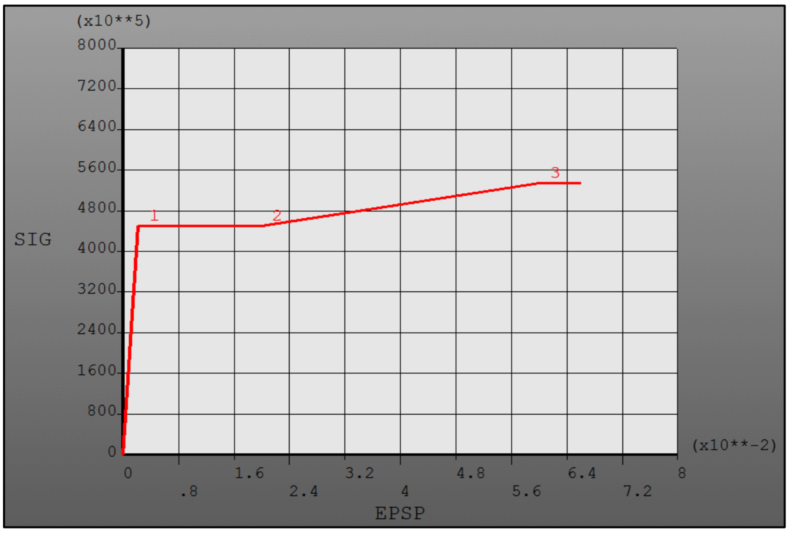


Figure 33: Stress strain relationship for pipe steel

6.2.5 PIPE DATA

Table 6, presents the spool wall thickness and material properties for the pipe analysed.

Table 6: Pipe data for analysis

Parameter		**Value**	
Material Designation		SWAL 450D + 326L CLAD	
Inner Diameter (mm)		266.7	
Bend radius		5 × ID	
Steel	Density (kg/m^3)	Thickness (mm)	
Backing	7800	Straight	16.1 + 3.0
Clad	7800	Bend	19.7 + 3.0
Young's Modulus (GPa)		207	
Thermal expansion coeff (per °C)		1.17×10^{-5}	
SMYS (MPa)		450	
SMTS (MPa)		535	
Coating	Density (kg/m^3)	Thickness (mm)	
Straights	900	49.4	
Bends	1040	61.3	

6.2.6 DESIGN AND OPERATING CONDITIONS

The spools are designed to take a 1.0 m end expansion applied to the PLET end. In addition typical values for design and operating pressure are applied at the relevant load steps. The following table presents the design and operating conditions used for the analysis.

Table 7: Design and operating conditions for analysis

Design and operating conditions	**Value**
Operating pressure (barg)	150
Design internal pressure (barg)	307
System test pressure	322
Operating temp (°C)	50
Design temp (°C)	78
Ambient temp (°C)	5
Content density (kg/m^3)	50
MEG density (kg/m^3)	1115
Sea water density (kg/m^3)	1026
Pressure reference elevation (m)	1234
Design life (years)	25

6.2.7 GOOSENECK GEOMETRY

The spools are designed with a gooseneck at both ends in order to connect the spool resting on the seabed with the elevated tie-in points of the PLET and the manifold. The manifold and PLET hub height are 3200 mm and 1500mm above the seabed respectively. The geometry of the goosenecks is described in Table 8.

Table 8: Gooseneck Geometry

End	Tie-in point elevation [mm]	Bend angle [˚]	
Manifold	3200	60	
PLET	1500	30	35
Note: 1. Hub heights are measured from seabed to the centerline of the hub			

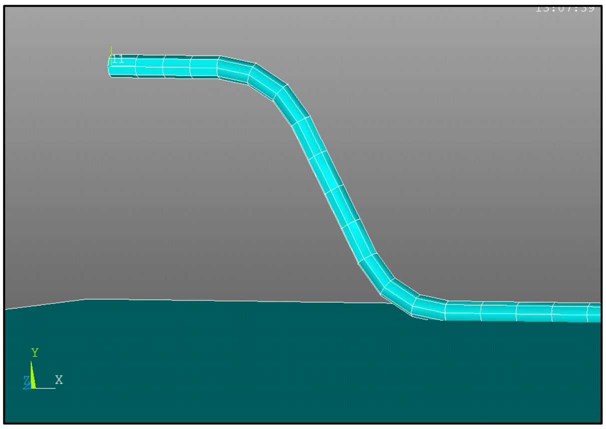


Figure 35: Manifold gooseneck

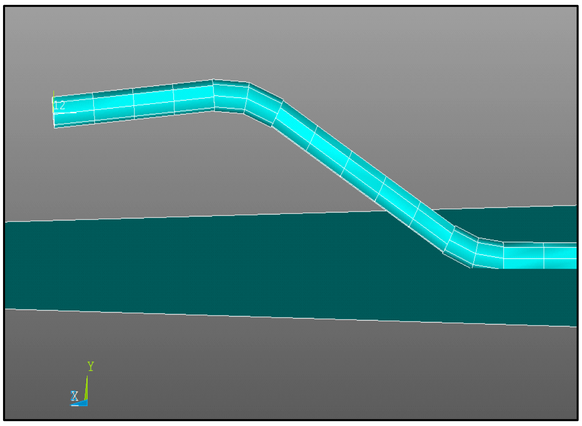


Figure 34: PLET gooseneck

6.2.8 TOLERANCES

In order to access the effect of the metrology and fabrication tolerances, the combination that gives the highest reaction forces is found. Since there are both linear tolerances for the spool length as well as angular tolerances for each hub, all different combinations of the tolerances need to be checked. This was done by applying the different tolerances to the spool and assessing witch one yielded in the worst load case.

The individual tolerances for linear and angular displacements are given in Table

Table 9: Fabrication and metrology tolerances for analysis

Spool	Horizontal [mm]	Vertical [mm]	Heading (ROTZ) [deg]	Tilting (ROTY) [deg]
Fabrication	±10	±10	±0.2	±0.2
Metrology	±100	±50	±0.5	±0.5
Total	±110(1)	±60(3)	±0.7(2)	±0.7(2)
Notes: 1. The value is measures from hub to hub 2. The value is measured from each hub 3. The value is measured relative to the seabed elevation				

6.2.9 END TERMINATIONS

The spool is modeled as being the connection between a subsea manifold and a pipeline. The pipeline end is supported using a PLET. Both the manifold end and the PLET end are modeled as fixed connections.

Table 10: Hub capacity values used in analysis

Forces [kN]			Moments [kNm]			
FX	FY	FZ	MX	MY	MZ	$\sqrt{M_y^2 + M_z^2}$
±100	±100	±30	±50	-	-	±300

6.2.10 FRICTION FACTORS

The following friction factors were used for the analysis:

Table 11: Friction factors using in analysis

Friction factors		
Lateral	μ_x	0.8
Axial	μ_y	0.8

6.2.11 LOAD SEQUENCE

The following load steps are considered in the analysis. The load sequences are run as a load history, thus ensuring that any loads potentially accumulated through the previous loads steps are included. The load steps are explained below and summarized with applied temperature and pressure values for each load step in Table 12.

Table 12: Load sequence

Load step	Description	Application	Temperature (deg)	Internal pressure (barg)
1	**Alignment**	• Set boundary conditions at ends • Apply hydrostatic pressure • Apply self-weight	None	None
2	**Tie-in**	• Apply metrology and fabrication tolerances	None	None
3	**Pressure Test**	• Set internal pressure to relative test pressure	None	322
4	**Operation condition**	• Internal pressure and temperature is set to operation pressure and temperature • Expansion applied to PLET end	50	150
5	**Design condition**	• Internal pressure and temperature is set to design pressure and temperature • Expansion applied to PLET end	78	307
6	**Shutdown**	• Internal pressure set to ambient pressure, and temperature set to ambient temperature • Expansion removed	0	0

- **Self-weight / Alignment**

  The spool is locked into its position at each end and the self-weight of the spool is applied. This is done by calculating the equivalent density of the empty pipe and assigning this as the spool density. The content density is then added. At this step the spool is modeled as being MEG filled. The external pressure on the spool is also applied.

- **Tie-in**

  During tie in, the spool is exposed to forces due to displacements based on the metrology and fabrication tolerances established. The combination of the tolerances, which presents the worst-case scenario, is applied to the spool ends.

  The stoking of the spool is not included in the analysis.

- **Pressure test**

  The internal test pressure is applied to the spool

- **Operation**

  During operating condition the spool content is changed to the operating content density and internal pressure and temperature is set to operating pressure and temperature.

  The pipeline expansion is applied as a horizontal displacement on the PLET end.

- **Design Condition**

  Internal pressure and temperature is set to design pressure and temperature

- **Shut down**

  The internal pressure is removed and the temperature is set to the ambient temperature. The expansion at the PLET end is removed.

CHAPTER 7

RESULTS AND DISCUSSION

The results from the spool analysis are presented in this chapter. Firstly a simplified model was analysed and the results compared to analytical results in order to get a feel for the validity of the ANSYS model. The results of the main analysis are then presented. Lastly a sensitivity analysis shows the effect of reduced mesh size.

Results are assessed in terms of the objective of the thesis which is:

- To design horizontal subsea spools able to accommodate a 1m pipeline expansion whilst complying with the limitations set by the hub capacities.
- Analyse different spool shapes in order to judge their ability to accommodate the imposed loads.

7.1 VALIDATION OF MODEL

A spool subject to pipe expansion will impose forces on the connecting hubs. The magnitude of the forces depends on a number of factors. Spools absorb deflection by bending, and thus, the inherent flexibility of the spool governs the resulting forces. Because spools are resting on the seabed, some flexibility is lost due to the passive soil restraint which acts between the spool and the seabed as shown in Figure 38, and the reaction forces are thus a function of the pipe-soil interaction (American Lifelines Alliance, 2001).

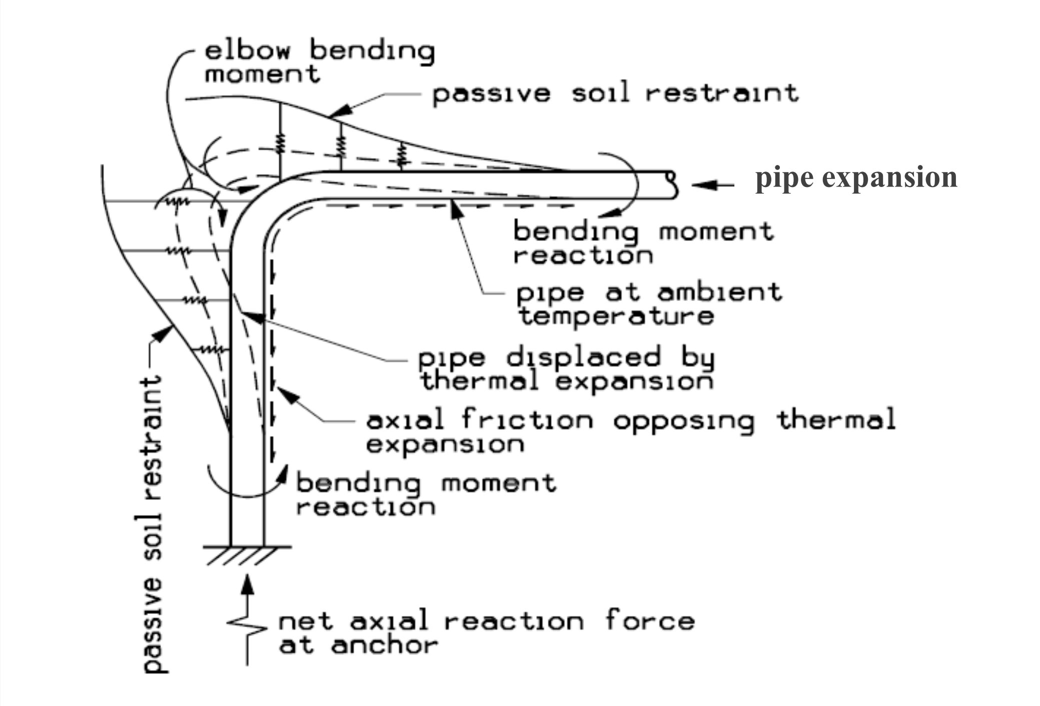


Figure 36: Bending moment and reaction forces of restrained spool due to pipe expansion (American Lifelines Alliance, 2001)

Non-linear analysis is required to capture the effects of large deformations and pipe-soil interaction, and a finite element analysis is therefore the preferred analysis tool for spool (American Lifelines Alliance, 2001). It is important however, to verify that the output from the FE analysis is correct. Ideally, this is done by replicating the structure in a model and verifying the results by physical testing of the model or by comparing with other experimental results. Other options are to perform the analysis in another FE program such as Orcaflex, to check if they yield similar results.

In order to test the FE analysis a simplified model was created so that it could be compared against analytical results. The hub capacities are often limiting for the design and the end reaction forces are therefore important to investigate. One possible approach is to model the spool as a rigid frame. “Roark’s Handbook” (Young & Budynas, 2002) offers equations to calculate reaction forces for a large range of different frame configurations.

The equations given for the rigid frames however, assume small deformations where second order effects are neglected. In a linear analysis we assume that the deflection and stresses are proportional to the load. Non-linear second order effects are a result of either joint deformation or the structure is deformed to such a degree that geometric nonlinearities come into effect. The result is that forces are not transmitted linearly from one member to another and thus introducing additional forces and increasing displacements and moments (Young & Budynas, 2002).

Spools are slender structures that are designed to accommodate large displacements. Second order effects can therefore normally not be ignored. The rigid frame approach also assume that corners remain the same angle through the deformation, but depending on the amount of expansion is applied to the spool this will generally not be the case.

The FE model was therefore designed so that the second order effects would be minimal and non-linear behavior such as soil interaction was ignored, in order to get reasonable results for comparison.

7.1.1 SIMPLIFIED ANSYS MODEL

The spool was modelled with 3 legs connected with 90-degree bends to minimize second order effects. In addition the spool was designed without goosenecks to minimize out of plane deflections. Figure 39 shows the simplified spool configurations.

As a first approach the following assumptions were applied:

- No seabed interaction
- Self-weight and buoyancy excluded
- Manifold end fixed
- Pipe cross section uniform throughout spool

The spool was designed to accommodate a 1m pipeline expansion applied at the PLET end. The ANSYS model was run and reaction forces at both the ends were extracted.

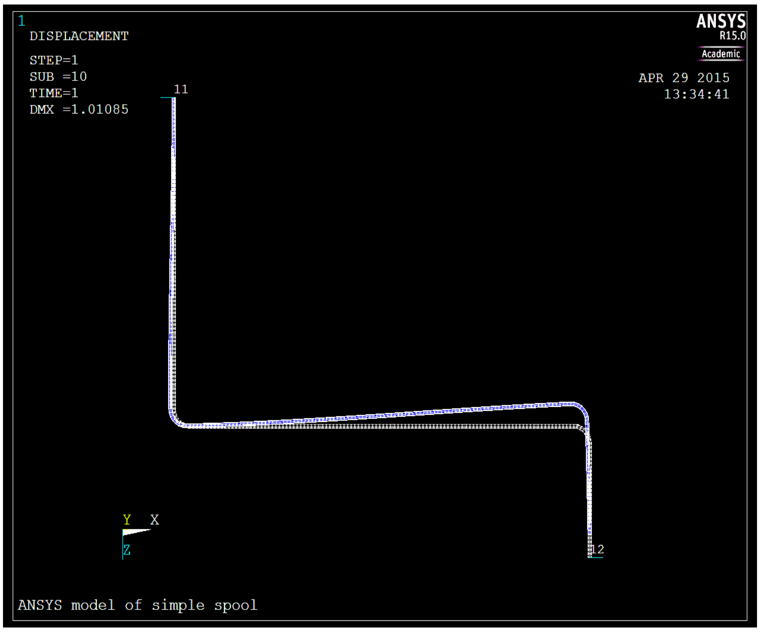


Figure 37: Simplified ANSYS model

7.1.2 ANALYTICAL APPROACH - RIGID FRAME

For many applications analytical methods provide a simple solution and good alternative to the often elaborate finite element programs. Roark's formulas provide simple and accurate formulas for stress analysis of a large range of structural components (Young & Budynas, 2002).

The approach is based on a number of assumptions. The most important to note for the purpose of this thesis are as follows:

- Liner elastic material behavior
- Small deformations –second order effects are ignored
- No bending at the corners – right angles remain right
- The system lies in one plane

Figure 40 show the frame used to simulate the spool response. The leg at the PLET end was reversed in order to get the shape of an L spool. The same pipe cross sections and material properties as in the simplified ANSYS model were assigned to the frame.

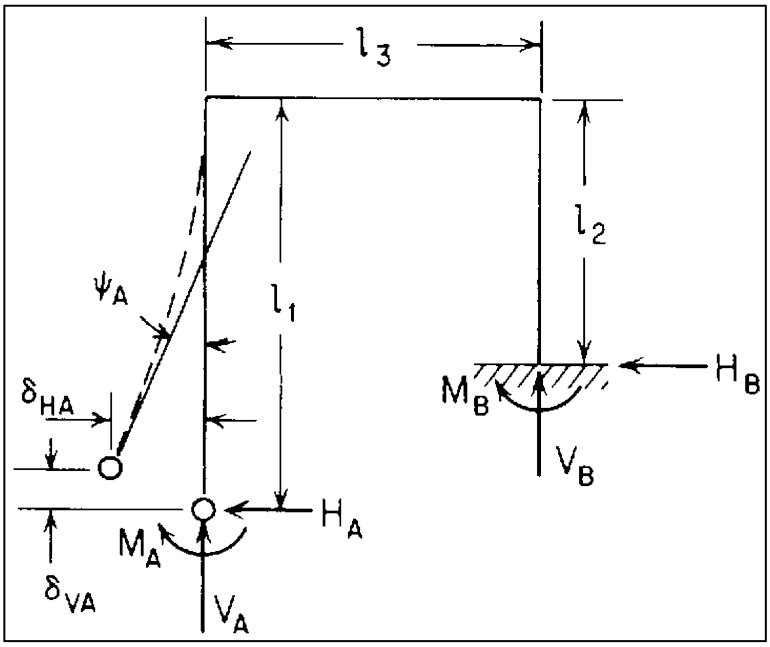


Figure 38: Rigid frame for analytical solution of spool (Young & Budynas, 2002)

The formula for vertical deflection at A is given by:

$$\delta_{VA} = C_{VH}H_A + C_{VV}V_A + C_{VM}M_A - LF_V \tag{7.1}$$

where

δ_{VA}	vertical deflection at A
C_{ij}	frame constants
H_A	horizontal force at A
M_A	moment at A
LF_V	vertical loading function
E	Young's Modulus
I	Second moment of area of pipe

The reaction forces at both the PLET and manifold end were extracted using both approaches. The results are presented in Table 13.

Table 13: End reactions for validation of model

End		ANSYS model	Rigid frame
Manifold	FX	0.804	0.810
	FY	0.69×10^{-33}	0
	FZ	-0.22×10^{-4}	0
	MX	0.50×10^{-20}	0
	MY	-28.15	-28.37
	MZ	0.165×10^{-11}	0
	$\sqrt{{M_y}^2 + {M_z}^2}$	-28.15	-28.37
PLET	FX	0.804	0.810
	FY	0	0
	FZ	0.49×10^{-13}	0
	MX	0	0
	MY	0	0
	MZ	0	0
	$\sqrt{{M_y}^2 + {M_z}^2}$	0	0

As the PLET end deforms, it will impose bending- and axial forces on the Manifold end hubs. We can see that the ANSYS model gives slightly lower moment (MX) and reaction force (FX) values. The FE model will achieve lower stress and load values due to the fact that the program takes into account stress redistribution as the structure deforms (Chan, Mylonas, & McKinnon, 2008). The rigid frame does not account for deformation at the spool bends, which means that the moments are transferred completely between the spool legs.

Based on the results from the two analyses, we can conclude that the FE model yields good results when seabed interaction is not included and the second order effects are kept to a minimum. For spools with different geometries however, these effects are often decisive for the results and cannot be ignored. The 45 degree approach angle for the spool design will have a force component at an angle to the support, which for large displacements will cause secondary load effects to occur, resulting in the displacement is no longer proportional to the spool stiffness.

The comparison is very simple and the analytical approach does not capture geometric nonlinearities, second order effects and seedbed interaction. The comparison does however give a degree of confidence to the fact that the ANSYS code made is of good quality and that it can produce accurate results

7.2 RESULTS FROM MAIN ANALYSIS

The following section presents the results from the analysis of the three spools. The governing factor for the spools was found to be the hub moment capacity. The pipe elements were modeled with nominal wall thickness and effects of wall thinning were not implemented in the analysis. The resulting end reaction forces are presented in tables 16-18. Each spool end is assigned a local coordinate system and all reaction forces are given in the local coordinate systems as shown in Figure 41.

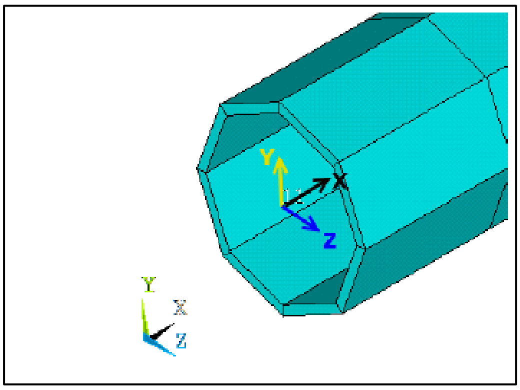


Figure 39: Local coordinate system at spool ends

7.2.1 TOLERANCES

The metrology and fabrication tolerances found in Table 9 were used in the analysis. In order to access the effect of the tolerances all different combination of both the angular and translational tolerances needs to be assessed, however, if is advisable to only consider the combinations which amount to the most extreme cases.

Several combinations of tolerances were analysed for each spool. By applying the maximum tolerances in both the longitudinal and lateral directions of the spool, compressive forces are applied to the spool, and thus adding to the compression applied by the pipe expansion during operation. By combining this with the rotational tolerances, which amounts to the larges compression of the spool, the worst case combination of tolerances is achieved.

The maximum linear tolerances are applied in the X and Z directions, whilst applying a negative rotation about the Y- axis at each hub, thus applying the maximum compressive forces to the spool. In addition, the maximum linear tolerance for the Y direction, along with tilting the hubs upwards (ROTZ) is applied. The directions of the rotations which give the worst combination will depend on the geometry of the spool and need to be checked for each case.

The combinations of tolerances representing the worst combination for all three spools are given in Table 14. The maximum tolerances in the x- and z- directions are applied at the PLET end whilst keeping the manifold end fixed.

Table 14: Combination of tolerances used for design

	Manifold					**PLET**				
DOF	**DX [mm]**	**DY [mm]**	**DZ [mm]**	**ROTZ [deg]**	**ROTY [deg]**	**DX [mm]**	**DY [mm]**	**DZ [mm]**	**ROTZ [deg]**	**ROTY [deg]**
Total	0	60	0	0.7	-0.7	110	60	110	0.7	-0.7

7.3.2 LEG LENGTHS

Table 15 presents the final leg lengths for the three spools. The largest spool was the 4-legged spool and the smallest overall spool length was found for the 5-legged spool.

Table 15: Minimum spool leg lengths and angles

Spool leg lengths and angles										
SPOOL	**A [mm]**	**B [mm]**	**C [mm]**	**D [mm]**	**E [mm]**	**V1 [deg]**	**V2 [deg]**	**V3 [deg]**	**V4 [deg]**	**TOTAL [mm]**
3- legged	28600	28600	12000	-	-	90	135	-	-	69200
4-legged	16600	34600	18000	12000	-	90	90	135	-	81200
5-legged	9100	10400	11000	20025	12000	120	90	90	90	62530

7.3.3 RESULTS FOR 3-LEGGED SPOOL

The reaction loads on the hub ends under the applied loadings and displacements for the 3-legged spool are presented in Table 16 along with the associated hub capacities. The governing factor for the design was found to be the moment capacity at the manifold end at the design stage.

The moment and axial force profiles for the spool is presented in Figures 42-44. See *Appendix D* for more detailed information.

Table 16: End reactions and associated hub capacities for 3-legged spool

	End reaction forces for 5-legged spool							
End		**Alignment**	**Tie-in**	**Pressure test**	**Operation**	**Design**	**Shut-down**	**Hub Capacity**
Manifold	FX	-19	16	13	10	22	-2	±100
	FY	22	24	24	24	23	25	±100
	FZ	0	3	4	13	13	4	±30
	MX	0	-11	-13	-37	-37	-12	±50
	MY	2	-63	-70	-211	-211	-75	-
	MZ	69	204	198	191	211	170	-
	$\sqrt{M_y^2 + M_z^2}$	**69**	**214**	**210**	**284**	**299**	**186**	**±300**
PLET	FX	-5	15	17	32	33	-14	±100
	FY	12	17	16	14	14	16	±100
	FZ	-2	14	15	17	18	-10	±30
	MX	6	-25	-26	-27	-28	-2	±50
	MY	13	-140	-140	-83	-87	55	-
	MZ	73	167	161	166	167	135	-
	$\sqrt{M_y^2 + M_z^2}$	**74**	**217**	**214**	**186**	**189**	**146**	**±300**

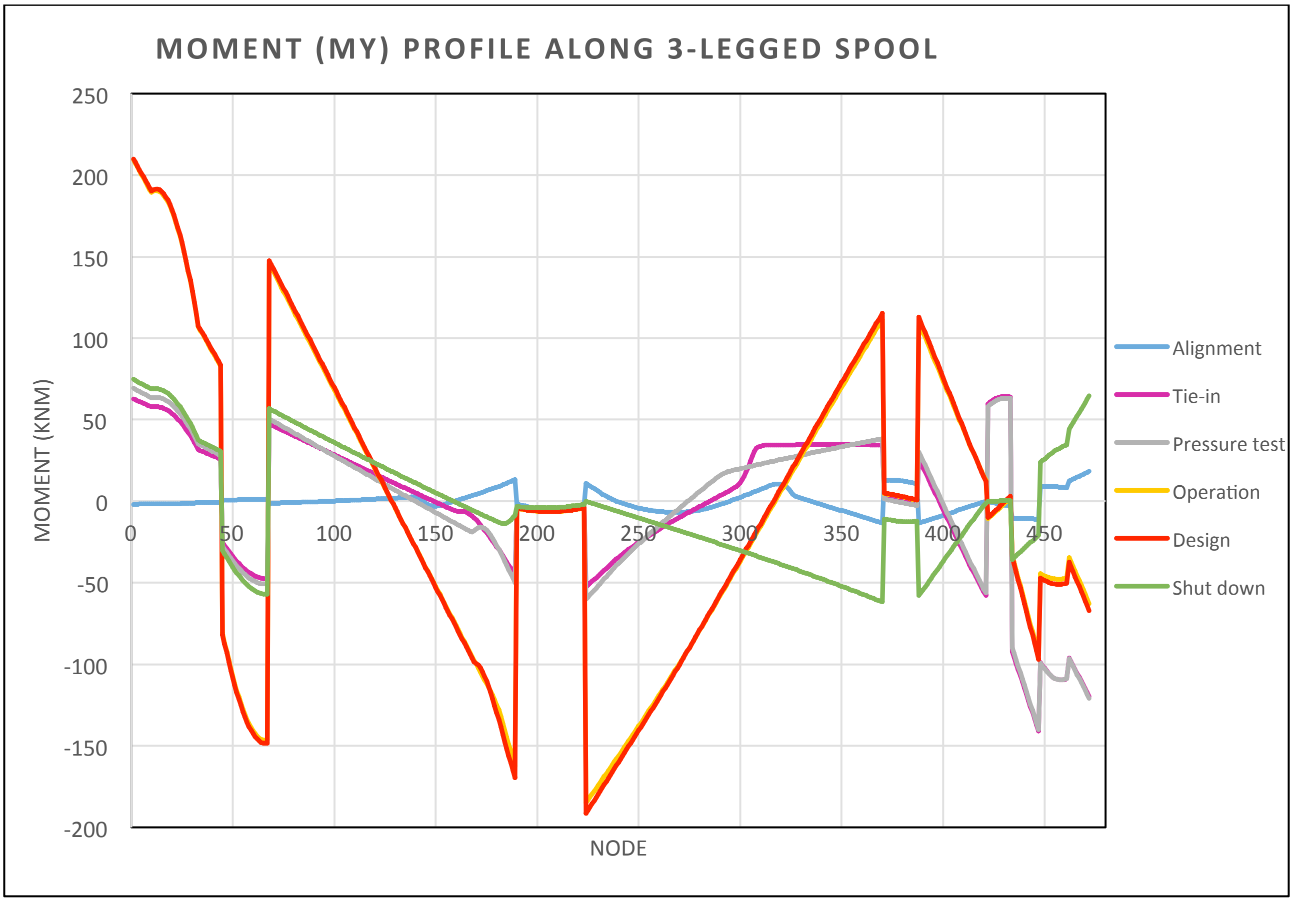


Figure 40: Moment (MY) profile along 3-legged spool

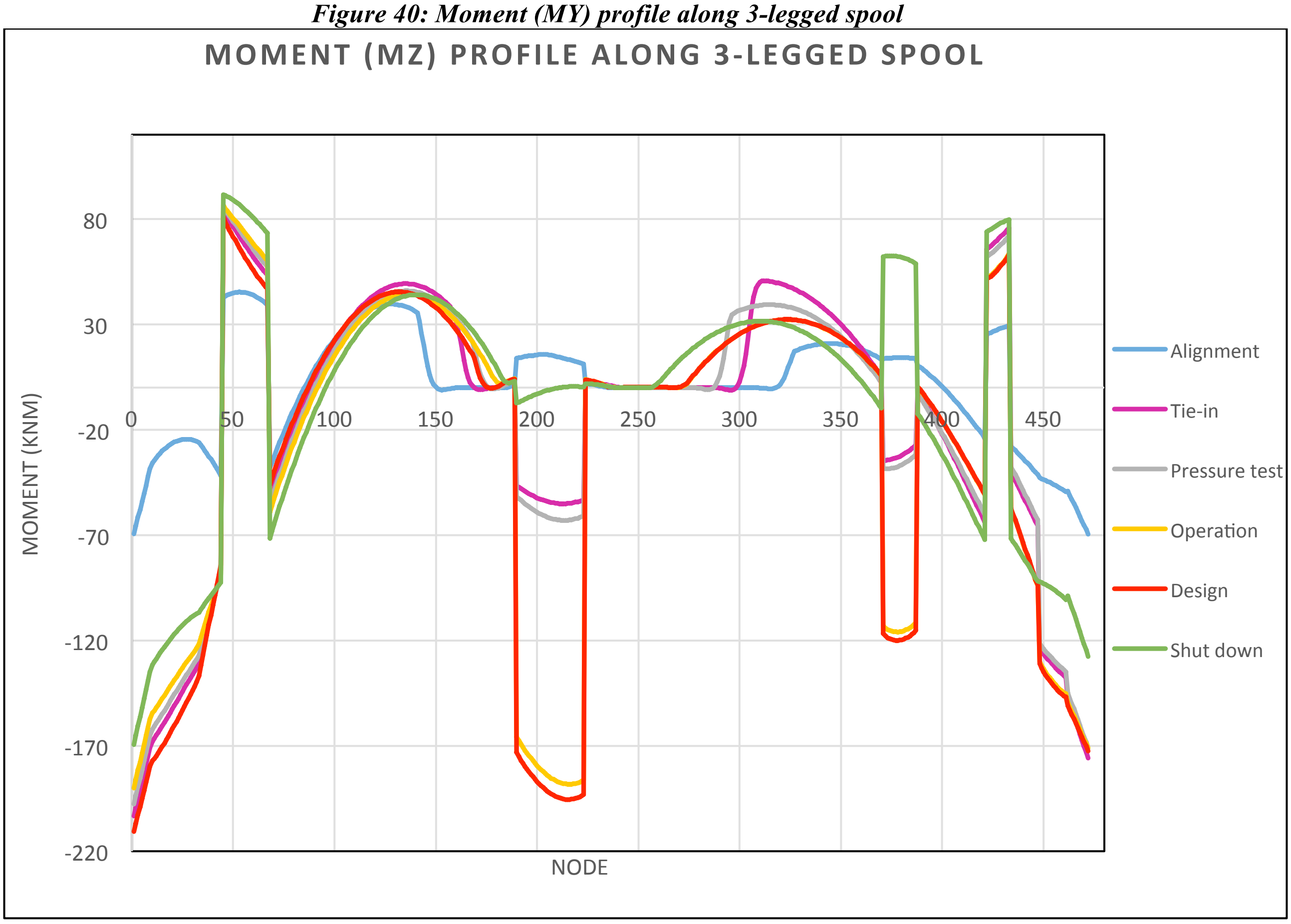


Figure 41: Moment (MZ) profile along 3-legged spool

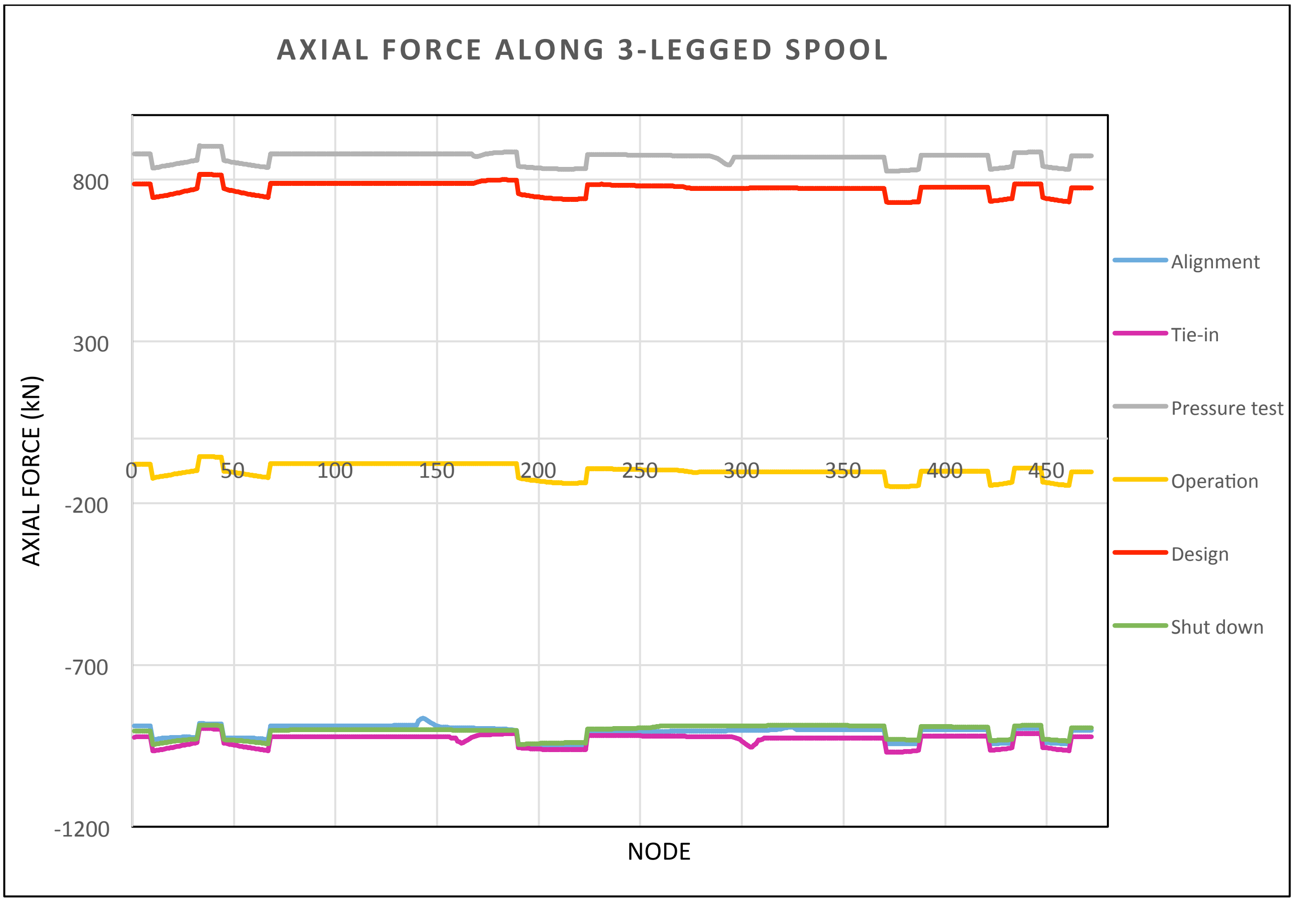


Figure 42: Axial force along 3-legged spool

7.3.3 RESULTS FOR 4-LEGGED SPOOL

The reaction loads on the hub ends under the applied loadings and displacements for the 4-legged spool are presented in Table 17. The governing factor for the design was found to be the torsional moment at the manifold end at the design stage.

The moment and axial force profiles for the spool is presented in Figures 45-47. See *Appendix D* for more detailed information.

Table 17: End reactions and associated hub capacities for 4-legged spool

	End reaction forces for 4-legged spool							
End		**Alignment**	**Tie-in**	**Pressure test**	**Operation**	**Design**	**Shut-down**	**Hub Capacity**
Manifold	FX	-7	5	6	4	13	5	±100
	FY	16	20	20	20	20	20	±100
	FZ	-1	9	10	17	18	6	±30
	MX	5	-25	-26	-46	-48	-17	±50
	MY	7	-100	-101	-150	-155	-58	-
	MZ	66	170	172	166	172	169	-
	$\sqrt{M_y^2 + M_z^2}$	67	197	199	224	232	179	**±300**
PLET	FX	-5	7	8	51	57	-17	±100
	FY	9	-4	-8	-20	-19	-3	±100
	FZ	2	9	11	-5	-5	9	±30
	MX	-2	-10	-12	3	3	-10	±50
	MY	-14	-109	-112	-97	-98	-103	-
	MZ	59	-74	-82	-100	-100	-78	-
	$\sqrt{M_y^2 + M_z^2}$	61	132	139	139	140	129	**±300**

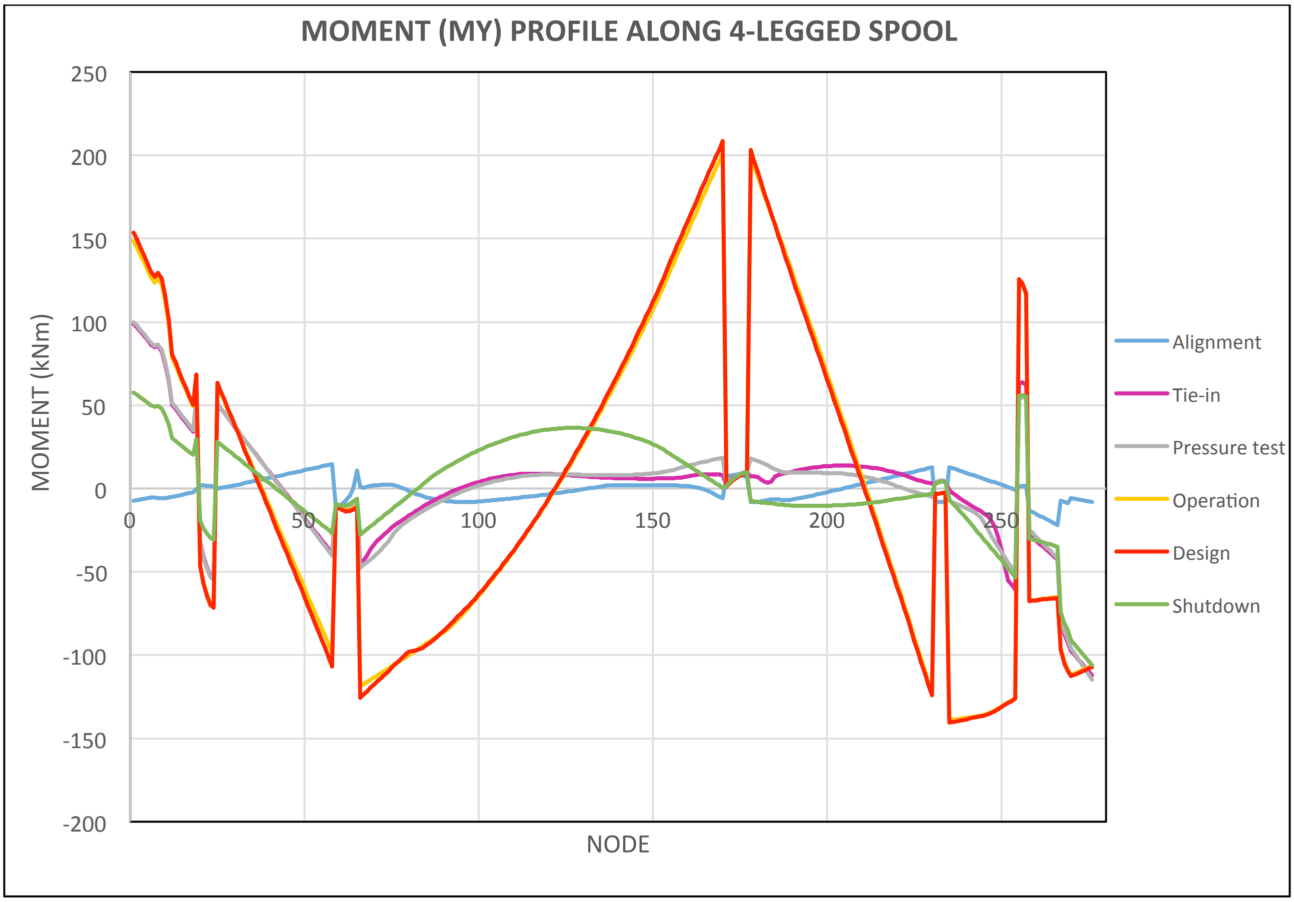


Figure 43: Moment (MY) profile along 4-legged spool

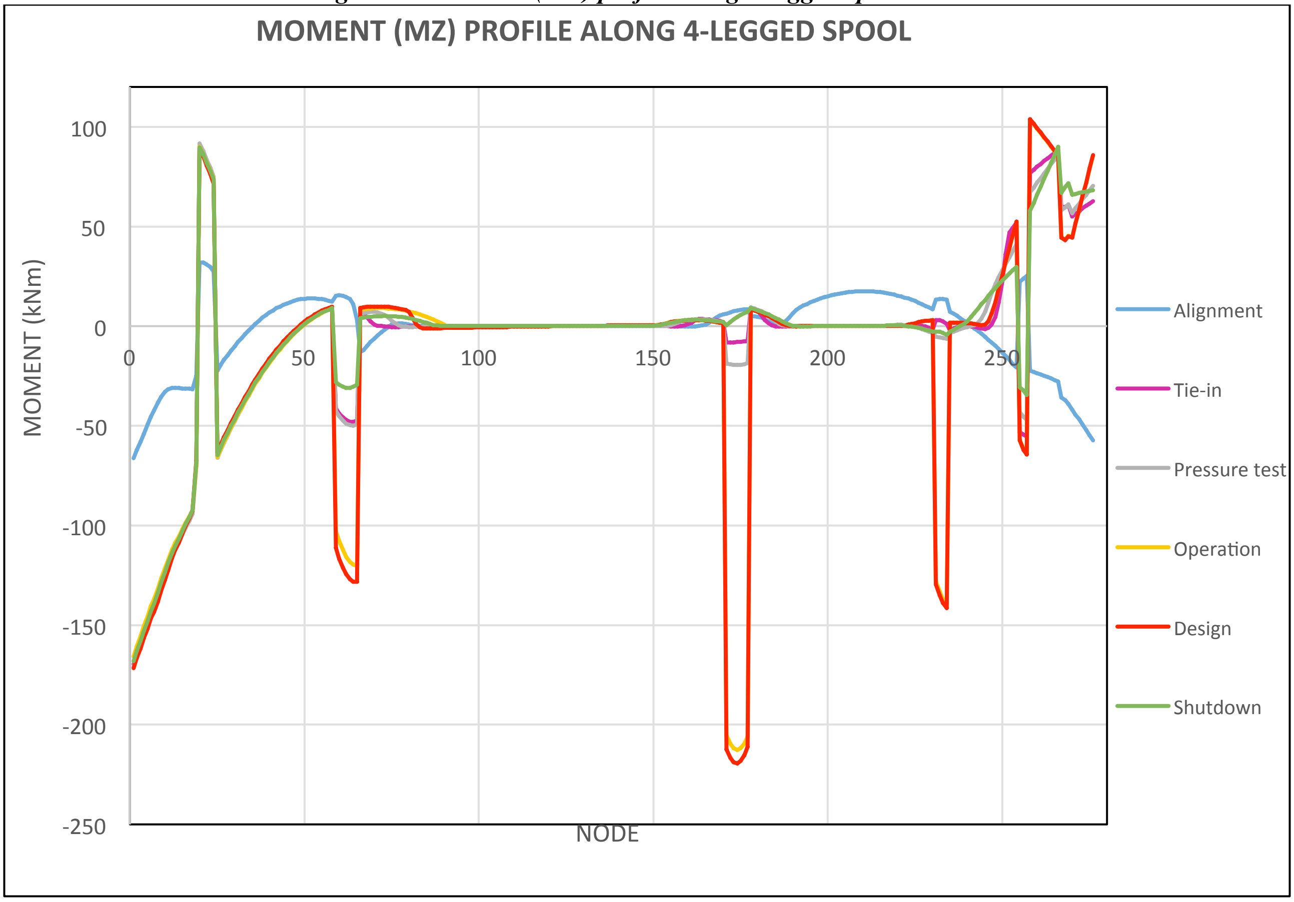


Figure 44: Moment (MZ) profile along 4-legged spool

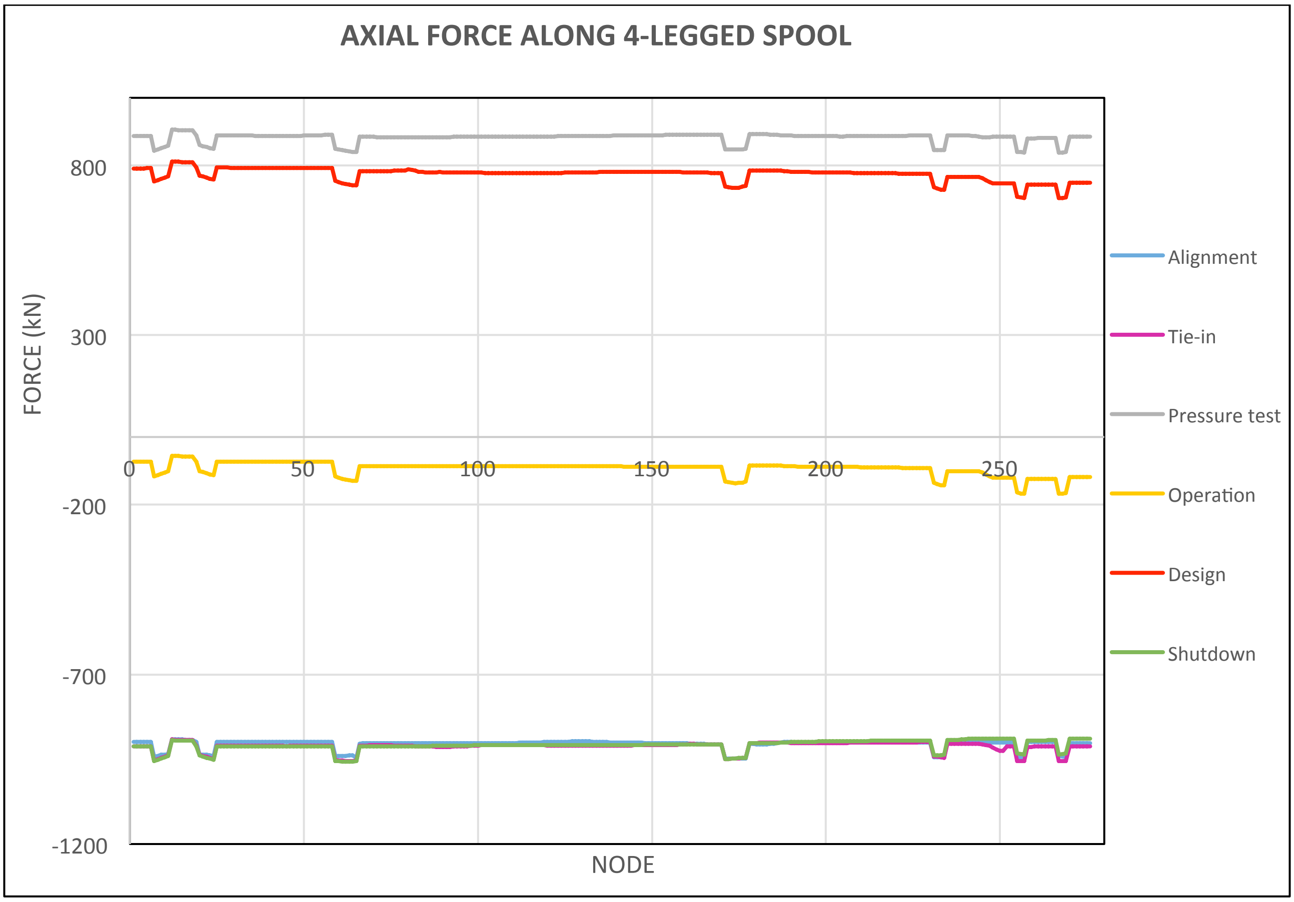


Figure 45: Axial force along 4-legged spool

7.3.4 RESULTS FOR 5-LEGGED SPOOL

The reaction loads on the hub ends under the applied loadings and displacements for the 5-legged spool are presented in Table 18.

The governing factor for the design was found to be the hub moment at the manifold end at the design stage. The moment and axial force profiles for the spool is presented in Figures 48-50. See *Appendix D* for more detailed information.

Table 18: End reactions and associated hub capacities for 5-legged spool

	End reaction forces for 5-legged spool							
End		**Alignment**	**Tie-in**	**Pressure test**	**Operation**	**Design**	**Shut-down**	**Hub Capacity**
Manifold	FX	-11	-3	2	-4	-1	8	±100
	FY	18	23	22	23	23	21	±100
	FZ	7	9	6	21	20	4	±30
	MX	-8	5	12	-24	-23	14	±50
	MY	-28	-95	-83	-241	-243	-98	-
	MZ	67	150	158	154	158	167	-
	$\sqrt{M_y^2 + M_z^2}$	**72**	**178**	**178**	**286**	**290**	**194**	**±300**
PLET	FX	-2	27	23	47	48	-16	±100
	FY	9	-10	-11	-18	-19	-3	±100
	FZ	-1	22	24	21	21	-8	±30
	MX	-2	-23	-26	-20	-20	8	±50
	MY	8	-181	-186	-91	-90	18	-
	MZ	65	-83	-87	-98	-99	-78	-
	$\sqrt{M_y^2 + M_z^2}$	**66**	**199**	**205**	**134**	**134**	**80**	**±300**

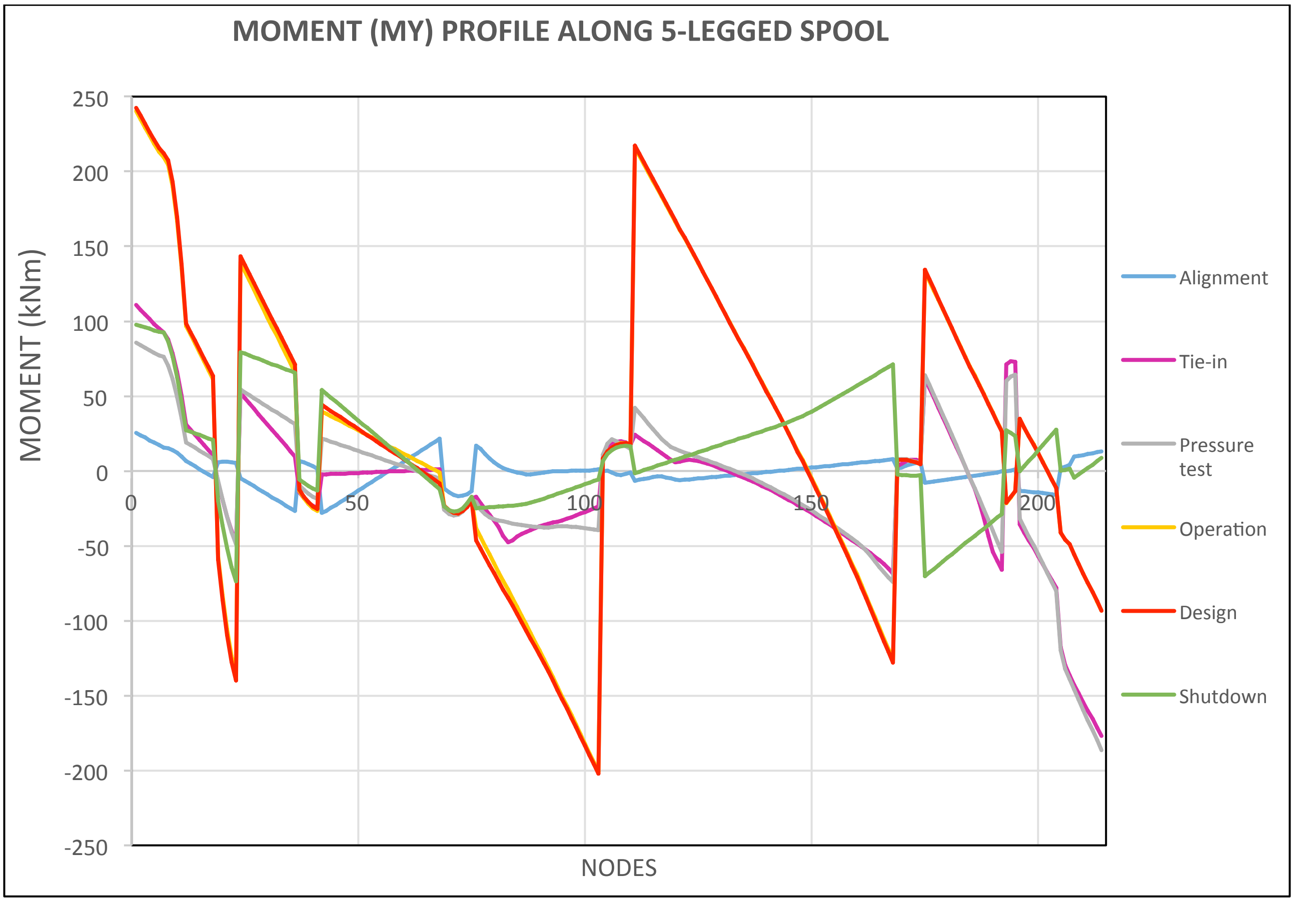


Figure 46: Moment (MY) profile along 5-legged spool

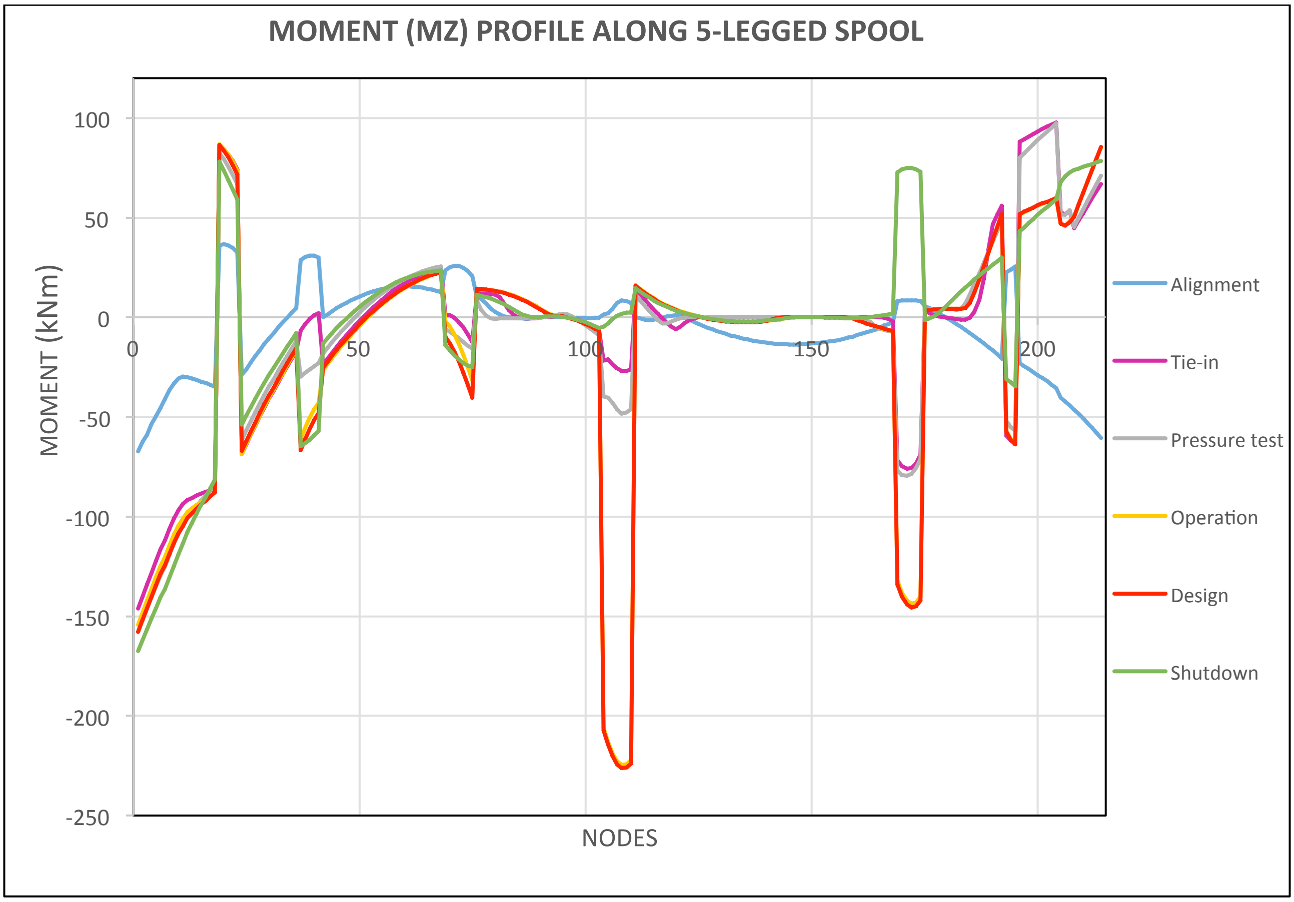


Figure 47: Moment (MZ) profile along 5-legged spool

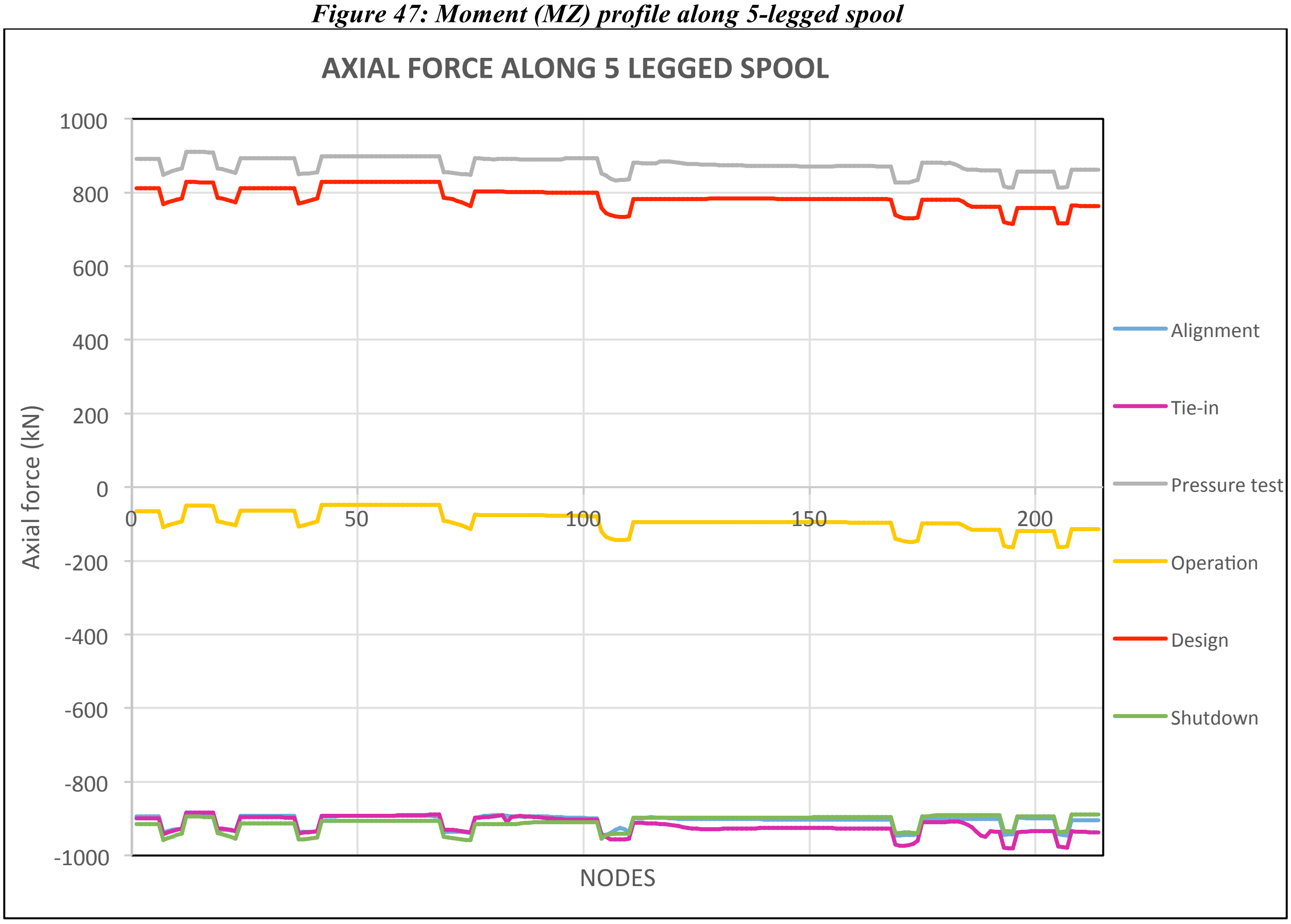


Figure 48: Axial force along 5-legged spool

7.3.5 CODE CHECKS

The spool straight sections and the bends have been checked according to DNV-OS-F101. The straight sections have been checked according to the combined buckling check and the spool bends have been assessed according to the ASD check in DNV-OS-F101, F200, Section 5.

Table 19 presents the maximum allowable moment for each spool and the utilization for the straight sections based on the obtained results. Table 20 presents the utilization at the design stage for the pipe bends based on the ASD check.

The results show that the wall thickness assigned to the spools is sufficient to withstand the imposed loads. The highest utilisation of the wall thickness is achieved in the 5-legged spool and the lowest is found in the 3-legged spool.

Table 19: Code check for straight sections

	Code check for straight section – Design condition			
Spool	**Maximum design moment [kNm]**	**Maximum design axial force [kN]**	**Allowable moment [kNm]**	**Utilization**
3-legged	213	853	393	0.54
4-legged	229	892	392	0.58
5-legged	267	912	392	0.68

Table 20: Code check for bends

	Code check for bends— Design condition			
Spool	**Allowable stress [MPa]**	**Longitudinal stress [MPa]**	**Equivalent stress [MPa]**	**Utilization**
3-legged	372	262.8	229.8	0.61
4-legged	372	276.3	240.6	0.64
5-legged	372	306.2	254.4	0.71

7.4 DISCUSSION

All three spools have been designed with the minimum possible overall length while keeping the reaction forces within the limits set by the hub capacities.

The variation in moment and axial force for the three spools is presented in Figures 42-50. The graphs show a discontinuity of the moment and axial forces at the junction of the straight pipe and elbow due to the varying wall thickness for the two sections. From Figures 42-50 we can see that large moments arise at the spool bends and particularly at the gooseneck bends on either end of the spool. During tie-in, the fabrication and metrology tolerances are applied. These displacements impose stresses on the spool, particularly in the region close to the hubs, causing the large moments in the gooseneck region. The magnitude of the stresses depends on the stiffness of this region (Jacobsen, Norland, & Tharigopula, 2015). In the analysis performed, the PLET was assumed fixed to the seafloor and the connecting hubs and framework were not included in the analysis. In a real structure however, these structures will provide additional stiffness to the system and by including them in the model, the tie-in forces on the goosenecks can be reduced (Chan, Mylonas, & McKinnon, 2008).

Due to the goosenecks, all the spool displacement is applied out of plane with the main spool. For a U-shaped loop, (Pan, Rafer, & Ahmed, 1980) found that when the imposed displacements occurs in line length of the spool the bending moment varies linearly in the straight pipe section, whilst of the displacement is out of plane with the loop all components of the force and moment are nonzero and the moment varies non-linearly. In the alignment and tie-in load cases in particular, the imposed displacements caused by the added self-weight and the fabrication/metrology tolerances are mostly out of plane, thus resulting in a non-linear response. From the MY variation of the moments, however, we can see that the variation is largely linear in the straight sections in the design and operating cases when the pipeline expansion is applied in line with the pipe.

The design check confirmed that the hub capacity is limiting for the spool design and that the wall thickness is sufficient to withstand the imposed loads.

7.4.1 COMPARISON OF SIZE AND SHAPE

The spool leg lengths have been assessed in the design process. The minimum leg length is designed based on the governing factor for the spool design, which was the hub moment capacity.

We can think of a spool in terms of a rigid frame, where a beam is supported by two columns. If one of the columns experiences settlement a moment will arise in the frame. If the length of the beam is increased, the moment will be smaller (McKeehan, 1993). In other words, the longer the legs, the more flexible the spool becomes. Longer legs therefore gives lower reaction forces at the hub end and the length of the legs must this be large enough to avoid that the moments developed by pipeline expansion causes the end moments to overstress the hubs. The poorer the spool performed when exposed to the deflection, the longer the spool becomes. The length of the spool legs was adjusted until the minimum leg length is obtained based on the limiting criteria.

Figure 51 show the variation in the end moment reaction with overall spool lengths for the three spools. We can see that the moment reaction force is reduced and as the spool overall length increases.

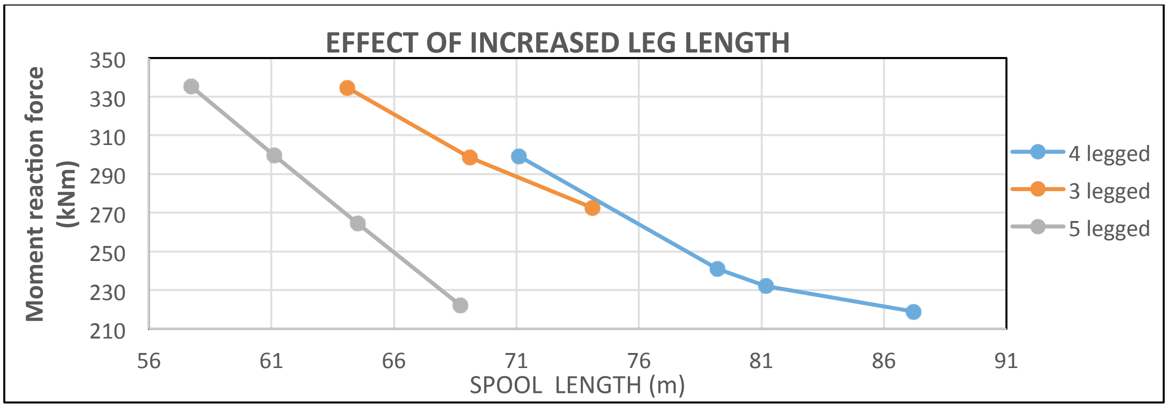


Figure 49: Variation in moment reaction force with leg length

The final spool length depends on the amount of loads that are transferred to the spool hubs. Different spool shapes all have different inherent flexibility and the end reaction forces will vary depending on the configuration. Bends and loops are added to the spools to improve the flexibility. Figure 52 shows a scaled drawing of the three spools.

The 5-legged spool was designed with the smallest overall length. The result is in line with the fact that pipe sections perpendicular to the expansion movement are most efficient at absorbing the displacement (Peng, n.d). The loop in the spool bends laterally and reduces the load being transmitted to the hub ends. The U-shape thus proves good flexibility and is well suited for absorbing the end expansion.

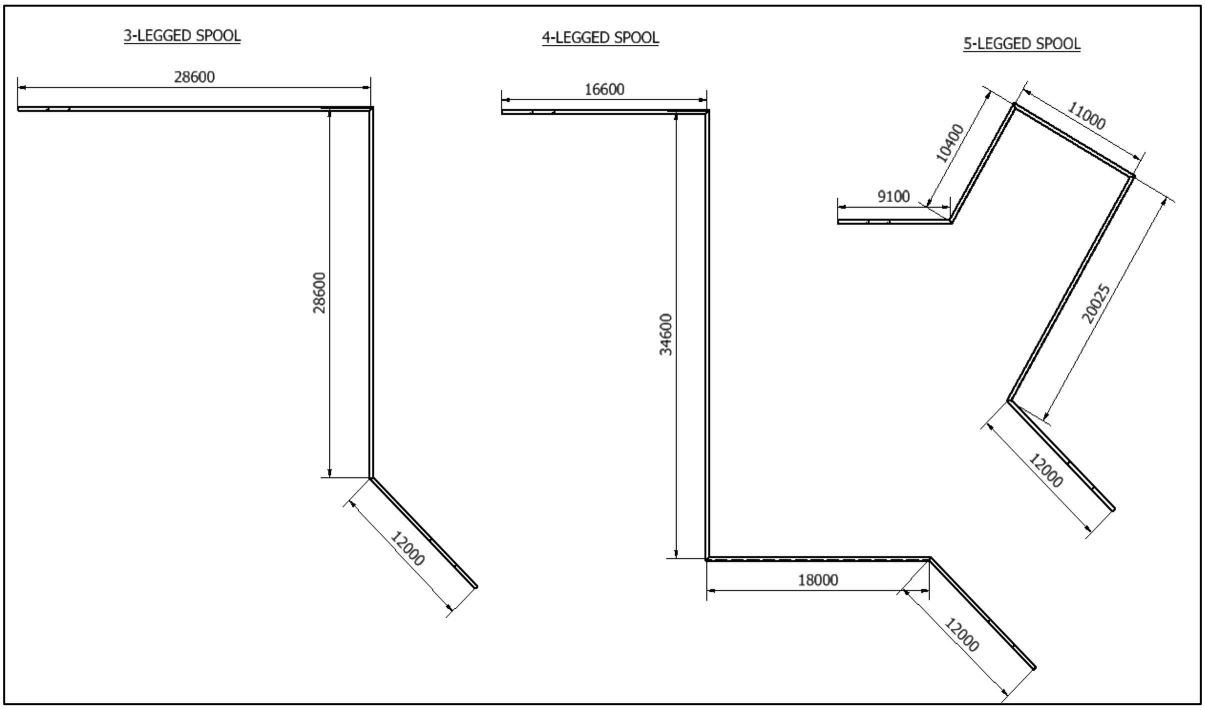


Figure 50: Scaled drawing of the three spools

The 4-legged spool needed the largest spool size to accommodate the imposed loads. One should perhaps think that the extra beds in the Z-spool would make it more flexible than the L-shaped spool (Peng, *n.d*). As seen from the analysis this may not be the case however. Increasing the number of bends from 2 in the 3-legged spool, to 3 bends in the 4-legged spool caused the spool length to increase when both were subject to the same pipe expansion. This indicates that the Z-shaped 4-legged spool does not provide the same degree of flexibility as the L- shaped spool.

When adding loops and bends in the spool, the most flexibility is obtained when the center of gravity of the spool is furthest away from the direction of the expansion movement as illustrated in Figure 53. In both the L- and U- spool configurations; the center of gravity is away from the movement vector (Peng, n.d). Even though some flexibility is achieved by adding an extra bend for the 4-legged spool this is offset by the loss of flexibility due to the fact that the bends are placed towards the center of gravity of the pipe (Peng, *n.d*).

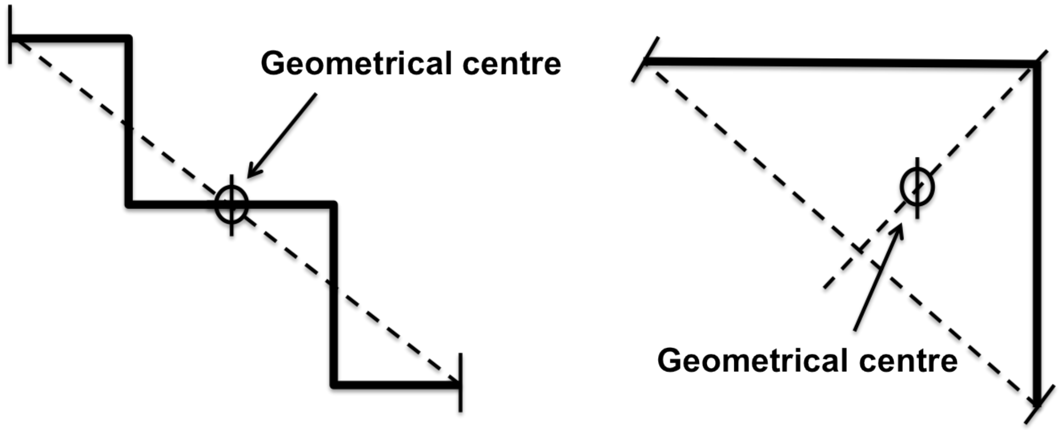


Figure 51: Bends at different distance from the geometrical centre of the spool

7.4.2 COST CONSIDERATIONS

When deciding on a spool configuration a number of factors should be considered. The size of the spool has implications for the cost. Smaller spools will provide cost saving as the need for piping material is reduced. The savings on material cost may be offset, however, if the smaller spool required more bends. Adding more bends to the spool will complicate the fabrication process. Additional welding is required and as poor welding may seriously compromise the integrity of the spool and the industry sets strict regulations to welding procedure and testing, making the welding process both expensive and time consuming (Miller, 1996). The complexity of the lifting equipment and the time required for installation will also have serious implication on the cost of the project.

7.4.3 IMPLICATIONS FOR INSTALLATION

For large subsea production systems, a number of spools will be needed to tie-in the subsea components. Transportation and installation vessels are expensive and if the number of spool that can be transported on deck is optimized then costs will be reduced. Smaller overall size may mean that more spools can fit on deck, but the shape will also be an important factor. L- and Z shapes spools may be easier to stack together if there are several spools of similar shapes and thus save deck space whilst the U-shaped spool, which is shaped more like a square may take up more deck space.

For installation, spreader bars are used to provide rigidity and support during the lift. Spools are slender and may collapse in not properly supported. A lifting analysis is usually required to define the rigging arrangement and to assess the number and locations of lift points on the spool.

Spreader bars often need to be designed specifically for each spool. The configuration of the lifting arrangement depends on the complexity of the spool shape and the overall size. Determining the centre of gravity is needed in order to determine the rigging arrangement (Sokol & Steffy, 2003). For slender spools, where the centre of gravity is along the main axis of the spool, the spreader bar system can be relatively simple, typically consisting of a single bar, with several connection points along its length. For a wide spool, a single straight bar may not be enough to support the structure and may require additional support perpendicular to the main bar as seen in Figure 16. The spreader bars also need to be transported on the transporting vessel, and large spreader arrangements will reduce the space for spools and other structures.

The advantage of the Z-shape is that the centre of gravity is more or less mid-way between the hubs, which may imply that only one spreader bar is needed along to mid-line of the spool during installations. The spool also has limited width, which is advantageous for the installation according to Chan et al (2008). The L-shape on the other hand has its centre of gravitity away from the main axis of the spool, and may also prove to be quite wide, thus requiring more carefull condideration into the spreader configuration (Chan, Mylonas, & McKinnon, 2008).

The spool shape and size also has implacations for the lifting vessel and crane. Spreader bars are massive structures and the weight of the spreader will significantly add to the weight of the lift (Sokol & Steffy, 2003). If the spools are small then the requirements for the vessel crane is reduced. However, a wide spool will increase the load on the vessel crane as the radius of the lift is increased and a larger installation vessel may be required. For a U-shaped spool the increaded flexibility means reduction in spool size which may reduce the weight of the rigging arrangement, however, if the spool is too wide, the weight reduction may be offset by lowering of crane capacity due to increased lift angle.

7.5 SENSITIVITY ANALYSIS

A finite element analysis can only be an approximation of what the real response of a structure is and error in the analysis is inevitable. Computational errors such as round-off errors from the computer calculations occur but are generally too small to be significant. Discretising errors however may cause errors that are significant (Young & Budynas, 2002). Discretizing errors occur due to limitations of the elements at representing the geometry and displacement behaviour of true structures. Some elements are better suited for certain application than other and the characteristics of the elements should be carefully investigated. If the wrong element is chosen, geometric problems such as difficulties with modelling sharp curves may arise. Elements that process constant strain characteristics will have difficulties displaying accurate strain values in regions where the strain varies greatly over a small area. (Young & Budynas, 2002)

Increasing the mesh density and thus increasing the number of elements in the finite element structure can often mitigate discretizing problems. Selecting more appropriate elements for the type of analysis will also improve the results (Young & Budynas, 2002).

The mesh density of the model depends on the selected element and the required detail level of the analysis. Sensitivity analysis should be performed to check if the chosen mesh sizing is adequate. To investigate the mesh convergence, the mesh can be refined repeatedly until the results are no longer significantly affected by the element size.

A sensitivity analysis is performed by checking the resulting deformation of the 3-legged spool as we increase the number of elements until the resulting deformation converges to a single value. This is done by investigating the deformation at the central corner for each run. The results of the analysis are presented in Figure 54.

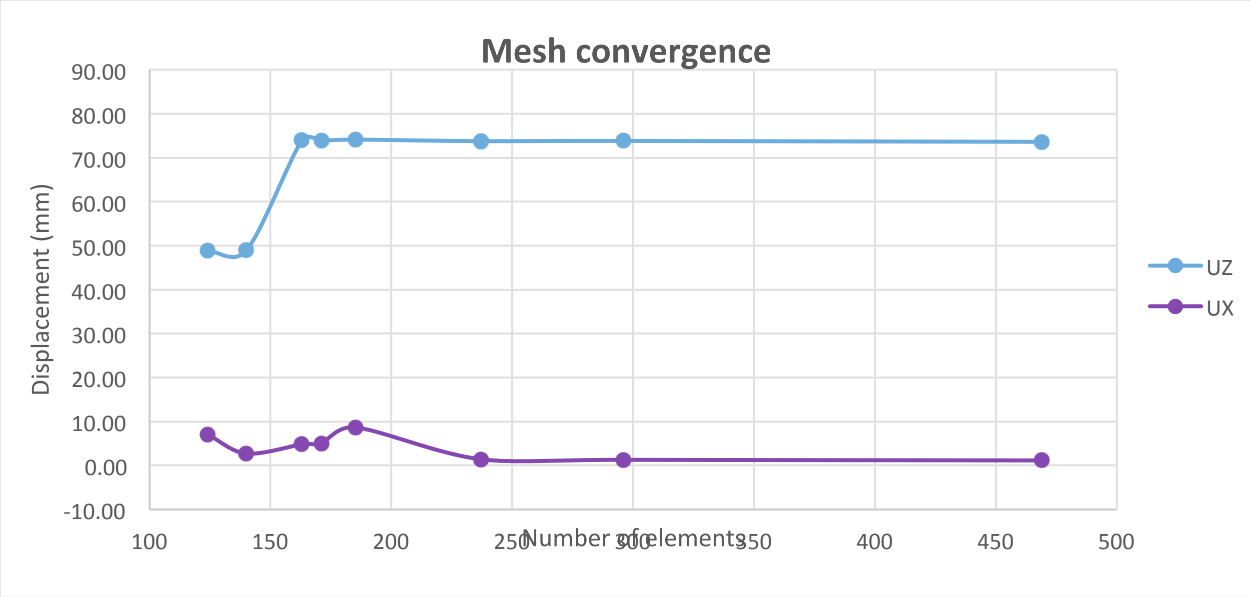


Figure 52: Mesh convergence for sensitivity analysis

We see from the analysis that the solution converges for both displacement directions after the spool has been divided into a total of 185 elements. This is equivalent of an elements size of 1.3*OD for both straights and bends. We should therefore choose an element size less than this for the analysis to be as accurate as possible.

CONCLUSION

Three rigid horizontal spools with different geometries were designed. The governing factor for the spool was found to be the allowable hub capacities and the spool size was optimized until the minimum size was obtained for each spool configuration. Table 21 provides a summary of the end reaction forces for the three spools. The values listed are the maximum values found for each reaction for both ends.

Table 21: Reaction force summary

Reaction	3-legged		4-legged		5-legged		System Capacity
	[kN]/ [kNm]	Load step	[kN]/ [kNm]	Load step	[kN]/ [kNm]	Load step	
FX	22	Design	57	Design	53	Design	±100
FY	25	Shutdown	20	Pressure test	26	Design	±100
FZ	13	Design	18	Design	24	Pressure test	±30
MX	-37	Design	-48	Design	-26	Pressure test	±50
MY	-211	Design	-155	Design	-245	Design	-
MZ	211	Design	172	Design	183	Shutdown	-
$\sqrt{M_y^2 + M}$	299	Design	232	Design	299	Design	±300

Several load conditions were tested including metrology/fabrication tolerances during tie-in and a 1m pipeline expansion applied during the operating/design load steps.

A 3-legged L-shaped spool, a 4-legged Z-shaped spool and a 5-legged U-shaped spool were investigated. The U-shaped spool was found to provide most flexibility and was thus the smallest. It was also found that the L-shaped spool provided better flexibility than the Z-

shaped spool and thus was found to be superior both in terms of flexibility but also due to the simplicity and less requirements for welding.
It was found that the number of bends is not the only factor governing the flexibility of the spool, but that the spool shape was more important. More flexibility is achieved by placing the bends away from the line of movement, thus shifting the center of gravity away from the direction of the displacement vector. Including another bend in the 4-legged spool did not improve flexibility and actually caused the spool size to increase, as the center of gravity was close to the line of movement.

RECOMMENDATIONS FOR FURTHER WORK

In order to get better understanding of the effect of the spool shape, a more complete analysis with more shapes and configurations would be recommended. The following recommendations are made in order to improve the analysis:

- Include connecting hubs, mounting structures and framework to the spool end in the analysis to capture the connecting structure stiffness contribution.
- Include a section of pipeline to allow the PLET end to slide in order to reduce tie-in loads.
- Introduce hydrostatic loads and potential loads from structure settlement as well as stroking of the spool during tie-in-

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APPENDIX A: CODE CHECKS

3-LEGGED SPOOL

ALLOWABLE STRESS DESIGN FOR BENDS

Inputs

Parameter	Value	Unit
OD	312.1	mm
Wall thickness	19.1	mm
Internal Pressure	33.7	MP
External Pressure	12.4	MP
Max Bending in operation	194	kNm
Max Effective Axial Force in operation	775	kN
Safety class	Medium	
$\boldsymbol{\eta}$	0.9	
SMYS	450	MP
De-rated yield stress	16.7	MPa

CALCULATIONS

Allowable longitudinal and equivalent stress:

Operation and design:

$$\sigma_{allowable} \leq \eta \cdot (SMYS - f_{y,temp}) \propto_u = 0.9 \times (450 - 16.7) \cdot 0.96 = 372MPa$$

Hoop stress

$$\sigma_h = (p_i - p_e)\frac{D-t}{2\cdot t}163.2\ MPa$$

Longitudinal stress

$$\sigma_{l,1} = \frac{N + p_i\frac{\pi}{4}(D-2t_2)^2 - p_e\frac{\pi}{4}D^2}{\pi\cdot(D-t_2)\cdot t_2} + \frac{M}{\frac{\pi\cdot(D^4-(D-2\cdot t_2)^4)}{32\cdot D}} = 262.8\ MPa$$

$$\sigma_{l,2} = \frac{N + p_i\frac{\pi}{4}(D-2t_2)^2 - p_e\frac{\pi}{4}D^2}{\pi\cdot(D-t_2)\cdot t_2} - \frac{M}{\frac{\pi\cdot(D^4-(D-2\cdot t_2)^4)}{32\cdot D}} = -56.8\ MPa$$

Equivalent stress **(τ_{hl} is assumed small and is ignored)**

$$\sigma_{e,1} = \sqrt{{\sigma_h}^2 + {\sigma_l}^2 - \sigma_h\cdot\sigma_l} = 229.8\ MPa$$

$$\sigma_{e,2} = \sqrt{{\sigma_h}^2 + {\sigma_l}^2 - \sigma_h\cdot\sigma_l} = 197.8\ MPa$$

Max stress utilization factor

$$\propto = \frac{max\left(|\sigma_{e,1}|, |\sigma_{e,2}|, |\sigma_{l,1}|, |\sigma_{l,1}|\right)}{f_y} = 0.61$$

COMBINED LOADING CHECK - STRAIGHT

Inputs

Parameter		**Value**	**Unit**
Outer diameter	OD	304.9	mm
Wall thickness used in code check	t	16.1	mm
Internal Pressure	p_i	337.7	bar

External Pressure	p_e	1240	bar
Max Bending in operation	M_F	194	kNm
Max Effective Axial Force in operation	S_F	775	kN
Safety class		Medium	
Yield stress	**SMYS**	450	MPa
De-rated yield stress	$f_{y,temp}$	16.7	MPa
INTERMEDIATE RESULTS			
Design yield stress	f_y	412.8	MPa
Plastic axial force resistance	S_p	6029.9	kN
Plastic moment resistance	M_p	554.3	kNm
Burst pressure	P_b	531.5	bar
	β	0.456	
Flow stress parameter	$\propto_c$	1.09	
Pressure factor	$\propto_p$	0.544	
Functional load factor	γ_F	1.1	
Condition load effect factor	γ_c	1.0	
Material resistance factor	γ_m	1.15	
Safety class resistance factor	γ_{SC}	1.14	

CALCULATIONS

Design moment $M_{Sd} = M_F \cdot \gamma_F \cdot \gamma_c = 213.4\ kNm$

Design axial force $S_{Sd} = S_F \cdot \gamma_F \cdot \gamma_c = 852.5$ kN

Maximum allowable moment:

$$M_{,max} = \left[\frac{\alpha_c}{\gamma_m \cdot \gamma_{SC}} \cdot \sqrt{1 - \propto_p \left(\frac{p_i - p_e}{\alpha_c}\right)^2} - \left(\frac{\gamma_m \cdot \gamma_{SC} \cdot S_{Sd}}{\alpha_c \cdot S_p}\right)^2\right] M_p \cdot \frac{1}{\gamma_F \cdot \gamma_C} = 392.9\ kNm$$

4-LEGGED SPOOL

ALLOWABLE STRESS DESIGN FOR BENDS

Inputs

Parameter	Value	Unit
OD	312.1	mm
Wall thickness	19.1	mm
Internal Pressure	33.7	MPa
External Pressure	12.4	MPa
Max Bending in operation	208	kNm
Max Effective Axial Force in operation	811	kN
Safety class	Medium	
$\boldsymbol{\eta}$	0.9	
SMYS	450	MPa
De-rated yield stress	16.7	MPa

CALCULATIONS

Allowable longitudinal and equivalent stress:

Operation and design:

$$\sigma_{allowable} \leq \eta \cdot (SMYS - f_{y,temp}) \propto_u = 0.9 \times (450 - 16.7) \cdot 0.96 = 372 MPa$$

Hoop stress

$$\sigma_h = (p_i - p_e) \frac{D - t}{2 \cdot t} = 163.2\ MPa$$

Longitudinal stress

$$\sigma_{l,1} = \frac{N + p_i \frac{\pi}{4}(D - 2t_2)^2 - p_e \frac{\pi}{4} D^2}{\pi \cdot (D - t_2) \cdot t_2} + \frac{M}{\frac{\pi \cdot (D^4 - (D - 2 \cdot t_2)^4)}{32 \cdot D}} = 276.3\ MPa$$

$$\sigma_{l,2} = \frac{N + p_i \frac{\pi}{4}(D - 2t_2)^2 - p_e \frac{\pi}{4} D^2}{\pi \cdot (D - t_2) \cdot t_2} - \frac{M}{\frac{\pi \cdot (D^4 - (D - 2 \cdot t_2)^4)}{32 \cdot D}} = -66.3\ MPa$$

Equivalent stress **(τ_{hl} is assumed small and is ignored)**

$$\sigma_{e,1} = \sqrt{{\sigma_h}^2 + {\sigma_l}^2 - \sigma_h \cdot \sigma_l} = 240.6\ MPa$$

$$\sigma_{e,2} = \sqrt{{\sigma_h}^2 + {\sigma_l}^2 - \sigma_h \cdot \sigma_l} = 204.5\ MPa$$

Max stress utilization factor

$$\propto = \frac{max(|\sigma_{e,1}|, |\sigma_{e,2}|, |\sigma_{l,1}|, |\sigma_{l,1}|)}{f_y} = 0.64$$

COMBINED LOADING CHECK - STRAIGHT

Inputs

Parameter		**Value**	**Unit**
Outer diameter	OD	304.9	mm
Wall thickness used in code check	t	16.1	mm
Internal Pressure	p_i	337.7	bar
External Pressure	p_e	1240	bar
Max Bending in operation	M_F	208	kNm
Max Effective Axial Force in operation	S_F	811	kN
Safety class		Medium	

Yield stress	**SMYS**	450	MPa
De-rated yield stress	$f_{y,temp}$	16.7	MPa
INTERMEDIATE RESULTS			
Design yield stress	f_y	412.8	MPa
Plastic axial force resistance	S_p	6029.9	kN
Plastic moment resistance	M_p	554.3	kNm
Burst pressure	P_b	531.5	bar
	β	0.456	
Flow stress parameter	$\propto_c$	1.09	
Pressure factor	$\propto_p$	0.544	
Functional load factor	γ_F	1.1	
Condition load effect factor	γ_c	1.0	
Material resistance factor	γ_m	1.15	
Safety class resistance factor	γ_{SC}	1.14	

CALCULATIONS

Design moment $M_{Sd} = M_F \cdot \gamma_F \cdot \gamma_c = 228.8\ kNm$

Design axial force $S_{Sd} = S_F \cdot \gamma_F \cdot \gamma_c = 892.1\ kN$

Maximum allowable moment:

$$M_{,max} = \left[\frac{\alpha_c}{\gamma_m \cdot \gamma_{SC}} \cdot \sqrt{1 - \propto_p \left(\frac{p_i - p_e}{\alpha_c}\right)^2} - \left(\frac{\gamma_m \cdot \gamma_{SC} \cdot S_{Sd}}{\alpha_c \cdot S_p}\right)^2\right] M_p \cdot \frac{1}{\gamma_F \cdot \gamma_C} = 391.5\ kNm$$

5-LEGGED SPOOL

ALLOWABLE STRESS DESIGN FOR BENDS

Inputs

Parameter	Value	Unit
OD	312.1	mm
Wall thickness	19.1	mm
Internal Pressure	33.7	MPa
External Pressure	12.4	MPa
Max Bending in operation	267	kNm
Max Effective Axial Force in operation	912	kN
Safety class	Medium	
$\boldsymbol{\eta}$	0.9	
SMYS	450	MPa
De-rated yield stress	16.7	MPa

CALCULATIONS

Allowable longitudinal and equivalent stress:

Operation and design:

$$\sigma_{allowable} \leq \eta \cdot (SMYS - f_{y,temp}) \propto_u = 0.9 \times (450 - 16.7) \cdot 0.96 = 372MPa$$

Hoop stress

$$\sigma_h = (p_i - p_e)\frac{D - t}{2 \cdot t} = 163.2\ MPa$$

Longitudinal stress

$$\sigma_{l,1} = \frac{N + p_i \frac{\pi}{4}(D - 2t_2)^2 - p_e \frac{\pi}{4} D^2}{\pi \cdot (D - t_2) \cdot t_2} + \frac{M}{\frac{\pi \cdot (D^4 - (D - 2 \cdot t_2)^4)}{32 \cdot D}} = 306.2\ MPa$$

$$\sigma_{l,2} = \frac{N + p_i \frac{\pi}{4}(D - 2t_2)^2 - p_e \frac{\pi}{4} D^2}{\pi \cdot (D - t_2) \cdot t_2} - \frac{M}{\frac{\pi \cdot (D^4 - (D - 2 \cdot t_2)^4)}{32 \cdot D}} = 94.1\ MPa$$

Equivalent stress **(τ_{hl} is assumed small and is ignored)**

$$\sigma_{e,1} = \sqrt{\sigma_h{}^2 + \sigma_l{}^2 - \sigma_h \cdot \sigma_l} = 265.4\ MPa$$

$$\sigma_{e,2} = \sqrt{\sigma_h{}^2 + \sigma_l{}^2 - \sigma_h \cdot \sigma_l} = 225.5\ MPa$$

Max stress utilization factor

$$\propto = \frac{max(|\sigma_{e,1}|, |\sigma_{e,2}|, |\sigma_{l,1}|, |\sigma_{l,1}|)}{f_y} = 0.71$$

COMBINED LOADING CHECK - STRAIGHT

Inputs

Parameter		**Value**	**Unit**
Outer diameter	OD	304.9	mm
Wall thickness used in code check	t	16.1	mm
Internal Pressure	p_i	337.7	bar
External Pressure	p_e	1240	bar
Max Bending in operation	M_F	243	kNm
Max Effective Axial Force in operation	S_F	829	kN
Safety class		Medium	

Yield stress	**SMYS**	450	MPa
De-rated yield stress	$f_{y,temp}$	20	MPa
INTERMEDIATE RESULTS			
Design yield stress	f_y	412.8	MPa
Plastic axial force resistance	S_p	6029.9	kN
Plastic moment resistance	M_p	554.3	kNm
Burst pressure	P_b	531.5	bar
	β	0.456	
Flow stress parameter	$\propto_c$	1.09	
Pressure factor	$\propto_p$	0.544	
Functional load factor	γ_F	1.1	
Condition load effect factor	γ_c	1.0	
Material resistance factor	γ_m	1.15	
Safety class resistance factor	γ_{SC}	1.14	

CALCULATIONS

Design moment $M_{Sd} = M_F \cdot \gamma_F \cdot \gamma_c$ =267 kNm

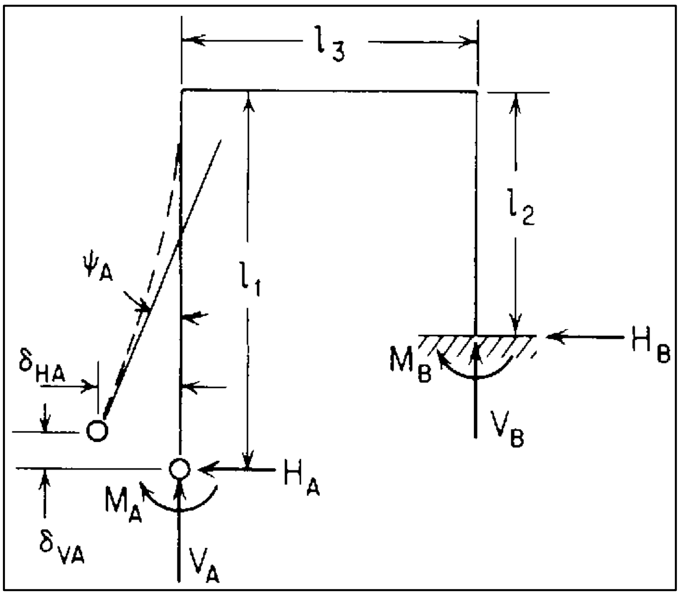
Design axial force $S_{Sd} = S_F \cdot \gamma_F \cdot \gamma_c$ =912 kN

Maximum allowable moment:

$$M_{,max} = \left[\frac{\alpha_c}{\gamma_m \cdot \gamma_{SC}} \cdot \sqrt{1 - \propto_p \left(\frac{p_i - p_e}{\alpha_c} \right)^2 - \left(\frac{\gamma_m \cdot \gamma_{SC} \cdot S_{Sd}}{\alpha_c \cdot S_p} \right)^2} \right] M_p \cdot \frac{1}{\gamma_F \cdot \gamma_C} = 391.8\ kNm$$

APPENDIX B: VALIDATION OF MODEL

RIGID FRAME



The formula for vertical deflection at A is given by

$$\delta_{VA} = C_{VH}H_A + C_{VV}V_A + C_{VM}M_A - LF_V$$

Since H_A=0 and $M_A = 0$ and we get $LF_V = 0$

$$\delta_{VA} = C_{VV}V_A$$

$$V_A = \frac{\delta_{VA}}{C_{VV}}$$

Input	
L1=-10m	$E_1 = E_2 = E_3 = 207GPa$

L2=25m L3=35m $D_o = 0.3049m$ $D_i = 0.2667m$	$I_1 = I_2 = I_3 = \left(\frac{\pi}{64}\right)\left(D_o{}^4 - D_i{}^4\right)$ $= 1.759\times10^{-4}m^4$ $E_1 I_1 = -E_2 I_2 = -E_3 I_3$

Constants
$C_{VV} = \left(\frac{L_2 L_3{}^2}{E_2 I_2}\right) + \left(\frac{L_3^3}{3E_3 I_3}\right) = 0.001234$

Reaction forces
For 1m pipe expansion $$\delta_{VA} = 1m$$ $$V_A = \frac{\delta_{VA}}{C_{VV}} = \frac{-1}{0.001234} = -810.5N$$ $$V_B = 810.5N$$ $$M_B = V_B \times L3 = 28369.1N = 28.27kN$$

APPENDIX C: ANSYS SCRIPT

```
!=====================================================================!
!       Title: Model script                                           !
!       Date: March 2013                                              !
!       Made by: Tonje Lyssand                                        !
!       File name:  3legged                                           !
!=====================================================================!
finish
/clear,all
/triad,lbot                     !Diplay XYZ triad in left bottom corner
/units,mks                      !Units in (m, kg, s, deg C)
!=====================================================================!

/PREP7                          !Start model creation preprocessor antype,0,new
                                !Static analysis and restart
!=====================================================================!
!       Defining parameters
!       Units are [m] [N] [kg] [s] [deg]
!=====================================================================!
pi=4*ATAN(1.0)                  !Pi
g=9.81                          !Gravitational acceleration [m/s^2]
WD=1234                         !Water depth [m]
Dwater=1026                     !Water density  [kg/m^3]
ID=0.2667                       !Inner diameter [m]
radb=5*ID                       !Bend radii
WT1=19.1e-3                     !Wall thickness straight section [m]
WT2=22.7e-3                     !Wall thickness bend [m]
OD1=ID+2*WT1                    !Outer diameter straight section [m]
OD2=ID+2*WT2                    !Outer diameter bend [m]
Tcoat1=49.4e-3                  !Thickness of external material [m]
Tcoat2=61.3e-3
!=====================================================================!
!       Element Types
!=====================================================================!
ET,1,pipe288,,,0                !Pipe element (straight section)
ET,2,elbow290,,,0               !Pipe element (bend)
ET,3,targe170,,,0               !Seabed element
ET,4,conta175,,,0                !Contact element

sectype,1,pipe                  !Define section type /element type 1 is pipe
secdata,OD1,WT1                 !Assign diameter and wall thickness
sectype,2,pipe                  !Define section type /element type 2 is pipe
secdata,OD2,WT2                 !Assign diameter and wall thickness

!=====================================================================!
```

```
!         Material Properties
!=====================================================================!
Dsteel=7850                              !Steel density   (kg/m^3)
Dcontent=50                              !Content density (kg/m^3)
DMEG=1115                                !MEG density
Dcoat1=900                               !Density of external material for straights
(kg/m^3)
Dcoat2=1040                              !Density of external material for bends (kg/m^3)


!         Submerged weight straight pipe
!-------------------------------------------------------------------------------------------------------------!
Dtot1=OD1+2*Tcoat1                            !Total diameter
Asteel1=(pi/4)*(OD1**2-ID**2)                 !Cross-sectional area of steel
Acoat1=(pi/4)*(((OD1+2*Tcoat1)**2)-(OD1**2))  !Cross-sectional area of external coating
AtotE1=(pi/4)*((Dtot1**2)-(ID**2))            !Total cross sectional area of material
Atot1=(pi/4)*Dtot1**2                         !Total cross section of filled pipe
AInner1=(pi/4)*ID**2                          !Inner cross section of pipe

Wsteel1=Asteel1*Dsteel*g                      !Steel mass (N/m)
Wcoat1=Acoat1*Dcoat1*g                        !Coating mass (N/m)
Wcontent1=AInner1*Dcontent*g                  !Content mass (N/m)
WMEG1=AInner1*DMEG*g                          !Mass of MEG (N/m)

WairE1=Wsteel1+Wcoat1                         !Mass in air empty (N/m)
WairC1=WairE1+Wcontent1                       !Mass in air content filled (N/m)
WairMEG1=WairE1+WMEG1                         !Mas in air MEG filled
Wboyancy1=Atot1*Dwater*g                      !Total buoyancy filled (content and MEG)

SubWeightE1=WairE1-Wboyancy1                  !Submerged weight empty (N/m)
SubWeightC1=WairC1-Wboyancy1                  !Submerged weight content filled (N/m)
SubWeightMEG1=WairMEG1-Wboyancy1              !Submerged weight content filled (N/m)

DENSeqvE1=SubWeightE1/(Asteel1*g)             !Equivalent density empty  (kg/m^3)!


MassContentMEG=WMEG1/g                        !Content mass MEG filled (kg/m)
MassContentC=Wcontent1/g                      !Content mass (kg/m)

!         Submerged weight pipe bend
!-------------------------------------------------------------------------------------------------------------!
Dtot2=OD2+2*Tcoat2                            !Total diameter

Asteel2=(pi/4)*(OD2**2-ID**2)                 !Cross-sectional area of steel
Acoat2=(pi/4)*(((OD2+2*Tcoat2)**2)-(OD2**2))  !Cross-sectional area of external coating
AtotE2=(pi/4)*((Dtot2**2)-(ID**2))            !Total cross sectional area of material
Atot2=(pi/4)*Dtot2**2                         !Total cross section of filled pipe
AInner2=(pi/4)*ID**2                          !Inner cross section of pipe
```

```
Wsteel2=Asteel2*Dsteel*g                  !Steel mass (N/m)
Wcoat2=Acoat2*Dcoat2*g                    !Coating mass (N/m)
Wcontent2=AInner2*Dcontent*g              !Content mass (N/m)
WMEG2=AInner2*DMEG*g                      !Mass of MEG (N/m)

WairE2=Wsteel2+Wcoat2                     !Mass in air empty (kg/m)
WairC2=WairE2+Wcontent2                   !Mass in air content filled (kg/m)
WairMEG2=WairE2+WMEG2                     !Mass in air MEG filled
Wboyancy2=Atot2*Dwater*g                  !Total buoyancy

SubWeightE2=WairE2-Wboyancy2              !Submerged weight empty
SubWeightC2=WairC2-Wboyancy2              !Submerged weight content filled
SubWeightMEG2=WairMEG2-Wboyancy2          !Submerged weight content filled

DENSeqvE2=SubWeightE2/(Asteel2*g)         !Equivalent density (kg/m^3)
!---------------------------------------------------------------------------------------------------------!

MP, EX, 1, 207e9                          !Young's Modulus for straight pipe
MP, PRXY,1, 0.3                           !Poisson's ratio for straight pipe
MP, ALPX,1, 1.17e-5                       !Secant coefficient of thermal expansion

MP, EX, 2, 207e9                          !Young's Modulus for pipe bend
MP, PRXY,2, 0.3                           !Poisson's ratio for pipe bend
MP, ALPX,2, 1.17e-5                       !Secant coefficient of thermal expansion

MP,DENS,60,DMEG                           !Assign material number to content
MP,DENS,70,Dcontent                       !Add content density (MEG filled)

        Material modelling
!---------------------------------------------------------------------------------------------------------!

TB,KINH,1,1,,4              !Data table for nonlinear material  (STRAIGHT)
TBTEMP,20.0                 !Temperature for material prop
TBPT,,0.0,0.0               !Strain=0.00,Stress=0.00
TBPT,,0.002174,450E6        !Elastic: strain = 0.0217%, Stress = 450E6 (Nm^-2)
TBPT,,0.020,450E6           !Yield Strain: strain = 2.0%, stress = 450E6 (Nm^-2)
TBPT,,0.060,535E6           !Plastic strain: strain = 6.0%, stress = 535E6 (Nm^-2)

TBLIST,KINH,1               !Lists the material data tables.
TBPLOT,KINH,1               !Plot material data

TB,KINH,2,1,4               !Data table for nonlinear material properties (BENDS)
TBTEMP,20.0
TBPT,,0.0,0.0               !Strain=0.00,Stress=0.00
TBPT,,0.002174,450E6        !Elastic: Strain = 0.0217%, Stress = 450E6 (Nm^-2)
TBPT,,0.020,450E6           !Yield Strain: strain = 2.0%, stress = 450E6 (Nm^-2)
TBPT,,0.060,535E6           !Plastic strain: strain = 6.0%, stress = 535E6 (Nm^-2)

TBLIST,KINH,2               !Lists the material data tables.
```

```
TBPLOT,KINH,2                    !Plot material data

TB,FRIC,50,,,ortho                     !Friction factor for seabed
FRICLAX=0.8                            !Soil friction in axial direction
FRICLAY=0.8                            !Soil friction in lateral direction
TBdata,1,FRICLAX,FRICLAY

!=====================================================================!
!      Operational Conditions
!=====================================================================!
OPress=150e5                           !Operating pressure (N/m^2)
DP=307e5                               !Design pressure (N/m^2)
TP=322e5                               !Test pressure (N/m^2)
Tamb=5                                 !Ambient temperature
T0=50                                  !Operating temperature
TD=78                                  !Design temp
!=====================================================================!
!      Define spool geometry
!=====================================================================!
!Define keypoints for spool
!KP1=Manifold end
!KP8=PLET end

K,   1    ,   0      ,   3.20     ,   0
K,   2    ,   2.38   ,   3.20     ,   0
K,   3    ,   4.10   ,   0        ,   0
K,   4    ,   11.60  ,   0        ,   0
K,   5    ,   15.60  ,   0        ,   0
K,   6    ,   15.60  ,   0        ,   28.60
K,   7    ,   28.60  ,   0        ,   28.60
K,   8    ,   33.43  ,   0        ,   33.43
K,   9    ,   35.40  ,   1.61     ,   35.40
K,   10   ,   37.09  ,   1.40     ,   37.09

!Define lines
L,1,2
L,2,3
L,3,4
L,4,5
L,5,6
L,6,7
L,7,8
L,8,9
L,9,10

!Define bends
LFILLT,1,2,radb,0
LFILLT,2,3,radb,0
LFILLT,4,5,radb,0
```

```
LFILLT,5,6,radb,0
LFILLT,6,7,radb,0
LFILLT,7,8,radb,0
LFILLT,8,9,radb,0

cm,pipelines,line                                   !Group lines
!=====================================================================!
!       Meshing of spool
!=====================================================================!
r,200,,                                             !Defines real constants

KEYOPT,1,1,0                                        !Temperature through wall gradient
KEYOPT,1,3,2                                        !Quadratic shape function
KEYOPT,1,4,1                                        !Thin Pipe Theory
KEYOPT,1,6,0                                        !End cap loads
KEYOPT,1,7,0                                        !Output control
KEYOPT,1,8,0
KEYOPT,1,9,2
KEYOPT,1,15,0

KEYOPT,2,1,0                                        !Temperature through wall gradient
KEYOPT,2,2,3                                        !Allow for ovalisation and c/s deformation.
KEYOPT,2,6,0                                        !End cap loads

alls
lsel,s,line,,1,9,                                   !Select straight section
mat,1                                               !Assign material type
type,1                                              !Assign element type
secnum,1
real,200                                            !Assign section properties
esize,1*OD1                                         !Assign element size
LMESH,all                                           !MESH STRAIGHTS

lsel,s,line,,10,17,                                 !Select bends
mat,2                                               !Assign material type
type,2                                              !Assign element type
secnum,2                                            !Assign section properties
real,200
esize,1*OD2                                         !Assign element size
LMESH,all                                           !MESH BENDS

!=====================================================================!
!       Seabed modelling
!=====================================================================!
FRK=0.8                                             !Normal contact stiffness factor
FTOLN=0.2                                           !Penetration tolerance factor
r,400,,,FRK,FTOLN,,,,,                              !Real constant seabed and contact elements

KEYOPT,4,2,1                                        !Penalty method
```

```
!KEYOPT,4,3,0                   !Contact model: (0)Contact Force Based
KEYOPT,4,4,2                    !Normal from contact nodes
KEYOPT,4,10,2                   !Constant stiffness update
KEYPOT,4,5,2                    !Reduce penetration with auto CNOF

!Define keypoints for seabed
K,    1001  ,    -10   ,    -0.10  ,    -10
K,    1002  ,    -10   ,    -0.10  ,    50
K,    1003  ,    50    ,    -0.10  ,    50
K,    1004  ,    50    ,    -0.10  ,    -10

A,1001,1002,1003,1004           !Define area for seabed
asel,s,area,,1                  !Select seabed area
type,3                          !Targe170
mat,50                          !Assign seabed material properties
real,400                        !Defines element real constants
esize,10000000                  !Assigns element size
AMESH,all                       !MESH SEABED

alls
lsel,s,line,,3,7                !Select lines in contact with seabed
lsel,a,line,,11,16
nsll,s,all                      !Choose elements associated with selected lines
type,4                          !Assign element type
real,400                        !Defines element real constants
ESURF                           !GENERATES CONTACT ELEMENTS

alls
esel,s,type,,1
cm,straight,elem                !Group straight section
esel,s,type,,2
cm,bend,elem                    !Group bends
esel,a,type,,1
cm,pipeelem,elem                !Group whole pipe into element name pipeelem
nsle,all
cm,pipenodes,node               !Groups all pipenodes

alls
ksel,s,kp,,1
nslk,s
cm,KP1node,node                 !Name node at KP1

alls
ksel,s,kp,,10
nslk,s
cm,KP10node,node                !Name node at KP10

!=================================================================!
!       Local Coordinate systems
```

```
!======================================================================!
!Generate nodes to define new coordinate system at nodes
K,      6001   ,        0       ,       5.500   ,       0               !Same as KP1
K,      6002   ,        37.09   ,       2.0     ,       37.09           !Same as KP10 (y=2.0)
K,      6003   ,        35.40   ,       1.4     ,       35.40           !KP9 (y=1.4)
alls
CSKP,11,CART,1,2,6001                   !Create local coordinate system at KP1
CSYS,11                                 !Local coordinates MANIFOLD end
cmsel,s,KP1node,node                    !Select KP1
NROTAT,all                              !Rotate nodal coordinate system to local system

alls
CSKP,12,CART,10,6003,6002               !Create local coordinate system at KP8
CSYS,12                                 !Local coordinates PLET end
cmsel,s,KP10node,node                   !Select KP1
NROTAT,all                              !Rotate nodal coordinate system to local system

/PSYMB,CS,1                             !Disply local coordiante systems
CSYS,0                                  !Active default coordinate system

alls
SAVE
PARSAV,ALL,Param_model,txt              !Save parameters to latbuck.txt
/ESHAPE,1
EPLOT                                   !Display element shape

FINISH
/eof


!======================================================================!
!       Title: Solution script                                         !
!       Date: March 2013                                               !
!       Made by: Tonje Lyssand                                         !
!       File name:  3legged                                            !
!======================================================================!
fini
/clear,all
/INPUT,model,txt                        !call file "model.mac" in same direct
!======================================================================!
!       Solution
!======================================================================!
/SOLU                                   !Start solution processor
NLGEOM,ON                               !Include large deformation
ANTYPE,0
AUTOTS,ON                               !Automatic time stepping
CNCHECK,AUTO                            !Adjust the initial status of contact pairs
NROPT,UNSYM                             !Stiffness matrix updated at every equilibrium
iteration
parres,change,Param_model,txt           !Reads parameters from a Parammodel.txt file
```

```
TREF,Tamb                                   !Ambient temp

!=========================================================================!
!       LOAD STEP1: Apply self weight
!=========================================================================!
TIME,1                              !Load step 1
/title,Apply self weight
NROP,UNSYM                          !Use full Newton-Raphson with unsymmetrical matrices
alls
DK,1,all,0                          !Defines constraint on key points
DK,10,all,0

!APPLY SELF WEIGHT
sfe,pipeelem,3,pres,0,ID/2          !Set free surface of internal fluid
MP,DENS,1,DENSeqvE1                 !Equivalent density for submerged weight

sectype,1,pipe                      !Chose pipe288 elements
secdata,OD1,WT1,,,,60               !Add content density (MEG filled)
ACEL,,g,ID/2

sfe,pipeelem,3,pres,0,ID/2          !Set free surface of internal fluid
MP,DENS,2,DENSeqvE2                 !Equivalent density for submerged weight!

sectype,2,pipe                      !Chose elbow290 elements
!seccontrols,DMEG
secdata,OD2,WT2,,,,60               !Add content density (MEG filled)
ACEL,,g,ID/2

!APPLY HYDROSTATIC PRESSURE
cmsel,s,pipeelem,elem                       !Select pipeelements
sfe,pipeelem,2,pres,0,Dwater*g*WD           !Hydrostatic pressure (N/m^2)

nsubst,1,1000,1
alls
lswrite,1                           !Define load step, write data to FILE=file.s01
!lssolve,1
save

!=========================================================================!
!       LOAD STEP 2: Tie-in
!=========================================================================!
TIME,2                                      !Load step 2
/title,Tie-in

CSYS,11                                     !Enter local coordinate system
deg_roty=0.7
deg_rotz=0.7
rad_roty=deg_roty*pi/180
rad_rotz=deg_rotz*pi/180
```

```
DK,1,UX,0.055                        !Apply tolerances
DK,1,UY,0.06
DK,1,UZ,0.055
DK,1,ROTY,-rad_roty
DK,1,ROTZ,rad_rotz

CSYS,12
DK,10,UX,0.055                       !Apply tolerances
DK,10,UY,-0.06
DK,10,UZ,0.055
DK,10,ROTY,-rad_roty
DK,10,ROTZ,-rad_rotz

nsubst,1,10000,1
alls
lswrite,2
!lssolve,1,2
CSYS,0                               !Back to global coordinate system

!==================================================================!
!       LOAD STEP 3: Pressure Test
!==================================================================!
TIME,3                          !Load step 3
/title,Pressure test
sfe,pipeelem,1,pres,0,TP        !Test pressure (N/m^2)

nsubst,1,10000,1
alls
lswrite,3                       !Define load step, write data to FILE=file.s01

!lssolve,1,3

!==================================================================!
!       LOAD STEP 4: Operational condition
!==================================================================!
TIME,4                               !Load step 4
/title, Operational condition

!OPRATIONAL CONTENT
Mcont=AINNER1*(DContent-DMEG)        !Add "negative" added mass to model content

nsubst,100,10000,100
MP,DENS,1,DENSeqvE1
seccontrols,Mcont

MP,DENS,2,DENSeqvE2
seccontrols,Mcont
```

```
sfe,pipeelem,1,pres,0,OPress                !Operating pressure (N/m^2)
bfe,pipeelem,temp,1,T0,Tamb                 !Operating temperature

!APPLY EXPANSION TO PLET END
CSYS,12
dcum,add                                    !Subsequent D's to be added
DK,10,UX,1.0                                !Apply expansion of 1.0m to PLET end

nsubst,100,100000,100
alls
lswrite,4                                   !Define load step, write data to FILE=file.s01

!lssolve,1,4
CSYS,0

!=====================================================================!
!       LOAD STEP 5: Design condition
!=====================================================================!
TIME,5
/title,Design condition

sfe,pipeelem,1,pres,0,DP                    !Design pressure (N/m^2)
bfe,pipeelem,temp,1,TD,Tamb                 !Design temperature

nsubst,1,10000,1
alls
lswrite,5                                   !Define load step, write data to FILE=file.s01

!lssolve,1,5

!=====================================================================!
!       LOAD STEP 6: Shutdown
!=====================================================================!
TIME,6
/title,Shutdown

sfe,pipeelem,1,pres,0,0                     !Set internal pressure to zero (N/m^2)
bfe,pipeelem,temp,1,0,Tamb                  !Set internal temperature to zero

!REMOVE EXPANSION TO PLET END
CSYS,12
DK,10,UX,0                                  !Remove expansion

alls
nsubst,100,100000,100
lswrite,6                                   !Define load step, write data to FILE=file.s01

lssolve,1,6
/eof
```

```
!==========================================================================!
!        Title: Post processor script                                      !
!        Date: March 2013                                                  !
!        Made by: Tonje Lyssand                                            !
!        File name:  3legged                                               !
!==========================================================================!
 fini
/clear,all
/INPUT,solution,txt                     !call solution file in same directory

!==========================================================================!
!        Post Processing
!==========================================================================!
/POST1
/DSCALE,1,1
!==========================================================================!
!        Self Weight
!==========================================================================!
set,1
/output,LC1,txt                         !Make file with name LC1
RSYS,11                                 !Change to local CS for manifold end
cmsel,s,KP1node,node
PRRSOL                                  !Reaction forces
RSYS,12                                 !Change to local CS for PLET end
cmsel,s,KP10node,node
PRRSOL                                  !Reaction forces
/output

alls
/output,LC1disp,txt
PRNSOL,DOF                              !Displacements
/output

alls
/output,LC1results,txt
cmsel,s,straight,elem                   !PIPE288
ETABLE,MYI,SMISC,2                      !Bending moment for nodeI
ETABLE,MYJ,SMISC,15                     !Bending moment for nodeJ
ETABLE,MZI,SMISC,3                      !Bending moment for nodeI
ETABLE,MZJ,SMISC,16                     !Bending moment for nodeJ
ETABLE,EffAxiI,SMISC,1                  !Axial force I
ETABLE,EffAxiJ,SMISC,14                 !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
cmsel,s,bend,elem                       !ELBOW290
ETABLE,MYI,SMISC,2                      !Bending moment for nodeI
ETABLE,MYJ,SMISC,37                     !Bending moment for nodeJ
ETABLE,MZI,SMISC,3                      !Bending moment for nodeI
ETABLE,MZJ,SMISC,38                     !Bending moment for nodeJ
```

```
ETABLE,EffAxiI,SMISC,1               !Axial force I
ETABLE,EffAxiJ,SMISC,36              !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
/output

!=====================================================================!
!       Tie-in
!=====================================================================!
set,2
/output,LC2,txt                      !Make file with name LC2
RSYS,11                              !Change to local CS for manifold end end
cmsel,s,KP1node,node
PRRSOL                               !Reaction forces
RSYS,12
cmsel,s,KP10node,node                !Change to local CS for PLET end
PRRSOL
/output

RSYS,0
alls
/output,LC2disp,txt
PRNSOL,DOF                           !Displacements
/output

alls
/output,LC2resutls,txt
cmsel,s,pipeelem,elem
cmsel,s,straight,elem                !PIPE288
ETABLE,MYI,SMISC,2                   !Bending moment for I
ETABLE,MYJ,SMISC,15                  !Bending moment for J
ETABLE,MZI,SMISC,3                   !Bending moment for I
ETABLE,MZJ,SMISC,16                  !Bending moment for J
ETABLE,EffAxiI,SMISC,1               !Axial force I
ETABLE,EffAxiJ,SMISC,14              !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
cmsel,s,bend,elem                    !ELBOW290
ETABLE,MYI,SMISC,2                   !Bending moment for I
ETABLE,MYJ,SMISC,37                  !Bending moment for J
ETABLE,MZI,SMISC,3                   !Bending moment for I
ETABLE,MZJ,SMISC,38                  !Bending moment for J
ETABLE,EffAxiI,SMISC,1               !Axial force I
ETABLE,EffAxiJ,SMISC,36              !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
/output

!=====================================================================!
!       Pressure Test
!=====================================================================!
set,3
```

```
/output,LC3,txt                     !Make file with name loadcase1
RSYS,11                             !Change to local CS for manifold end end
cmsel,s,KP1node,node
PRRSOL                              !Reaction forces
RSYS,12
cmsel,s,KP10node,node               !Change to local CS for PLET end
PRRSOL                              !Reaction forces
/output
esel,all


RSYS,0
alls
/output,LC3disp,txt
PRNSOL,DOF                          !Displacements
/output

alls
/output,LC3resutls,txt
cmsel,s,pipeelem,elem
cmsel,s,straight,elem               !PIPE288
ETABLE,MYI,SMISC,2                  !Bending moment for I
ETABLE,MYJ,SMISC,15                 !Bending moment for J
ETABLE,MZI,SMISC,3                  !Bending moment for I
ETABLE,MZJ,SMISC,16                 !Bending moment for J
ETABLE,EffAxiI,SMISC,1              !Axial force I
ETABLE,EffAxiJ,SMISC,14             !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
cmsel,s,bend,elem                   !ELBOW290
ETABLE,MYI,SMISC,2                  !Bending moment for I
ETABLE,MYJ,SMISC,37                 !Bending moment for J
ETABLE,MZI,SMISC,3                  !Bending moment for I
ETABLE,MZJ,SMISC,38                 !Bending moment for J
ETABLE,EffAxiI,SMISC,1              !Axial force I
ETABLE,EffAxiJ,SMISC,36             !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
/output

!=====================================================================!
!       Operating condition
!=====================================================================!
set,4
/output,LC4,txt                     !Make file with name loadcase1
RSYS,11                             !Change to local CS for manifold end end
cmsel,s,KP1node,node
PRRSOL                              !Reaction forces
RSYS,12
cmsel,s,KP10node,node               !Change to local CS for PLET end
PRRSOL                              !Reaction forces
```

```
/output

RSYS,0
alls
/output,LC4disp,txt
PRNSOL,DOF                              !Displacements
/output

alls
/output,LC4resutls,txt
cmsel,s,pipeelem,elem
cmsel,s,straight,elem                   !PIPE288
ETABLE,MYI,SMISC,2                      !Bending moment for I
ETABLE,MYJ,SMISC,15                     !Bending moment for J
ETABLE,MZI,SMISC,3                      !Bending moment for I
ETABLE,MZJ,SMISC,16                     !Bending moment for J
ETABLE,EffAxiI,SMISC,1                  !Axial force I
ETABLE,EffAxiJ,SMISC,14                 !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
cmsel,s,bend,elem                       !ELBOW290
ETABLE,MYI,SMISC,2                      !Bending moment for I
ETABLE,MYJ,SMISC,37                     !Bending moment for J
ETABLE,MZI,SMISC,3                      !Bending moment for I
ETABLE,MZJ,SMISC,38                     !Bending moment for J
ETABLE,EffAxiI,SMISC,1                  !Axial force I
ETABLE,EffAxiJ,SMISC,36                 !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
/output
!===========================================================================!
!          Design condition
!===========================================================================!
set,5
/output,LC5,txt                         !Make file with name loadcase1
RSYS,11                                 !Change to local CS for manifold end end
cmsel,s,KP1node,node
PRRSOL                                  !Reaction forces
RSYS,12
cmsel,s,KP10node,node                   !Change to local CS for PLET end
PRRSOL                                  !Reaction forces
/output

RSYS,0
alls
/output,LC5disp,txt
PRNSOL,DOF                              !Displacements
/output

alls
/output,LC5resutls,txt
```

```
cmsel,s,pipeelem,elem
cmsel,s,straight,elem                 !PIPE288
ETABLE,MYI,SMISC,2                    !Bending moment for I
ETABLE,MYJ,SMISC,15                   !Bending moment for J
ETABLE,MZI,SMISC,3                    !Bending moment for I
ETABLE,MZJ,SMISC,16                   !Bending moment for J
ETABLE,EffAxiI,SMISC,1                !Axial force I
ETABLE,EffAxiJ,SMISC,14               !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
cmsel,s,bend,elem                     !ELBOW290
ETABLE,MYI,SMISC,2                    !Bending moment for I
ETABLE,MYJ,SMISC,37                   !Bending moment for J
ETABLE,MZI,SMISC,3                    !Bending moment for I
ETABLE,MZJ,SMISC,38                   !Bending moment for J
ETABLE,EffAxiI,SMISC,1                !Axial force I
ETABLE,EffAxiJ,SMISC,36               !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
/output
!=================================================================!
!       Shutdown
!=================================================================!
set,6
/output,LC6,txt                       !Make file with name loadcase1
RSYS,11                               !Change to local CS for manifold end end
cmsel,s,KP1node,node
PRRSOL                                !Reaction forces
RSYS,12
cmsel,s,KP10node,node                 !Change to local CS for PLET end
PRRSOL                                !Reaction forces
/output
alls
/output,LC6disp,txt
PRNSOL,DOF                            !Displacements
/output

RSYS,0
alls
/output,LC6resutls,txt
cmsel,s,pipeelem,elem
cmsel,s,straight,elem                 !PIPE288
ETABLE,MYI,SMISC,2                    !Bending moment for I
ETABLE,MYJ,SMISC,15                   !Bending moment for J
ETABLE,MZI,SMISC,3                    !Bending moment for I
ETABLE,MZJ,SMISC,16                   !Bending moment for J
ETABLE,EffAxiI,SMISC,1                !Axial force I
ETABLE,EffAxiJ,SMISC,14               !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
cmsel,s,bend,elem                     !ELBOW290
ETABLE,MYI,SMISC,2                    !Bending moment for I
```

```
ETABLE,MYJ,SMISC,37                 !Bending moment for J
ETABLE,MZI,SMISC,3                  !Bending moment for I
ETABLE,MZJ,SMISC,38                 !Bending moment for J
ETABLE,EffAxiI,SMISC,1              !Axial force I
ETABLE,EffAxiJ,SMISC,36             !Axial force J
PRETAB,MYI,MYJ,MZI,MZJ,EFFAxiI,EffAXiJ
/output

alls
esel,all
/ESHAPE,1
EPLOT
PLDISP,2                        !Display deformed shape
/eof
```

COORDINATES FOR SCRIPT

3 LEGGED

K,	1	,	0	,	3.20	,	0
K,	2	,	2.38	,	3.20	,	0
K,	3	,	4.10	,	0	,	0
K,	4	,	11.60	,	0	,	0
K,	5	,	15.60	,	0	,	0
K,	6	,	15.60	,	0	,	28.60
K,	7	,	28.60	,	0	,	28.60
K,	8	,	33.43	,	0	,	33.43
K,	9	,	35.40	,	1.61	,	35.40
K,	10	,	37.09	,	1.40	,	37.09

4LEGGED

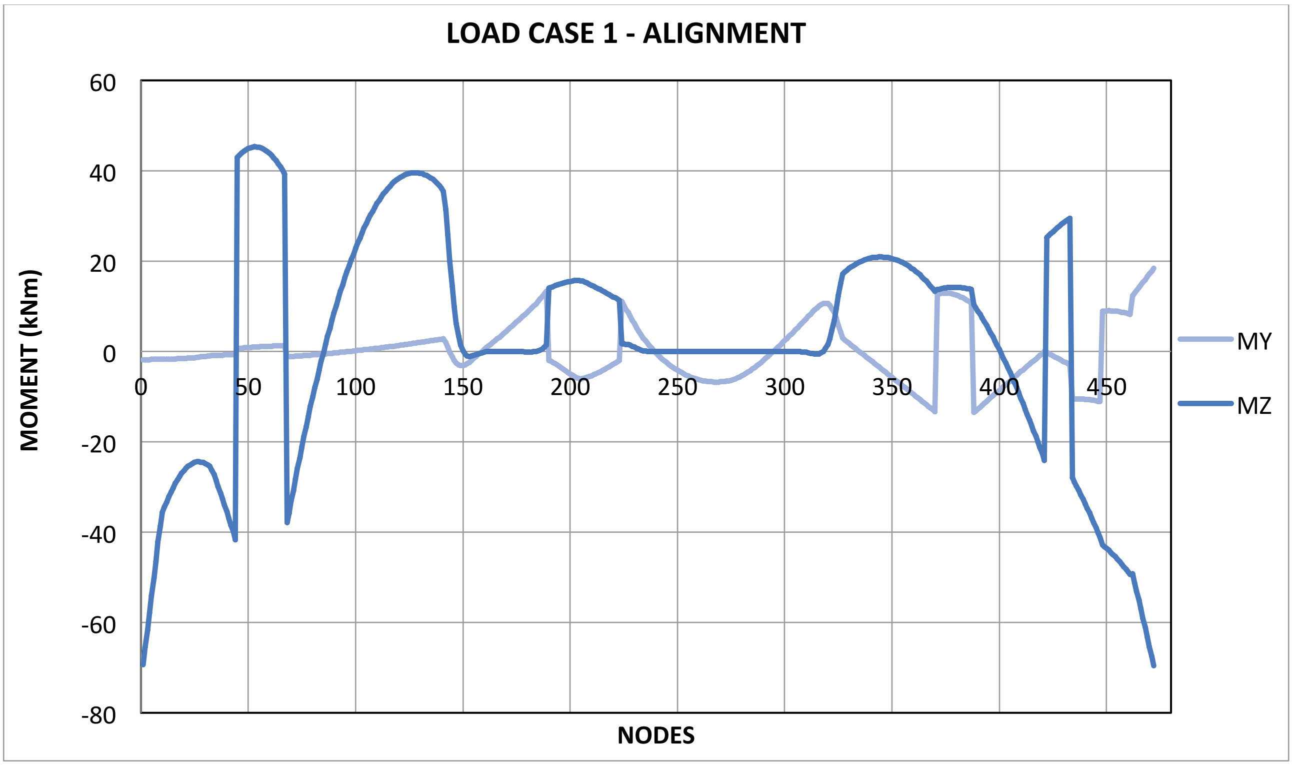
K,	1	,	0	,	3.20	,	0
K,	2	,	2.38	,	3.20	,	0
K,	3	,	4.10	,	0	,	0
K,	4	,	12.60	,	0	,	0
K,	5	,	16.60	,	0	,	0
K,	6	,	16.60	,	0	,	34.60
K,	7	,	34.60	,	0	,	34.60
K,	8	,	39.43	,	0	,	39.43
K,	9	,	41.41	,	1.61	,	41.40
K,	10	,	43.09	,	1.40	,	43.09

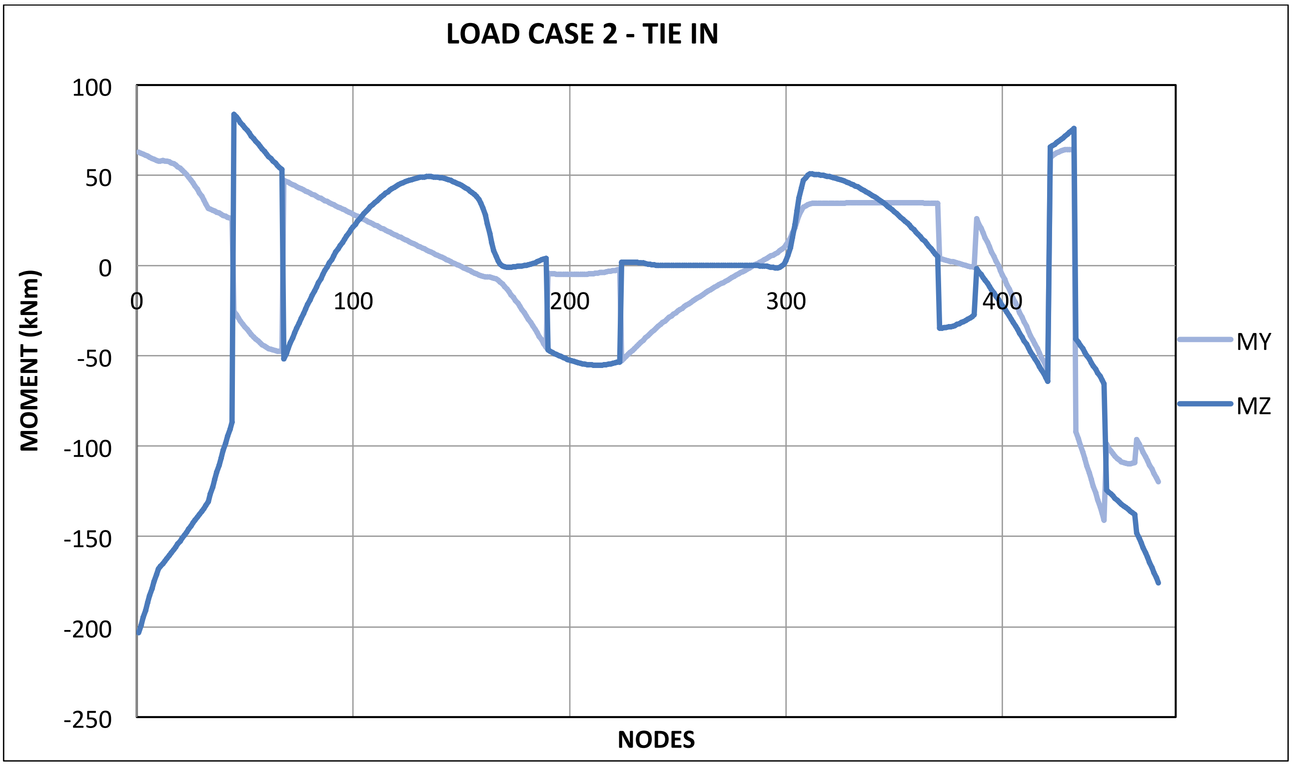
5LEGGED

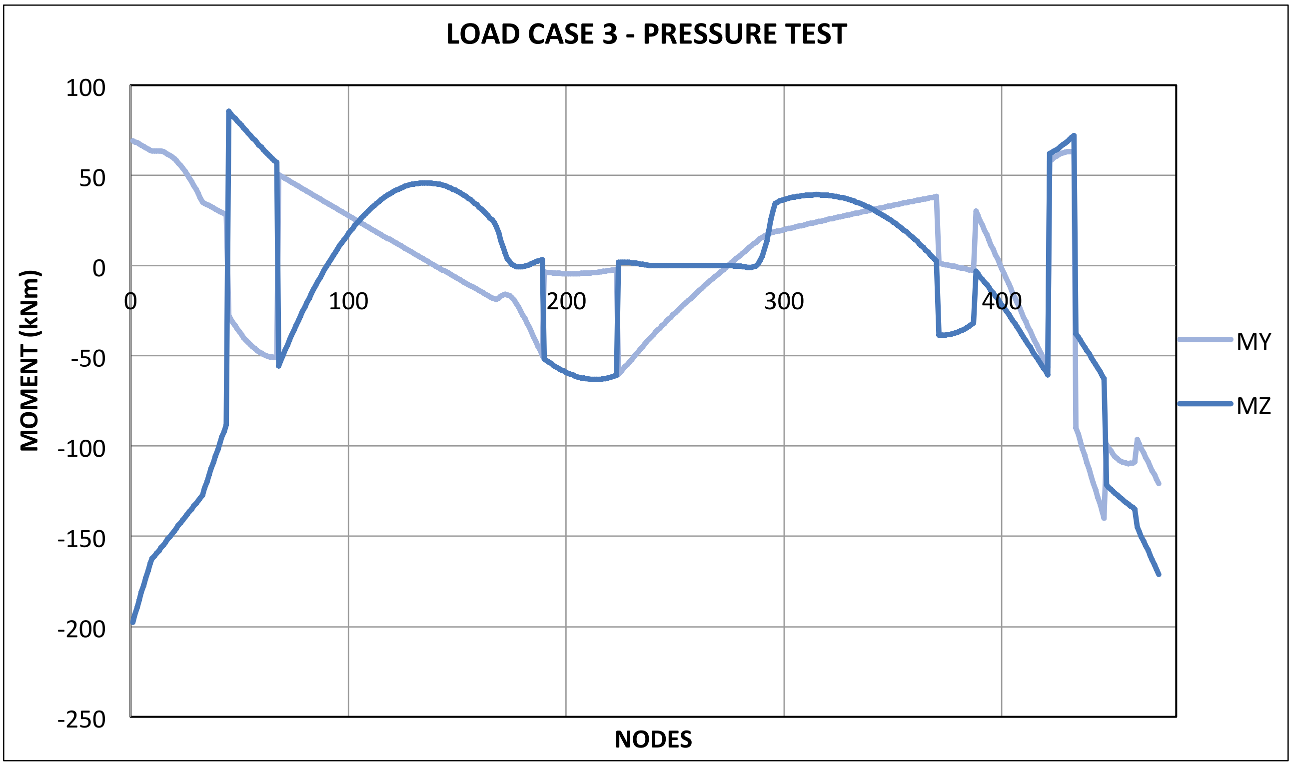
K,	1	,	0	,	3.20	,	0
K,	2	,	2.38	,	3.20	,	0
K,	3	,	4.10	,	0	,	0
K,	4	,	9.60	,	0	,	0
K,	5	,	14.30	,	0	,	-9.01
K,	6	,	23.83	,	0	,	-3.51
K,	7	,	13.82	,	0	,	13.84
K,	8	,	18.64	,	0	,	18.67
K,	9	,	20.62	,	1.61	,	20.64
K,	10	,	22.30	,	1.40	,	22.30

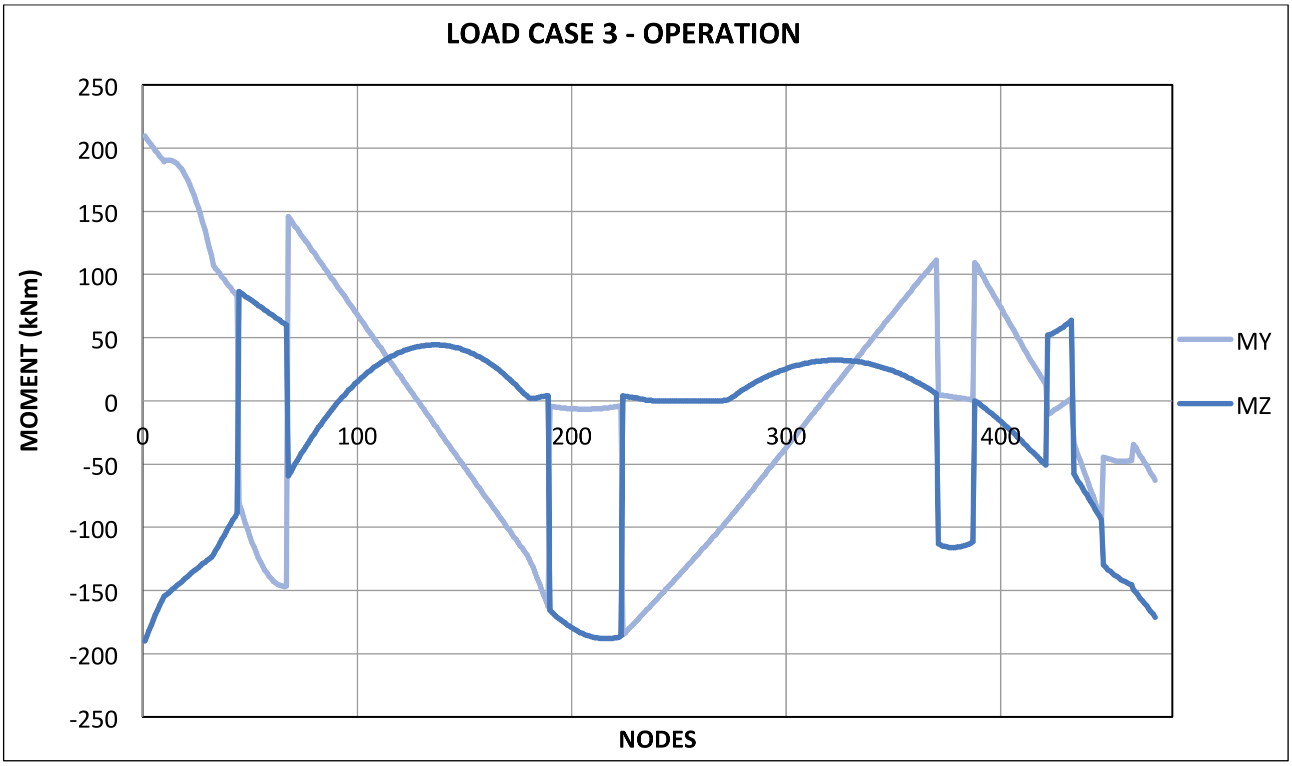
APPENDIX D: MOMENT PROFILES

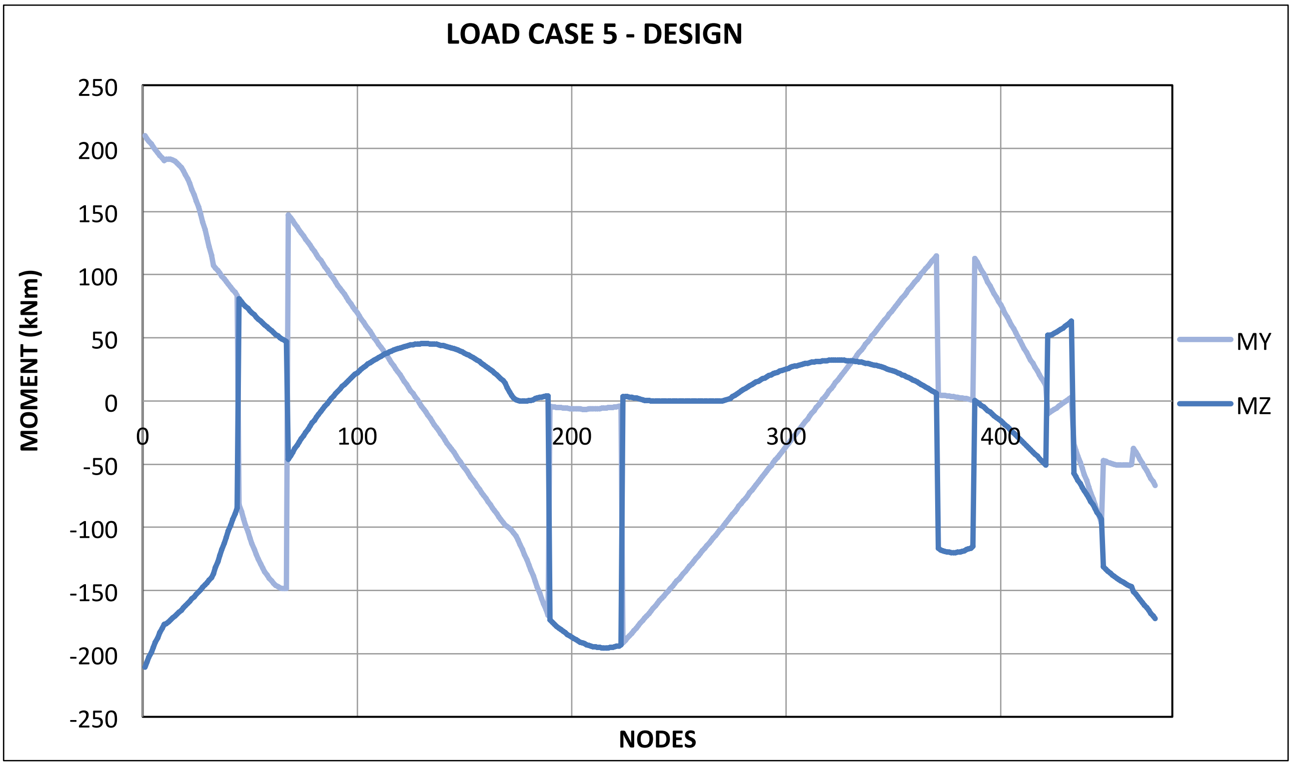
3-LEGGED SPOOL

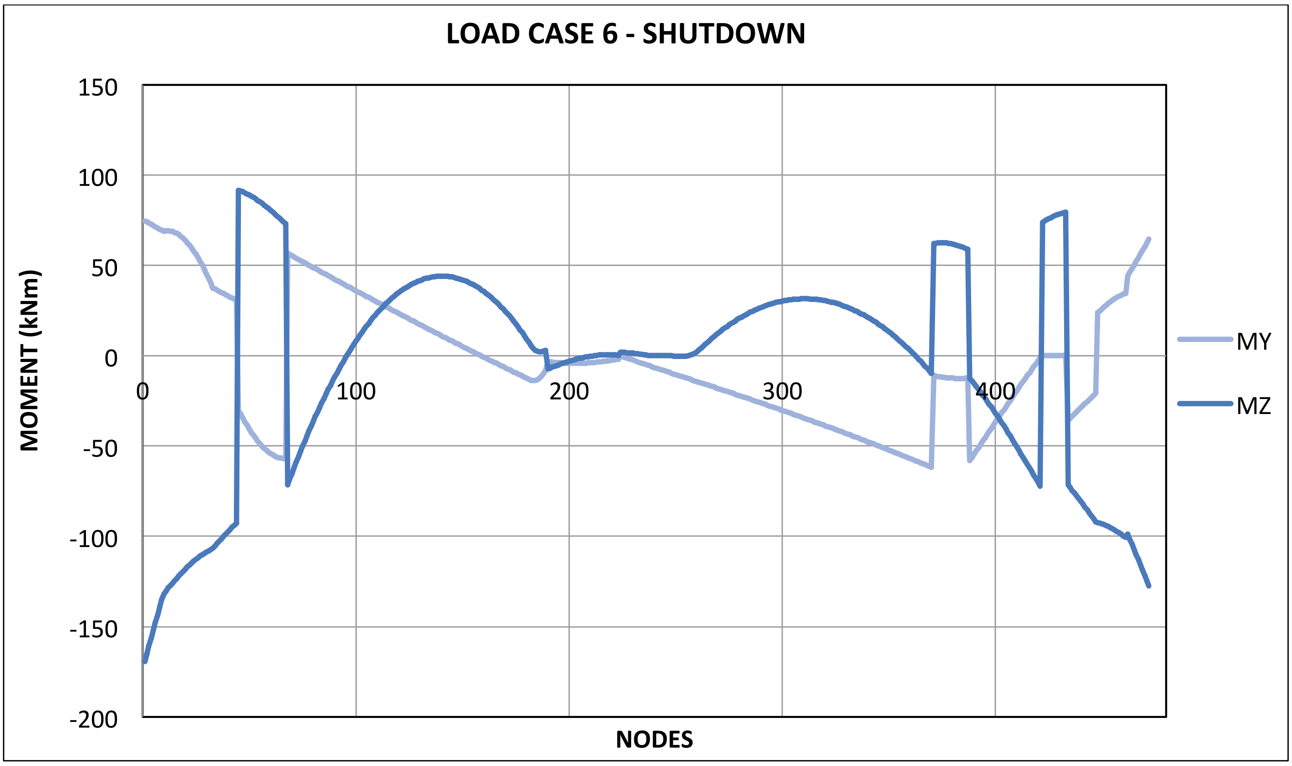




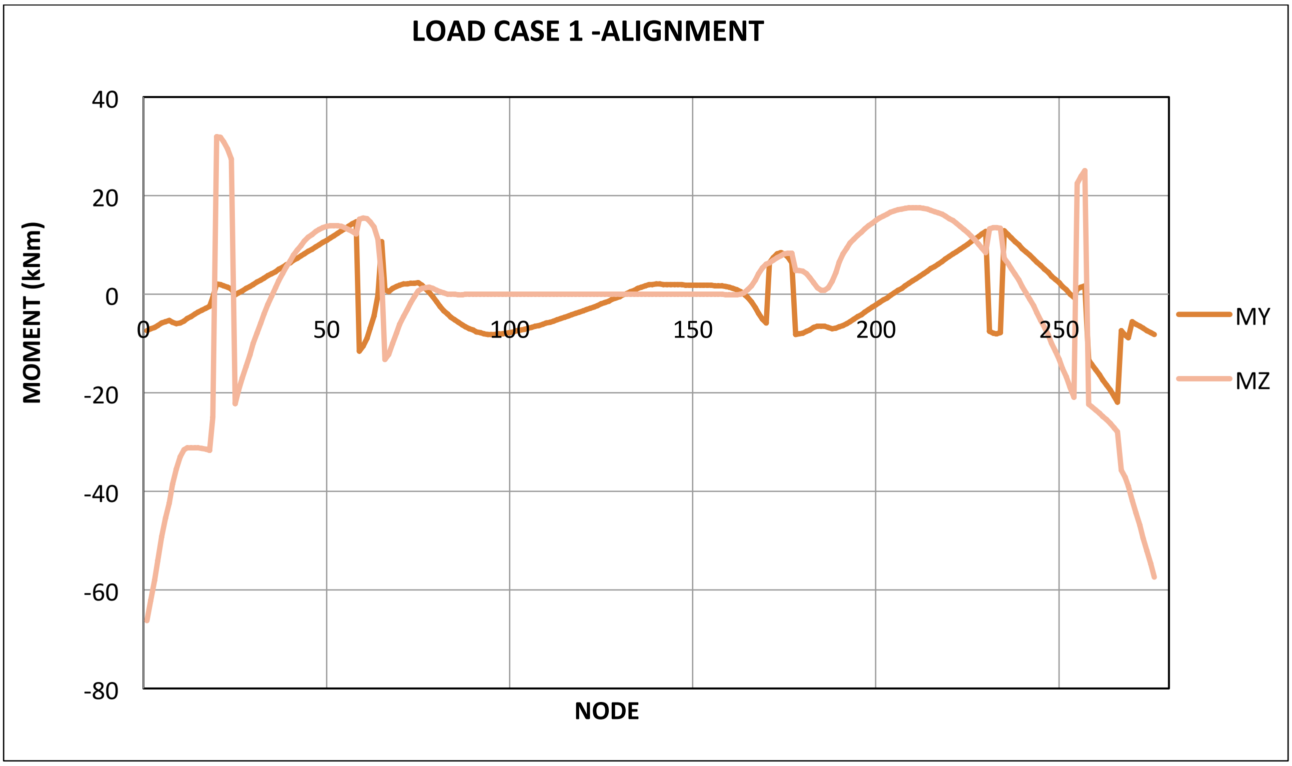

LOAD CASE 3 - PRESSURE TEST
100
50
0
-50
-100
-150
-200
-250
0
100
200
300
400
MOMENT (kNm)
NODES
MY
MZ

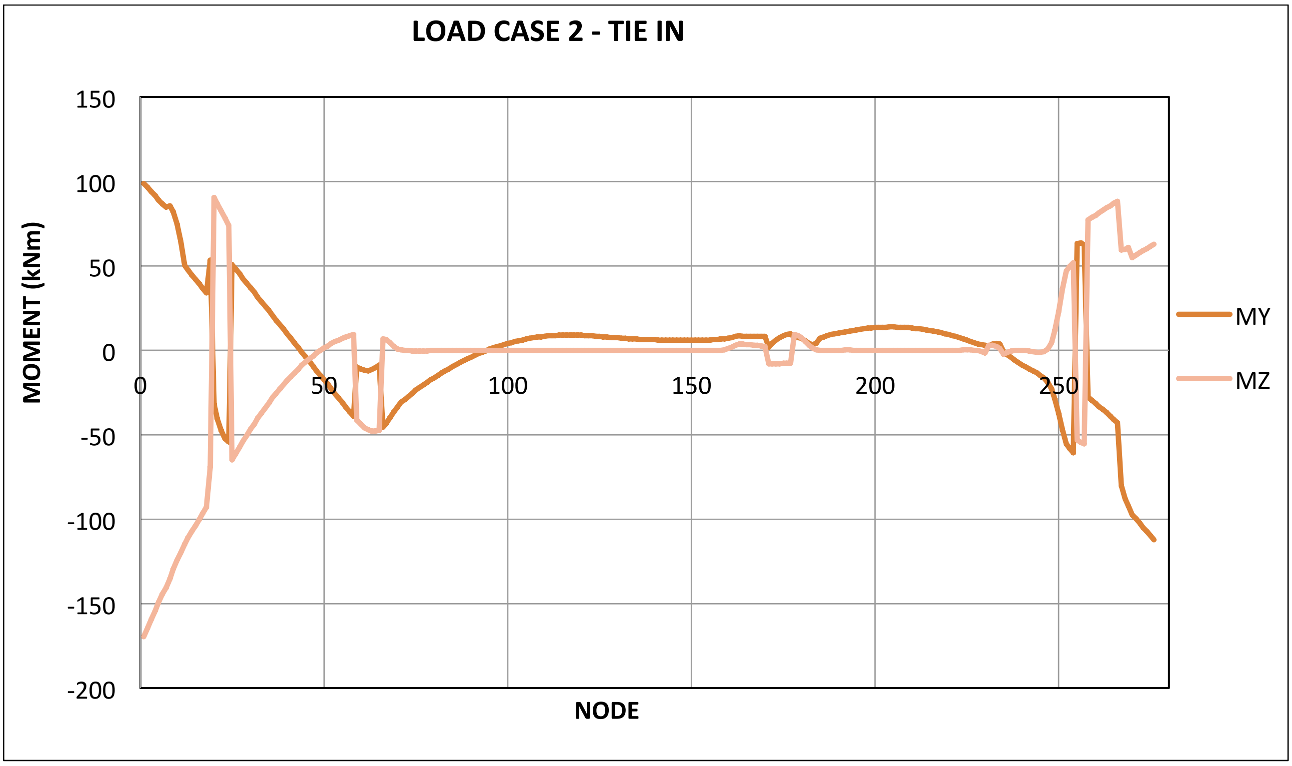

LOAD CASE 3 - OPERATION
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
100
200
300
400
MOMENT (kNm)
NODES
MY
MZ

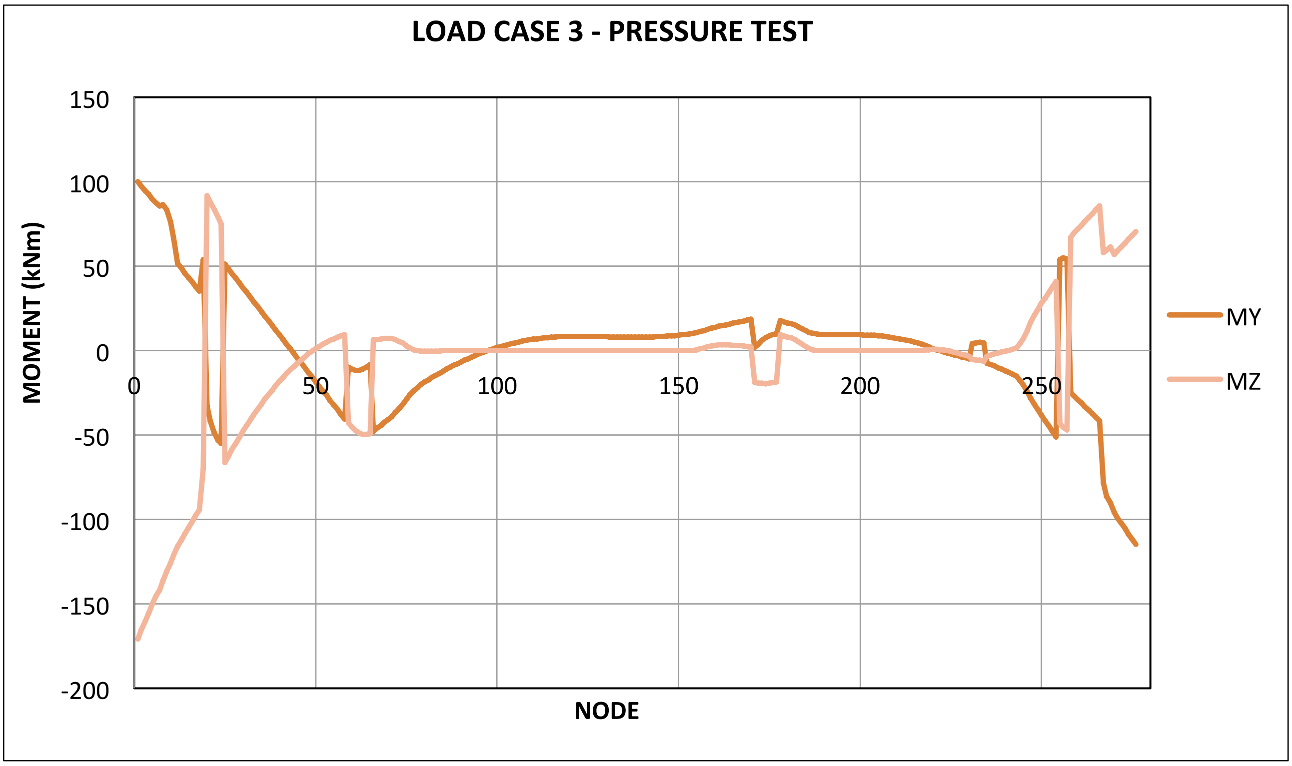

LOAD CASE 5 - DESIGN
250
200
150
100
50
0
-50
-100
-150
-200
-250
MOMENT (kNm)
0
100
200
300
400
NODES
MY
MZ

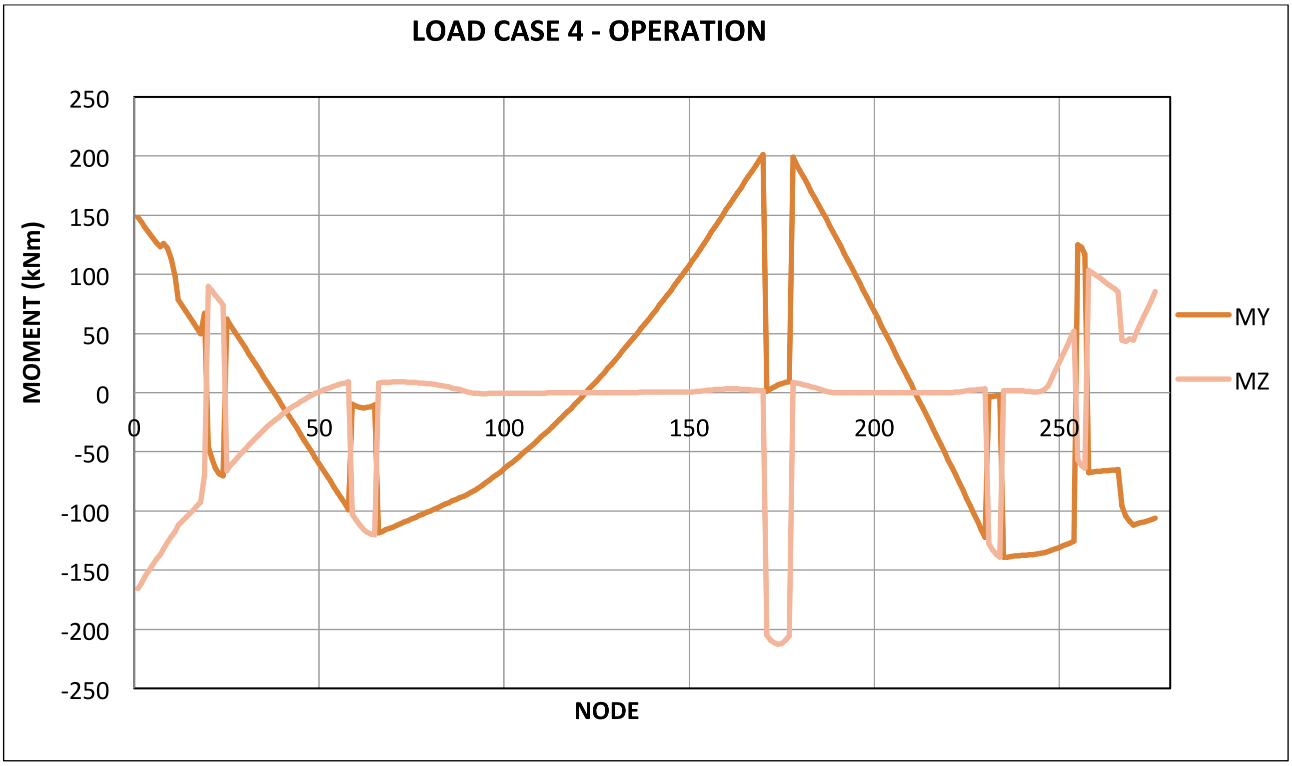

LOAD CASE 6 - SHUTDOWN
150
100
50
0
-50
-100
-150
-200
MOMENT (kNm)
0
100
200
300
400
NODES
MY
MZ

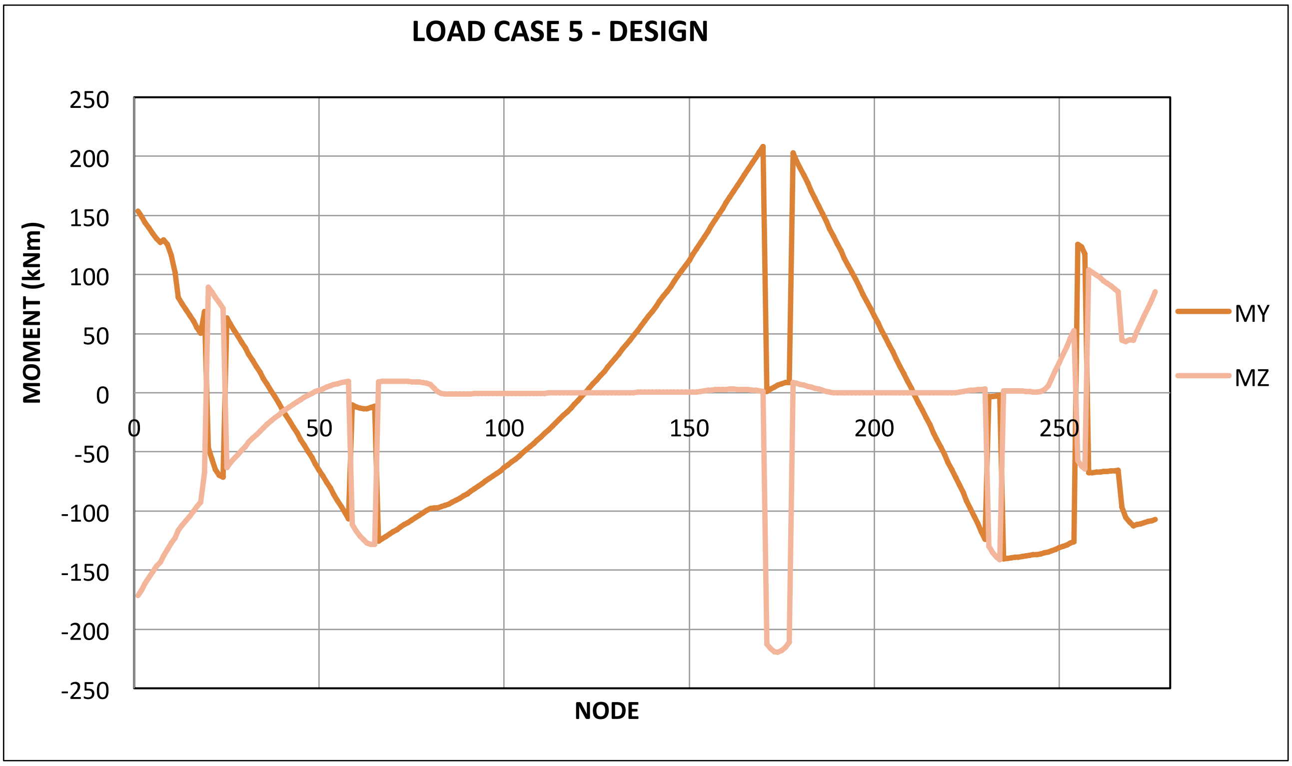
4 LEGGED SPOOL

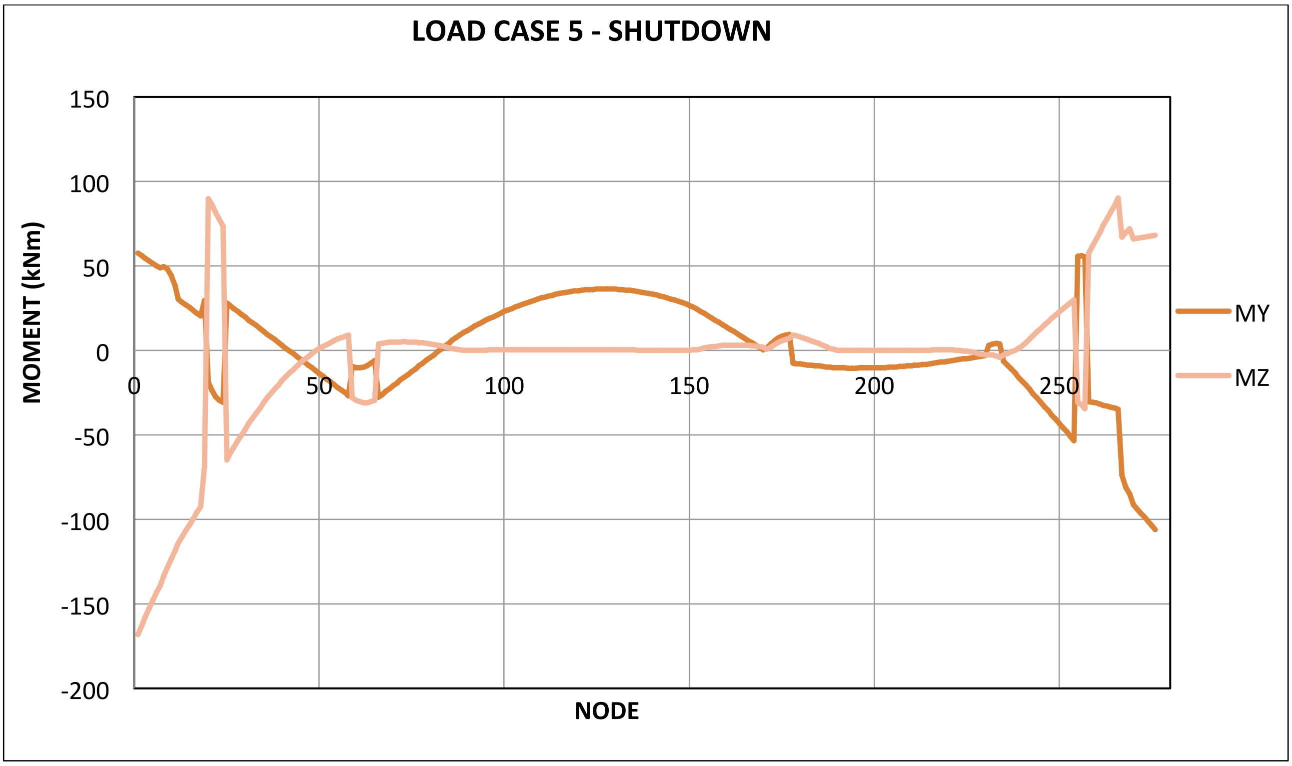




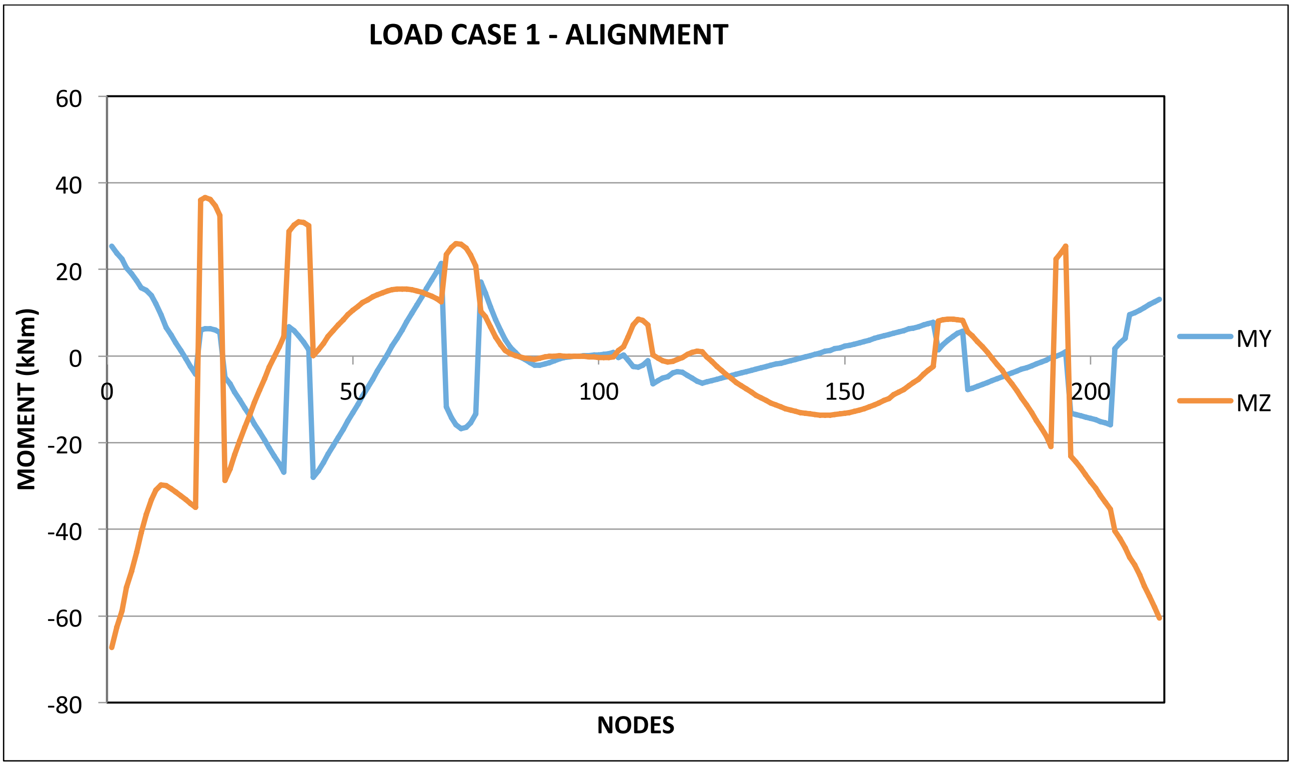

LOAD CASE 3 - PRESSURE TEST
150
100
50
0
-50
-100
-150
-200
MOMENT (kNm)
0
50
100
150
200
250
NODE
MY
MZ

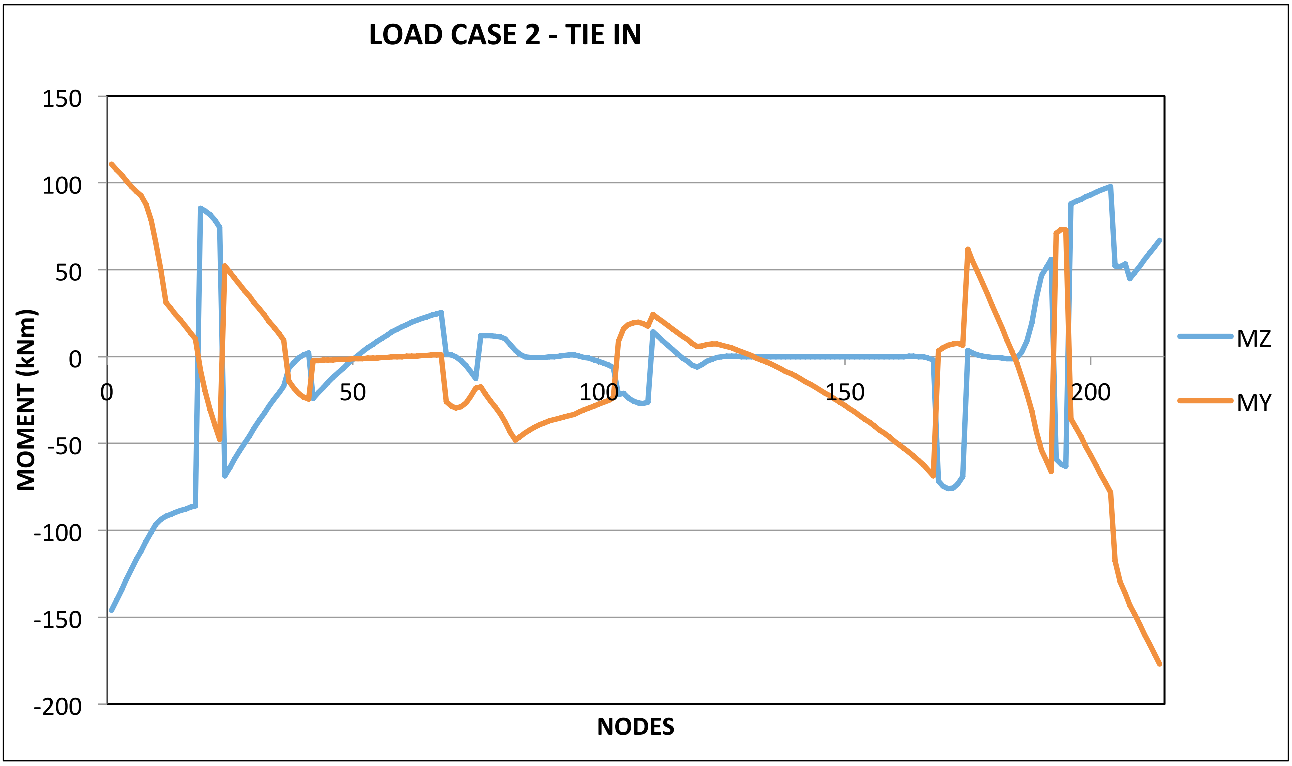

LOAD CASE 4 - OPERATION
250
200
150
100
50
0
-50
-100
-150
-200
-250
MOMENT (kNm)
0
50
100
150
200
250
NODE
MY
MZ

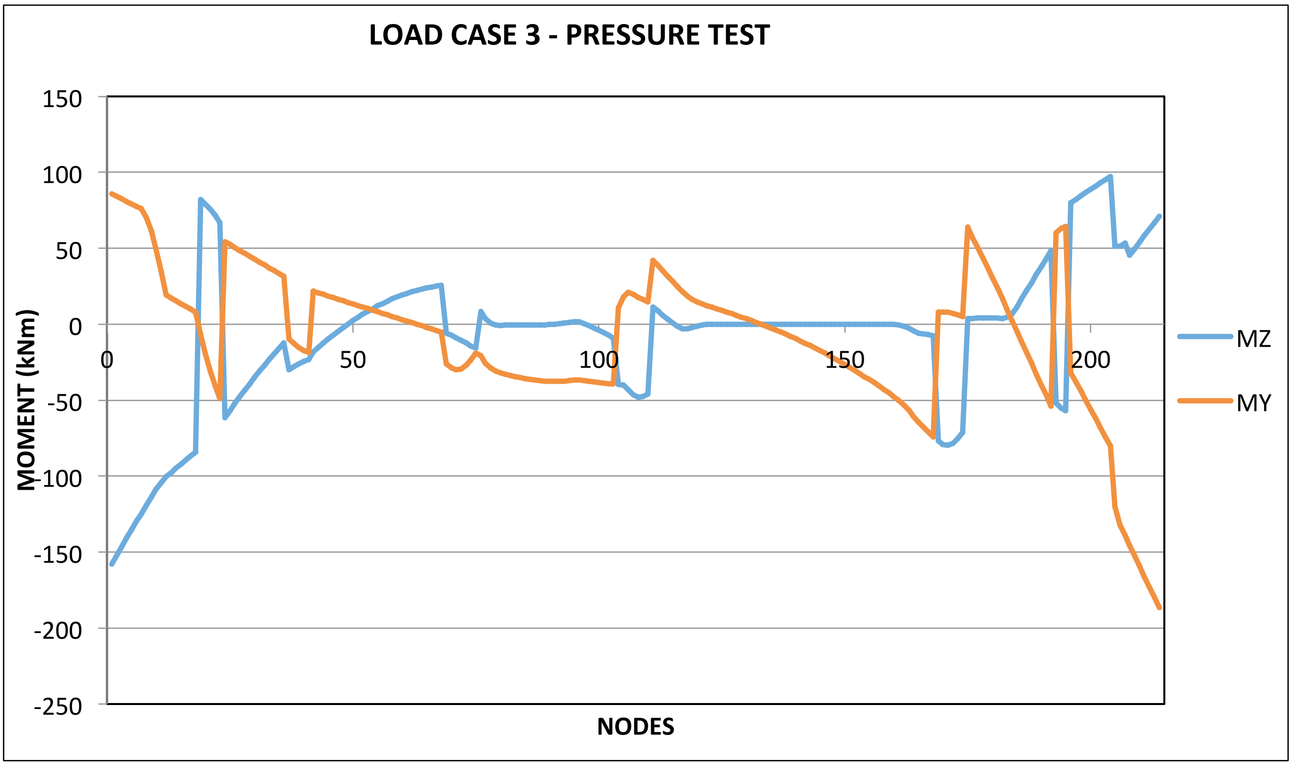

LOAD CASE 5 - DESIGN
MOMENT (kNm)
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
50
100
150
200
250
NODE
MY
MZ


LOAD CASE 5 - SHUTDOWN
MOMENT (kNm)
150
100
50
0
-50
-100
-150
-200
0
50
100
150
200
250
NODE
MY
MZ

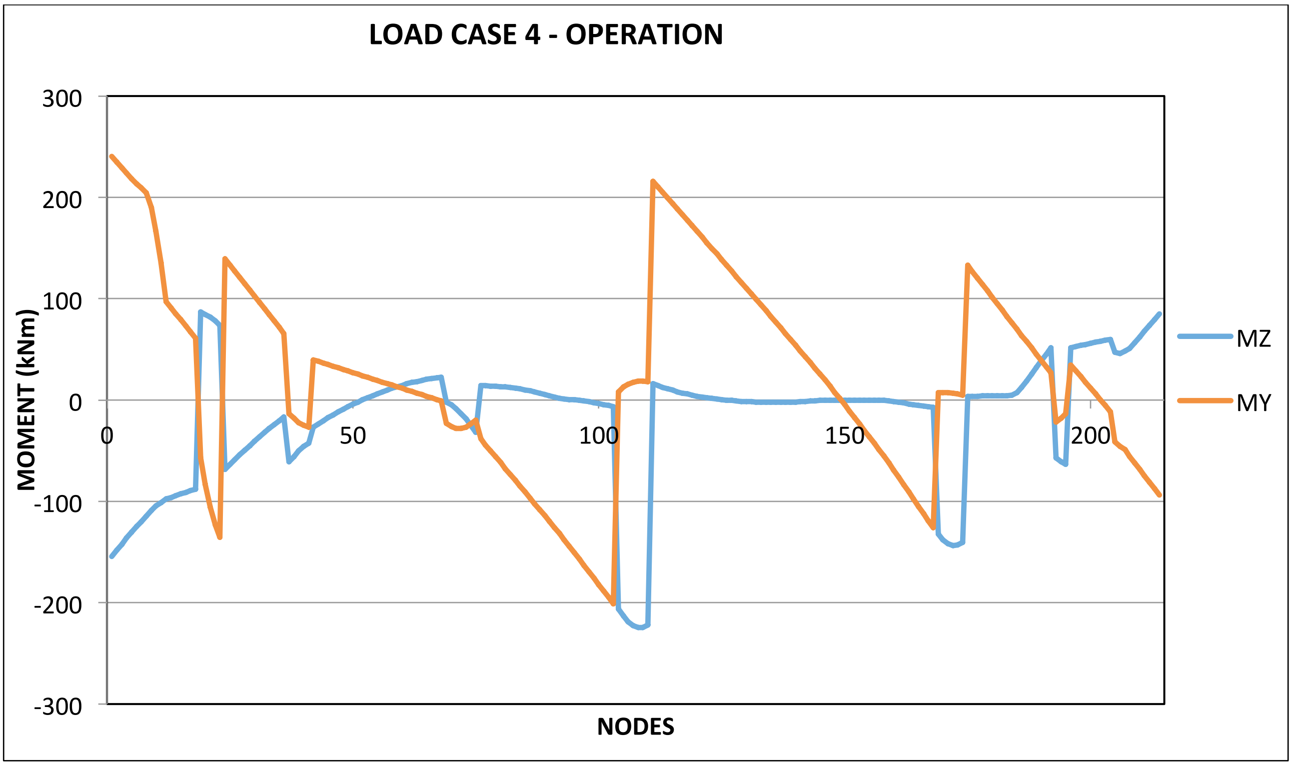
5-LEGGED SPOOL



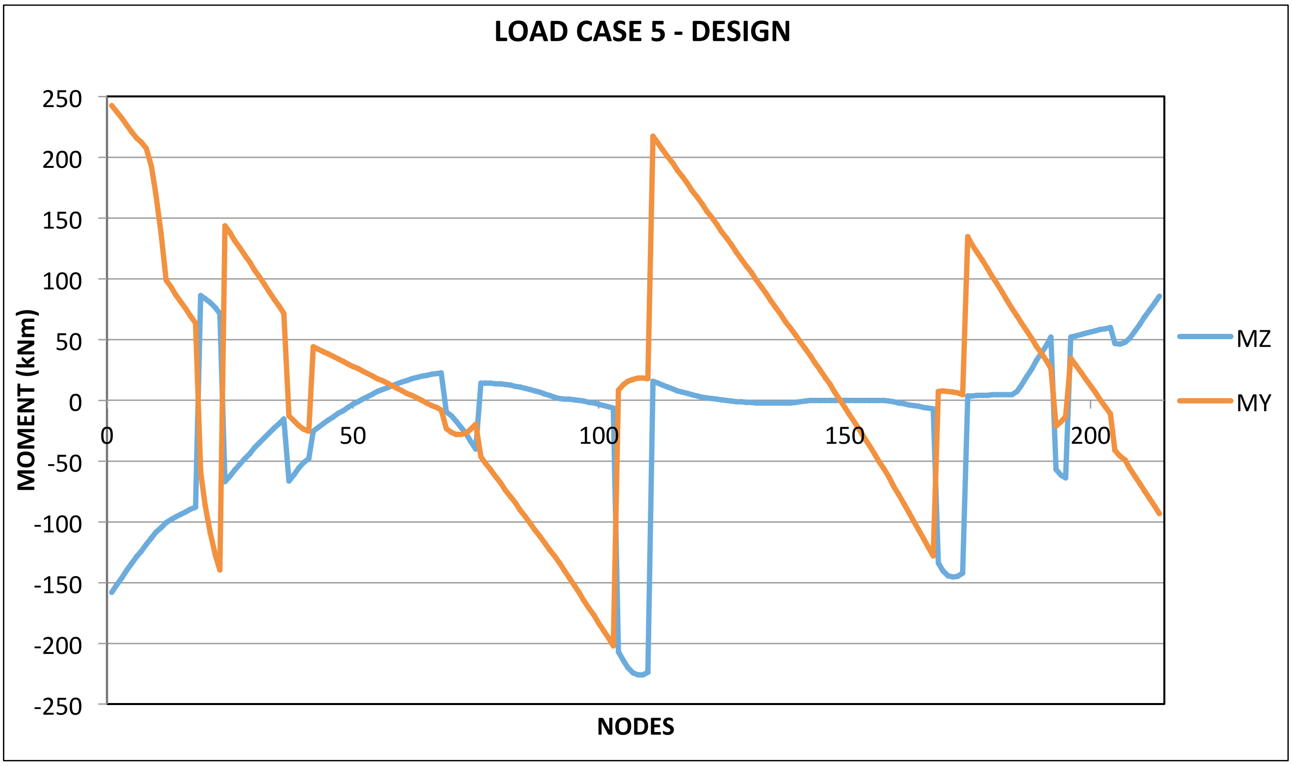


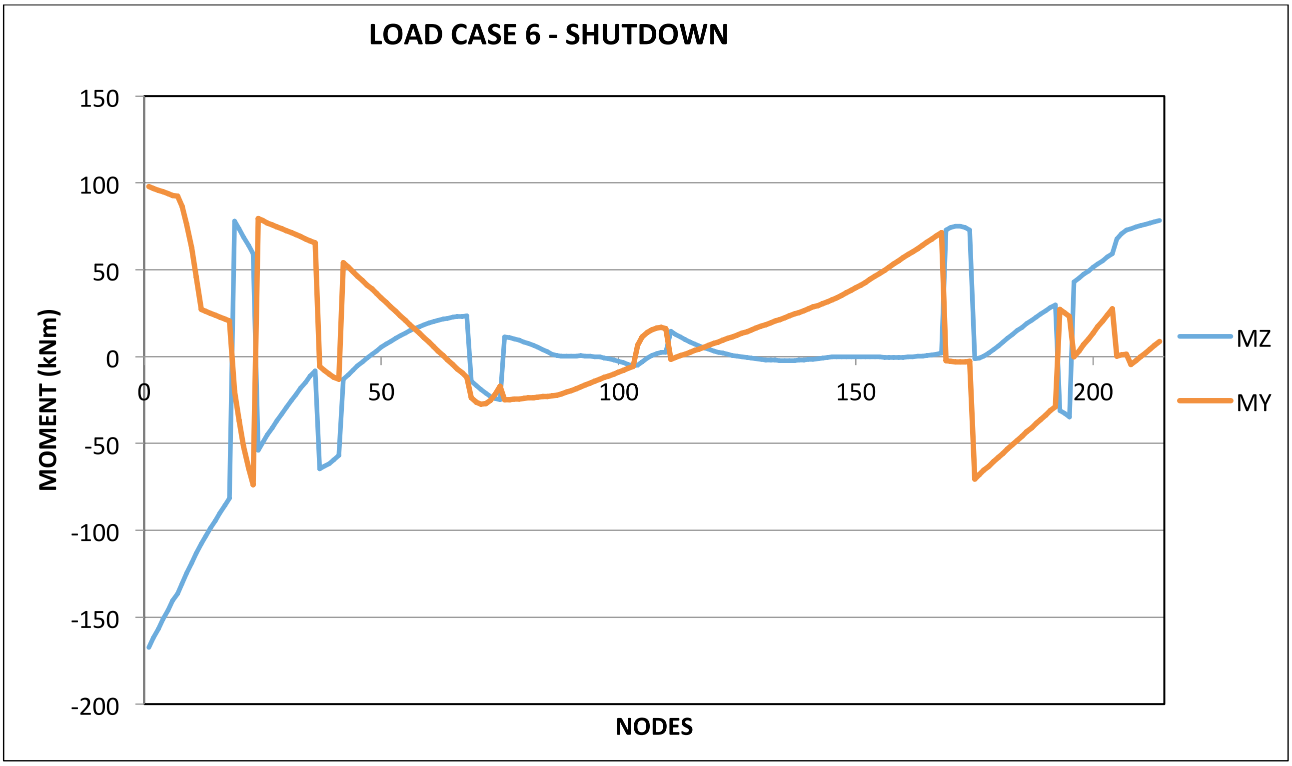


LOAD CASE 3 - PRESSURE TEST
150
100
50
0
-50
-100
-150
-200
-250
MOMENT (kNm)
0
50
100
150
200
NODES
MZ
MY



LOAD CASE 4 - OPERATION
300
200
100
0
-100
-200
-300
MOMENT (kNm)
0
50
100
150
200
NODES
MZ
MY


LOAD CASE 5 - DESIGN
250
200
150
100
50
0
-50
-100
-150
-200
-250
MOMENT (kNm)
0
50
100
150
200
NODES
MZ
MY


LOAD CASE 6 - SHUTDOWN
150
100
50
0
-50
-100
-150
-200
MOMENT (kNm)
0
50
100
150
200
NODES
MZ
MY