University of Stavanger

Faculty of Science and Technology

MASTER'S THESIS

Study program/ Specialization: Offshore Technology, Marine and Subsea Technology	Spring semester, 2012 Restricted access
Author: Thomas Sola Larsen	………………………………………… (Author's signature)
Faculty supervisor: Ove Tobias Gudmestad External supervisor(s): John Hansen	
Title of thesis: Analysis of new titanium Modular Stress Joint (MSJ) for subsea completion and workover operations. Patent Pending with application #US13/506,352	
Credits (ECTS): 30	
Key words: Titanium, Stress Joint, MSJ, Modular, Riser, FlexJoint, Modular Stress Joint,	Pages: ………………… + enclosure: ………… Stavanger, ……………….. Date/year

Abstract

A stress joint is a transitional joint that when used can be located both at the top and bottom of an offshore riser string. The main idea of the stress joint is to have a tapered section between the movement in the riser string and the fixed connection to the subsea equipment. This is done to control the curvature of the connection and to limit the local bending stresses.

This master thesis looks at the development of the titanium Modular Stress Joint (MSJ). The process started during the summer of 2011 with the development of the design basis with the following 4 key-objectives: Lowering manufacturing cost and lead time, and increasing transportability and versatility. A prototype was manufactured in a 1/3 scale size to be tested for form and function. The scale model was tested in accordance with the test procedure (Appendix A) and results from attached strain gauges were compared with Finite Element Analysis (FEA) results.

The results from the testing of the prototype showed that the titanium MSJ was performing better than expected and provided confidence in the FEA analysis performed. These results were taken into the modeling of the full size MSJ in a global riser modeling performed with Orcaflex. The titanium MSJ performance was compared with a steel Tapered Stress Joint (TSJ). The results show that the MSJ outperforms a steel TSJ of equal size.

The results showed stresses in the material in unwanted areas, further development of the titanium MSJ is recommended to incorporate changes listed in the Conclusions in Chapter 5.

The titanium MSJ is patent pending (April 2012) with application # US13/506,352.

Table of Contents

Nomenclature

List of symbols

D_i	Internal diameter, ID
D_o	Outside diameter, OD
E	Modulus of elasticity – Young's modulus – Material specific
$F_{bending}$	Force in bending
f_m	JONSWAP wave parameter
$F_{tension}$	Force in tension
H_{MAX}	Max wave heigh
H_s	Significant wave height
L	Length
M_d	Design Bending moment
M_k	Plastic bending moment resistance
σ_y	Yield strength
$P_b(t)$	Burst resistance
P_e	External Pressure
P_{in}	Local Incidental Pressure
$P_{i,test}$	Internal test pressure
$P_{i,work}$	Internal work pressure
t	Wall thickness
T_{ed}	Design effective tension
T_k	Plastic axial force resistance
T_p	JONSWAP wave parameter – Period of wave
T_z	JONSWAP wave parameter – Zero-crossing period
x	Bending deflection of prototype MSJ

List of Greek letters

α_c	Parameter accounting for strain hardening and wall thinning
α_{tb}	Titanium Material correction factor
α_{tm}	Titanium Material correction factor 2
γ	JONSWAP wave parameter
γ_c	Condition factor for bending, torsion and internal overpressure
γ_m	Material resistance factor
γ_{sc}	Safety class resistance factor
ε_{xx}	Strain in xx direction $(rr, \theta\theta, zz)$
ν	Poisson ratio of material
σ_{xx}	Stress in xx direction $(rr, \theta\theta, zz)$
σ_u	Tensile strength
σ_y	Yield strength

Unit conversions

1 Psi	6,895 kPa
1 Psi	0,006895 MPa
1 Inch	25,4 mm
1 MPa	10 bar
1 lbs	453,59 gram

List of Abbreviation

BOP	Blow-Out Preventer
CRA	Corrosion Resistant Alloy
DNV	Det Norske Veritas
DOF	Degrees Of Freedom
EDP	Emergency Disconnect Package
FEA/FEM	Finite Element Analysis/Method
HBM	Hottinger Baldwin Messtechnik GmbH
ID	Internal Diameter
ISO	International Standard Organization
JONSWAP	JOint North Sea WAve Project
Ksi	Kilo pounds per square inch
LMRP	Lower Marine Riser Package
MSJ	Modular Stress Joint
NACE	National Association of Corrosion Engineers
OD	Outside Diameter
Pa	Pascal
Psi	Pounds per square inch
RP	Recommended Practice
RAO	Response Amplitude Operator
SRP	Subsea Riser Products Ltd.
TSJ	Tapered Stress Joint

Figures

List of Figures

Tables

List of Tables

Acknowledgements

There are numerous people and institutions to acknowledge in their support of this master thesis. I would first and foremost like to acknowledge Titanium Engineers AS and its Technical Director Mitch Dziekonski for allowing me to write this thesis. I have received help from both the Houston and Birmingham office of Titanium Engineers with various problems along the way.

I would also like to acknowledge the help provided by our sister engineering company AlTiSS Technologies and especially John Hansen, Cody DeHart and Dr. Young-Hoon Han. Their help with Finite Element Analysis and testing is greatly appreciated.

I would like to acknowledge our partner in the product development of the MSJ, Subsea Technologies Ltd. They provided help with the testing and Orcaflex modeling. A special acknowledgement is sent to their Principal Analysis Engineer, Ashley Bird.

I would like to send a special acknowledgement to the personnel at the University of Stavanger and especially; Head engineer Ahmad Yaaseen Amith, Professor II Ljiljana D. Oosterkamp and Professor Ove Tobias Gudmestad. Their help with various problems, guidance and suggestions have been greatly appreciated and valued

I would finally like to thank my cohabitant Lillan Linnestrand Dammen for her love and support.

____

Thomas Sola Larsen

Stavanger ________/06-2012

1. Introduction

1.1 Background

A stress joint is a transitional joint that when used can be located both at the top and bottom of an offshore riser string. The riser string comprises of a set of pipes connected with threaded connections. The riser string is the pressure vessel that allows well access to subsea wells from the surface. A schematic found in (ISO 13628-7 2006) is shows in Figure 1.1.1 on the following page. The figure provides an explanation to the position of the equipment and the location of the stress joint.

The surface structure may be a floating or permanent on site structure. The stress joint is utilized as a transitional joint between the potential movement in the riser string and the fixed subsea well equipment. As this is largely caused by the motions from floating structures, this thesis will look at floating structures.

The main idea of the stress joint is to have a tapered section between the movement and the fixed connection. This is done to control the curvature of the connection and to limit the local bending stresses. The use of a tapered section to do this is a well known solution that has existed in designs for centuries. It is difficult to specifically pinpoint when the tapered stress joint was first introduced into the offshore riser string. Several patents ranging from 1978 (Conoco 1978) to 2003 (Abadi and ABB Vetco Gray 2003) have been identified.

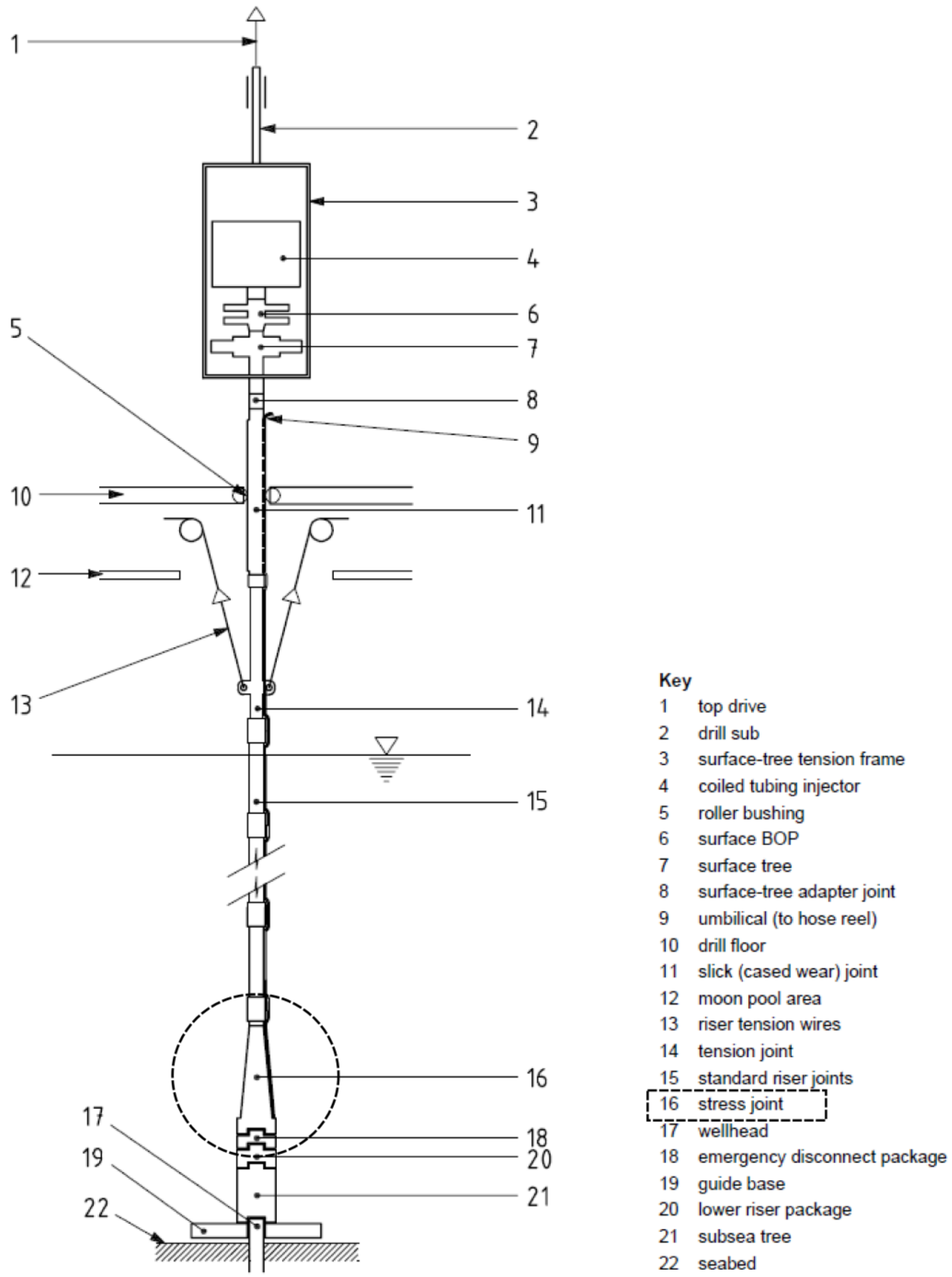


Figure 1.1.1: Typical completion / workover riser arrangement (ISO 13628-7 2006)

I was first introduced to the Modular Stress Joint during the summer of 2011. I worked for Titanium Engineers Inc. based out of Houston, Texas. The idea was to utilize Titanium Engineers accumulated knowledge and understanding of titanium as a material to develop a new product. The technical director, Mitch Dziekonski at Titanium Engineers had the idea of creating a Modular Stress Joint (Patent Pending #13/506352). Current stress joint designs are made out of a single cast and the dimensions require a special ingot and forgings to be performed. Large volumes of material are machined away using this method. This is both a time consuming and costly process.

The idea of the modular stress joint is to create a system of interchangeable parts that may be prefabricated and assembled for a particular use. This would then dramatically reduce both lead time and cost of purchase. The advantages and improvements in this modular design are listed in Chapter 1.2 Design basis.

1.2 Design basis

1.2.1 Introduction

The design basis for the Modular Stress Joint (MSJ) was developed during June/July 2011. The design basis may be divided into four main categories. They are manufacturing cost, production lead time, transportability and versatility. The main objective is to create a product that performs as good or better than other available products currently on the market. A set of competitive products are listed in Chapter 2: State of the Art.

The design basis is outlining a product design that shall have better performance in all four categories compared to the current designs. It is important to note that this thesis will look at both a scale prototype for testing and a full size modular stress joint. The inherent difference in their usage is reflected in the design basis and will be commented in their respective analysis chapters, Chapter 3 and 4.

1.2.2 Manufacturing Cost

The current steel or titanium stress joint design is made up of a large piece of forged metal with either upset forged flanges or welded on flanges. In the case of upset forging, a large amount of excess material is present over the length of the stress joint. The forged metal is then machined into a taper over its entire length. This is both a costly and time consuming process. The only potential off-the-shelf part would be the welded on flanges.

The Modular Stress Joint (MSJ) design comprise of several sections that may be more easily and cheaper acquired due to their size and decreased complexity. This design removes the large and costly special forging that is required otherwise. The new MSJ design will reduce costs by using more off-the-shelf parts, including flanges and straight section titanium tubes. The Modular design would directly contribute to a considerate reduction in manufacturing costs.

1.2.3 Lead time

Another important part of the design basis is to reduce the lead time from order to delivery of a stress joint. The current stress joint design utilizes custom forging and machining. The operations required for manufacturing reduces the list of potential contractors, both because of equipment and the knowledge that is required. It is not uncommon to talk about a lead time of up to and around 18 months from start of design to delivery of a stress joint.

The MSJ design will reduce the lead time by utilizing off-the-shelf parts and the modular design. The large time driver for the forged stress joint is the special forging required. By removing this step, the lead time from order to delivery may be reduced from 18 months. By combining the inherent characteristics and versatility of a modular design with pre-manufactured parts we believe that the lead time may be reduced to 3-4 weeks.

1.2.4 Transportability

The stress joint design of today is both large and heavy. Transport of stress joints have mainly been performed with ocean transport. A great advantage to the modular design is that the MSJ may be dismantled and transported in pieces/sections. Contact with air freight companies were established during the design basis development. Maximum length and weight of each part to allow simple and quick air freight were established and incorporated in the design.

1.2.5 Versatility

Versatility is the large cost and time saver for the MSJ. Current stress joints are custom designed for specific field/area/conditions with certain wave and vessel motions. The result being that one requires several specially designed stress joints to perform intervention on various subsea wells in one geographical area with changing conditions. The MSJ solves this customization problem with its built in modular design. By adding or removing titanium tube sections, the characteristics of the MSJ may be altered to fit a certain set of conditions. The base of the MSJ will not be changed while the titanium tubes may be added or removed to get the required bending characteristics. This will thus reduce costs by removing the need for expensive customized stress joints with limited use outside its design parameters.

Another benefit with the MSJ is the fact that parts may be replaced in case of damage. A scenario where one of the modular pieces in the MSJ is damaged during installation of the riser string, may be solved by replacing that part. This replacement will also be performed on the vessel and thus reduce the downtime.

1.2.6 Relevant standards

The governing standard within intervention riser systems is ISO 13628-7 "Petroleum and natural gas industries – Design and operation of subsea production systems – Part 7: Completion/workover riser systems" (ISO 13628-7 2006).

The first item to note when working with this standard is the following sentence in Chapter 1 Scope: "This part of ISO 13628 is limited to risers, manufactured from low alloy carbon steels. Risers fabricated from special materials such as titanium, composite materials and flexible pipes are beyond the scope of this part of ISO 13628. (ISO 13628-7 2006)". A decision was made early in the design process to use the standard but to use Titanium Engineers developed specialist knowledge as good engineering practice over the standard.

This stress joint is mainly covered in Chapter "5.4.13 stress joint" in the ISO standard. The Chapter is limited by the current stress joint designs. It does not cover connections on the stress joint. A decision was made to follow the Chapter on connectors for the riser in general.

Another important document in the development of the MSJ is DNVs DNV-RP-F201 "Design of Titanium Risers" (Det Norske Veritas October 2002). This recommended practice covers the design of a titanium riser system and also its components like the stress joint. It must be emphasized that the DNV RP covers several grades of titanium but it does not specifically cover the titanium grade chosen for the MSJ. This is not seen as a problem as Titanium Engineers have developed large expertise with this specific type of titanium. This is further covered in Chapter 3

Any material in contact with hydrocarbons and especially sour well conditions will be required to be in compliance with relevant NACE standards and especially the NACE MR0175. This standard covers the corrosion resistance of materials. The MSJ is designed with this in mind so that all material in contact with the well fluid is NACE compliant.

1.2.7 Modular design

The MSJ general design idea was completed at an early stage of the project. More time was spent on how to connect the modular pieces together into a functioning system. Several designs were laid out in sketch form. A decision was made to use threaded connections in a Pin-Box-Pin system. Titanium Engineers have developed several proprietary thread profiles over the last few years. The threads are designed to limit galling between the steel in the box and the titanium tube pin in the Pin-Box-Pin system. A connector design was developed for the prototype which is tested in Chapter 3. Testing of the prototype showed potential problems with this thread profile, these are further covered in Chapter 3.4.5. No final decision has been made with regards to connector design for the full size MSJ. They have been kept the same for the purpose of this master thesis.

The core idea with the threaded connections is how the system can be assembled and then disassembled quickly. It is important that any threads developed are customized for multiple break-in and break-outs. Wear on threads and sealing surfaces must be kept to a minimum.

1.2.8 Prototype

The prototype is designed as an approximately 1/3 scale model of the full size MSJ. A decision was made to go ahead with the prototype without a finished thread and connector design. The design of the connectors was based on another related project worked on by Titanium Engineers. The connector design shall allow for required testing while still providing pressure and structural integrity. The prototype was scaled with geometric scaling laws with a factor of 38%. This is further covered in Chapter 3.

1.3 Design

The design basis provided input in the design process moving from concept to drawings. The process started June/July 2011 and finished with a design by September 2011. Several different designs were discussed and tested with FEA software. The end result is seen below in Figure 1.3.1. In general it comprises of a tapered base piece and two interchangeable straight tubes. The flanges on each end will be interchangeable to allow connections to different riser systems. Comments on the design will be provided on a parts basis.

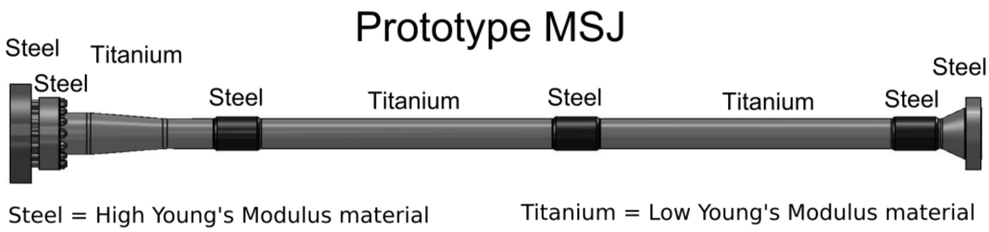


Figure 1.3.1: Prototype modular stress joint materials

1.3.1 The parts

The titanium base is the lowermost titanium piece with a slight tapered outside diameter. This piece is kept in place with a steel swivel flange connected with bolts to the lowermost steel flange. A metal-to-metal gasket is installed between the titanium base and the bottom steel flange.

Over the titanium base is the first of three steel connectors. The connectors are threaded with dual metal-to-metal seals. ISO 13628-7 requires only one metal-to-metal seal but a decision was made to incorporate two sealing surfaces due to the movement and cyclic forces induced by real world conditions. The titanium tubes are standard off-the shelf components with threaded ends.

The top and bottom steel flanges may be replaced to match different riser system connections. In the prototype, these are blind flanges with a test port in the lower flange for pressure testing.

1.4 Selection of materials

The selection of materials is a very important aspect of any development work. Especially when conditions dictate that high strength, corrosion resistance and large deflection are required. It is important to find a material that provides adequate bending stiffness while still allowing required bending with reasonable stress and strain distribution. The material must be able to handle the prolonged cyclical loading with a high fatigue life.

Another important aspect to consider in the selection of materials is the corrosive environments it will be subjected to. As the MSJ will be in contact with seawater, possible sour well conditions and other metals, material selection is extremely important. The MSJ is also prefabricated and materials should be selected to perform adequately in all relevant conditions.

The National Association of Corrosion Engineers (NACE) provides a standard that look at materials that will be in contact with sour wells. The most important standard for this work is the NACE MR0175 also known as ISO15156- 2009. The standard is titled "Petroleum and natural gas industries -Materials for use in H_2S-containing environments in oil and gas production".

Titanium is a natural choice when looking at products that require large elastic bending. This is due to the titanium's low Young`s Modulus of about half of regular steel. The use of a titanium alloy with enough molybdenum added makes the material NACE compliant for work in sour well conditions. Cheaper and more readily available materials like steels and nickel alloys may be used in the flanges to keep costs down.

The prototype was machined from high strength titanium and a steel alloy named AISI 4145. The material certificates for the steel may be found in Appendix C. The specific titanium grade used for the prototype and its material certificates have been redacted due to an ongoing patent application.

2. State of the Art

2.1 Introduction

There are currently a few different products either available on the market or under development to perform the purpose of a stress joint. That is the FlexJoint (Oil States Industries 2004), the steel or titanium Tapered Stress Joint (Peacock 1996) and the Shrink Fit Stress Composite Joint (Brett, Jan and Luffrum 2010). The FlexJoint is a modified ball joint developed and made by Oil States Industries. The steel or titanium stress joint is made by many companies in various designs. The Shrink-Fit Stress Composite Joint is under development by Subsea Riser Products Ltd.

All three of the mentioned products are custom designed for each application to perform under a certain set of conditions. Their advantages and disadvantages will be briefly discussed in the next subchapters. The technologies they represent are considered the competition technology for the Modular Stress Joint.

2.2 FlexJoint

The FlexJoint was developed by Oil States Industries over 25 years ago. The system was introduced as a "flexible, frictionless and maintenance free-" (Groves, et al. 2010) system to be installed between the moving riser and the stationary BOP stack. The FlexJoint utilizes an elastomeric seal and flex element design that allow for large angular deflection. This summary is based on info from (Groves, et al. 2010).

The FlexJoint connection is a system that allows relatively large angular deflection over a short radius. The FlexJoint may best be described as a ball joint with a flex element constructed from an elastomeric material. The flex element in the FlexJoint is utilized as the mechanism to absorb the bending moment induced from the riser. One of the limiting factors of the FlexJoint connections is this elastomeric flex element. The Subsea FlexJoint provided by Oil States Industries is currently limited to 6000 psi[1] well pressure conditions. The elastomeric seal design also limits what well fluids / chemicals and temperatures that the FlexJoint may operate with. An advantage of this design is how the elastomeric seal will help damp out mechanical vibrations in the system.

The FlexJoint design leads to an abrupt direction change in the internal bore between the riser and the subsea tree. This change of direction may be up to 10 degrees. This can potentially cause problems for tools or drilling operations through the riser stack. The wear on equipment traveling through the riser FlexJoint and on the FlexJoint itself must be considered when designing and choosing a solution. A 3-D illustration of the FlexJoint may be seen in Figure 2.2.1 below.

Figure 2.2.1: FlexJoint illustration (Groves, et al. 2010)

[1] 6000 psi = 41,37 MPa =413,7 Bar

Recent investigations into a leaking FlexJoint from a Gulf of Mexico TLP in 2004 has lead to a redesign of the FlexJoint. A detailed sketch of the FlexJoint may be seen in Figure 2.2.3 on the following page. The forces in operational use were seen to be larger than expected and not to be incorporated in the current design. This was specifically important with regards to the axial loading and thermal loads and their effects on the elastomeric flex element. The investigation discovered that the well conditions in the pressurized system were pulsating both with pressure and temperature over a 15-60 min period. This effect was drastically reducing the fatigue life of the elastomeric Flex element. Changes to the flex element were proposed and accepted as a solution to this problem. The following Table 2.2.2 was later generated to illustrate the improvements in the new design.

Table 2.2.2: Improvements in fatigue life with redesign of elastomer (Groves, et al. 2010)

Flex Joint Design		Elastomer Fatigue Life Claculations [Years]		
		Original life	Original life +20k cycles of $\pm 200psi$	Original life +20k cycles of $\pm 300psi$
Oil Export	Original design	130 444	139	19
	Redesign	201 563	1 293	218

Table 2.2.2 shows how pulsating pressure effects will dramatically reduce the fatigue life of a FlexJoint. The table also shows how the redesign has increased the fatigue life in all three categories. The increase in fatigue life with the redesign comes with an increase in joint stiffness. This tradeoff must be incorporated in a global riser analysis before replacing an original design with a redesign. Large pressure changes will dramatically reduce the fatigue life and in such cases be a large disadvantage for the FlexJoint.

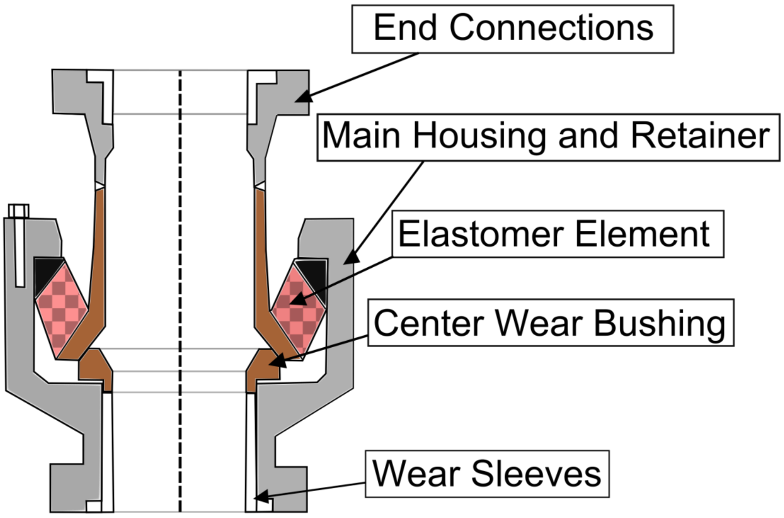


Figure 2.2.3: Illustration of FlexJoint design based on (Oil States Industries 2004)

2.3 Tapered stress Joint

The tapered titanium stress joint was first introduced and installed offshore in 1987. The development work had been performed by Cameron Houston on a one third scale model. The model was tested under 100 year North Sea wave conditions and the results provided confidence in this application of titanium. The first full scale titanium stress joint was installed on the Green Canyon field for Placid Oil in the Gulf of Mexico. The stress joint was installed on the field for 2 years and was subject to 100 year wave conditions in the 2 year period. The stress joint survived undamaged and was later installed on a similar field (Peacock 1996).

The tapered stress joint may be produced in either steel or titanium. The stress joint comprise of a lower flange that connects to the subsea equipment, a long tapered sections and a top connector to the riser stack. The design will in general be the same but the inherent differences in material properties between steel and titanium will make room for some modifications. As titanium has a lower modulus of elasticity than steel, the titanium will allow more bending and better fatigue life than a steel design. Since the titanium can bend more easily, it may be shorter than an equivalent steel design while still providing the same amount of deflection.

The following list provides situations where titanium would be the favored material compared to the stiffer, heavier and less corrosion resistant steel tapered stress joint. The list is found in (Stainless Steel World 2010).

- Vessel motions are great and sea state severe
- Fatigue and bending stresses are a potential problem
- Vortex induced vibration fatigue is a potential problem
- Weight loading on the vessel or platform is critical
- Shallow water place high bend loads on steel riser
- Corrosive and hot brines and sour fluids are produced

The most noticeable with the stress joint is the tapered profile. An illustration of a welded and single cast stress joint may be seen in Figure 2.3.1 on the following page. The tapered design is found to be the best way of reducing the localized bending stresses found between the subsea tree and the riser. One of the large cost drivers for the Tapered Stress

Joint is the machining of this tapered section. It is very important that the machining is flawless as not to induce stress concentrations.

The stress joints are manufactured with different techniques according to which material is being used. The steel stress joint may be forged as a solid part or more frequently as a long tapered section with welded-on flanges. The design is similar but the welded stress joint will often be longer than an equivalent forged solid part. This is done because the weld will become the weakest link and one wishes to move the stresses away from the weld. An example of this may be seen in Figure 2.3.1. This is generally done by having the weld in a straight section and moving the tapered section further up and thus lengthening the stress joint.

Titanium on the other hand is not as easily welded. Titanium is more sensitive against the heat from welding. Especially because the welds will lower the metals strength and fatigue life as compared to with steel. Titanium stress joints may also be forged as solid pieces but this becomes very costly and time consuming (Avery, et al. 1995).

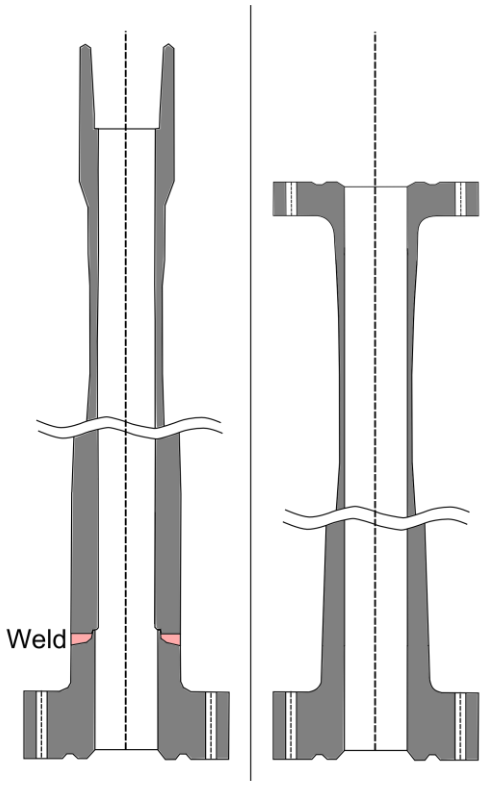


Figure 2.3.1: Illustration of welded and upset forged steel and titanium stress joint

2.4 Shrink-Fit Stress Composite Joint

The Shrink-Fit Stress Composite Joint is a proprietary technology developed by Subsea Riser Products Ltd (SRP). The general design idea is to use both steel and titanium in a stress joint with shrink-fit connections. This section is in large based on SRPs article written by (Brett, Jan and Luffrum 2010).

The Shrink-Fit Stress Composite Joint is a modified tapered stress joint utilizing both titanium and steel. Subsea Riser Products Ltd is currently developing technology to shrink fit steel flanges to titanium pipe. This development is driven by titanium cost versus steel and that this design development will allow the use of both steel and titanium. The design comprise of a straight titanium tube and two steel flanges. An illustration may be seen in Figure 2.4.1 on the next page. As titanium tubes are cheaper and more readily available than custom forgings, this option becomes attractive. The flanges will be heated during assembly, pushed onto the titanium tube and then allowed to cool and shrink-fit onto the titanium pipe. This design will then use the titanium for its better modulus of elasticity in the high stress areas and the steel in the low stress flanged areas.

The use of shrink-fit connections is a relatively new design in riser fabrication. Welding titanium and steel together is not possible because of the differences in material properties. Subsea Riser Products believe that the shrink-fit of steel and titanium would allow for a connection between the materials that are just as good as a weld.

Subsea Riser Products list the following advantages of their technology as compared to having a regular tapered titanium stress joint (Brett, Jan and Luffrum 2010).

- Reducing lead time due to improved sourcing of titanium pipe or smaller forgings only
- Reduced raw material cost due to thick, large diameter flange sections being constructed from steel
- No titanium welding required
- Fasteners and sealing gaskets can be standard and do not have to overcome galvanic coupling issues
- Flange sealing surfaces can be CRA (Corrosion-resistance alloy) weld inlaid
- Reduced complexity in interfacing with surrounding riser components that will generally be constructed from steel.

There are some disadvantages with the shrink-fit approach. They are mainly related to the assembly of the different metals. The temperature required to expand the steel flanges enough for it to pass over the titanium pipe is higher than the heat treatment temperature for the titanium. As the steel flange is placed on the titanium tube, it will start heating the titanium tube as it is cooling. The combination of this and the lower thermal conductivity of titanium compared to steel could adversely affected the strength of the titanium. Further development is required to understand this effect.

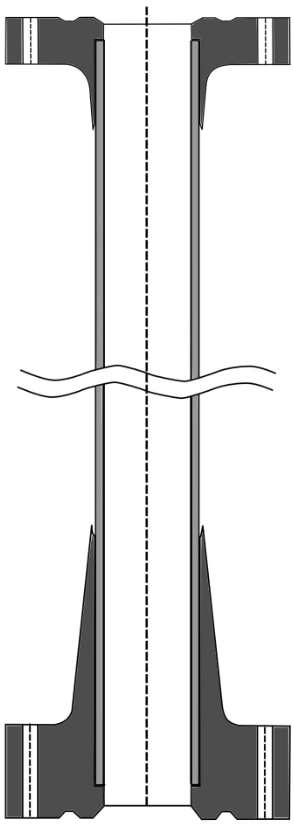


Figure 2.4.1: Illustration of Shrink-Fit Stress Composite Joint

3. MSJ Prototype

3.1 Introduction

This Chapter is divided into several subchapters each involving a step in the process of testing a prototype of the modular stress joint. The subchapters represent the natural process from hand calculations through testing and commenting on data. The subchapters are listed below:

- Chapter 3.2: Calculations using Relevant Standards
- Chapter 3.3: Finite Element Analysis
- Chapter 3.4: Testing
- Chapter 3.5: Strain Gauges
- Chapter 3.6: Test data
- Chapter 3.7: Conclusion from prototype test data

The prototype was scaled to 38% of a full size MSJ. This was done to match a standard 3,5[2] inch OD titanium pipe. The wall thickness has been altered to keep hoop stresses equal between full size and prototype size. This was performed with Finite Element Analysis in Abaqus.

Some key Mechanical and Material properties and Testing parameters are listed on the next page in Table 3.1.1-3.

[2] 3,5 in = 88,9mm

3.1.1 Mechanical properties of prototype MSJ

The relevant mechanical properties of the prototype are listed in Table 3.1.1

Table 3.1.1: Mechanical Properties of prototype MSJ

Symbol	Description	Imperial	SI
L	Length of prototype MSJ	116 inch	2,94 meter
D_o	Outside diameter of titanium tube	3,53 inch	89,66 mm
D_i	Inside diameter of titanium tube	2,869 inch	72,87 mm
t	Wall thickness of titanium tube	0,3305 inch	8,39 mm

3.1.2 Material Properties of prototype MSJ

Typical material properties for high strength titanium used in prototype MSJ are listed in Table 3.1.2

Table 3.1.2: Typical Material Properties for High strength titanium

Symbol	Description	Imperial	SI
σ_Y	Yield strength of titanium tube	155 ksi	1068 MPa
σ_U	Tensile strength of titanium tube	165 ksi	1137 MPa
E	Modulus of Elasticity for titanium	16'500 ksi	113,4 GPa
v	Poisson Ratio for titanium	0,33	0,33

3.1.3 Testing Parameters for prototype MSJ

The relevant testing parameters for the prototype MSJ is listed in Table 3.1.3

Table 3.1.3: Testing Parameters for prototype MSJ

Symbol	Description	Imperial	SI
$P_{i,test}$	Internal pressure at test	Max 15 000 psi	Max 103,4 MPa
$P_{i,work}$	Internal pressure at work	Max 10 000 psi	Max 68,95 MPa
P_e	External pressure during testing	14,5 psi	0,1 MPa
x	Max bending of top flange	5,38 inch	136,65mm
$F_{tension}$	Force in tension	Max 3500 lbs	Max 15600 N
$F_{bending}$	Force in bending	Max 1100 lbs	Max 4900 N

3.2 Calculations using relevant standards

3.2.1 Relevant Standards

There are several relevant Standards within subsea equipment and riser systems. The most comprehensive of them is ISO 13628-7. The problem with this standard is that it explicitly states that it only covers low alloy carbon steels and not materials like titanium. The natural replacement for this standard is DNVs Recommended Practices "Design of Titanium Risers" DNV-RP-F201. The recommended practice refers frequently to DNVs Offshore Standard "Dynamic Risers" DNV-OS-F201. The following calculations will use DNV-RP-F201 as the guiding document and DNV-OS-F201 as a reference document.

Any assumptions and uncertainties in the calculations will be noted as such for each equation. The equations found in the DNV standard are developed for straight pipe members. The worst-case calculations would be given by using the smallest pipe wall thickness. The wall thickness has been given as a set figure with no corrosion allowance as the Modular Stress Joint will not be operating for extended periods without inspection.

3.2.2 Burst pressure design

The Burst pressure design has been calculated in accordance with DNV-RP-F201 Section 5 chapter E200. This chapter states that "Pipe members subjected to net internal overpressure shall be designed to satisfy the following conditions at all cross sections:".

$$(p_{in} - p_e) \leq \frac{p_b(t_1)}{\gamma_m \gamma_{SC}} \tag{3.2.1}$$

p_{in} = Local incidental pressure = 110% design pressure = $1{,}1 * 68{,}98 \text{ MPa} = 75{,}85 \text{ MPa}$

$p_b(t_1)$ = Burst resistance calculated with wall thickness $t_1 = 8{,}39 \text{ mm}$ (Table 3.1.1)

γ_m = Material resistance factor from Table 5-2 in DNV-RP-F201, Set at 1,15 for ULS/ALS

γ_{SC} = Safety class resistance factor from Table 5.1 in DNV-RP-F201. Set at 1,26 for High

Equation 5.10 in DNV-RP-F201 for burst resistance is as follows:

$$P_b(t) = \frac{2}{\sqrt{3}} * \frac{2t}{D_O - t} \min\left(\sigma_Y; \frac{\sigma_U}{1{,}15}\right) * \alpha_{tb} \tag{3.2.2}$$

α_{tb} = Titanium material correction factor (from Table 5-5 in DNV-RP-F201) Set at 1,0 for Burst (Burst factor is assumed to be the same for titanium and steel).

σ_Y = 1068 MPa and $\frac{\sigma_U}{1{,}15} = \frac{1137}{1{,}15}$ = 988,7 MPa. --> $\frac{\sigma_U}{1{,}15}$ is the lowest variable.

$$P_b(t) = \frac{2}{\sqrt{3}} * \frac{2 * 8{,}39 \text{ mm}}{89{,}66 \text{ mm} - 8{,}39 \text{ mm}} * \frac{1137 MPa}{1{,}15} * 1{,}0$$

$$P_b(t) = 2{,}357 * 10^8 \text{Pa} = 235{,}718 \text{ MPa} = 34'187 \text{psi}$$

Inserting all numbers into Equation 3.2.1

$$(75{,}85 \text{ MPa} - 0 \text{ MPa}) \leq \frac{235{,}718 \text{ MPa}}{1{,}15 * 1{,}26}$$

$$75{,}85 \text{ MPa} \leq 162{,}67 \text{ MPa}$$

PASS

3.2.3 Combined Loading criteria

It is stated in DNV-RP-F201 Section 5 E500 that "Pipe members subjected to bending moment, effective tension and net internal overpressure shall be designed to satisfy the following equation:". The following equation assumes that the entire prototype MSJ is made of titanium tube as this is the thinnest cross section and thus the weakest link.

$$\gamma_c\{\gamma_m\gamma_{SC}\}\left\{\left(\frac{|M_d|}{M_k}\sqrt{1-\left(\frac{p_i-p_e}{p_b(t_2)}\right)^2}\right)+\left(\frac{T_{ed}}{T_k}\right)\right\}+\left(\frac{p_i-p_e}{p_b(t_2)}\right)^2 \leq 1 \qquad \textbf{(3.2.3)}$$

M_d = Design bending moment = $F_{bending} * L = 14406$ Nm (Table 3.1.3)

T_{ed} = Design effective tension = 15600 N (Table 3.1.3)

p_i = Local internal design pressure = $68{,}95$ MPa (Table 3.1.3)

γ_c = Condition factor for bending, torsion and internal overpressure (from Table 5-6 in DNV-RP-F201), set at $0{,}95$ for Normal

M_k = Plastic bending moment resistance

$$M_k = \sigma_y * \alpha_{tm} * \alpha_c * (D-t_2)^2 * t_2 \qquad \textbf{(3.2.4)}$$

α_{tm} = Titanium Material correction factor (from Table 5-5 in DNV-RP-F201)

$$\alpha_{tm} = 1{,}1 - \frac{\mathrm{D}}{150t_2} \qquad \textbf{(3.2.5)}$$

$$\alpha_{tm} = 1{,}1 - \frac{89{,}66\ \mathrm{mm}}{150 * 8{,}39\ \mathrm{mm}} = 1{,}0287 \qquad \textbf{(3.2.6)}$$

α_c = A parameter accounting for strain hardening and wall thinning given by:

$$\alpha_c = (1-\beta) + \beta\frac{\sigma_\mathrm{u}}{\sigma_Y} \qquad \textbf{(3.2.7)}$$

$$\beta = 0{,}4 + q_h \quad \text{for } D/t_2 < 15 \qquad \textbf{(3.2.8)}$$

$$q_h = \frac{p_{ld}-p_e}{p_b(t_2)} * \frac{2}{\sqrt{3}} \quad \text{for} \quad p_{ld} > p_e \qquad \textbf{(3.2.9)}$$

$$q_h = \frac{68{,}95\ \mathrm{MPa} - 0}{235{,}718\ \mathrm{MPa}} * \frac{2}{\sqrt{3}} = 0{,}3377$$

$$\beta = 0{,}4 + 0{,}3377 = 0{,}7377$$

$$\alpha_c = (1-0{,}7377) + 0{,}7377 * \frac{1137}{1068} = 1{,}04766$$

$$M_k = 1068\ \text{MPa} * 1{,}0287 * 1{,}04766 * (89{,}66\ \text{mm} - 8{,}39\ \text{mm})^2 * 8{,}39\ \text{mm} \quad \textbf{(3.2.10)}$$

$$M_k = 63782\ \text{Nm}$$

T_k = Plastic axial force resistance

$$T_k = \sigma_y * \alpha_c * \pi * (D - t_2) * t_2 \quad \textbf{(3.2.11)}$$

$$T_k = 1068 * 1{,}04766 * \pi * (89{,}66\ \text{mm} - 8{,}39\ \text{mm}) * 8{,}39\ \text{mm}$$

T_k =2’396’810 N = 2396,8kN

Inserting all variables into equation 3.2.3

$$0{,}95\{1{,}15 * 1{,}26\}\left\{\left(\frac{|14628\ \text{Nm}|}{63782\ \text{Nm}}\sqrt{1 - \left(\frac{68{,}95\ \text{MPa} - 0}{235{,}718\ \text{MPa}}\right)^2}\right) + \left(\frac{15556\ \text{N}}{2396{,}8\ \text{kN}}\right)\right\} + \left(\frac{68{,}95\ \text{MPa} - 0}{235{,}718\ \text{MPa}}\right)^2 = 0{,}39633$$

PASS
$\underline{0{,}396 < 1}$

The prototype is designed in accordance with the mentioned standards. As these standards are based on assumptions that are not valid for the entire prototype, further Finite Element Analysis of the prototype is required. This is performed in Chapter 3.3

3.3 Finite Element Analysis

This chapter consists of both theoretical hand calculations and full computer aided Finite Element Analysis (FEA) with Abaqus software. The FEA was performed by AlTiSS Technologies, an independent engineering company specializing in design and FEA using specialty materials and titanium. The theoretical hand calculations were performed with a number of assumptions as stated below.

3.3.1 Theoretical calculations with pressure and/or tension

The following calculations will be used to theoretically find the stress and strain in the prototype MSJ. A number of assumptions have been made when performing these calculations. These assumptions are based on calculations that are defined for straight section tubes with no variance in geometry or material. Figures 3.3.1.1 and 3.3.1.2 on the next page shows illustrations of the variables used in equations 3.3.1-6.

Assumption 1: The following calculations are based on the thinnest wall section of the prototype, the titanium tubes.

Assumption 2: The following calculations for pressure and tension effect are based on thick-wall assumptions. The calculations are only valid for a thick-walled cylinder with end caps, no temperature effects and for a location far away from the end caps.

Assumption 3: The titanium is assumed to be both isotropic and linearly elastic for the stress-strain relations to be valid.

$$\sigma_{rr} = \frac{P_i a^2 - P_o b^2}{b^2 - a^2} - \frac{a^2 b^2}{r^2(b^2 - a^2)}(P_i - P_o) \quad \textbf{(3.3.1)}$$

$$\sigma_{\theta\theta} = \frac{P_i a^2 - P_o b^2}{b^2 - a^2} + \frac{a^2 b^2}{r^2(b^2 - a^2)}(P_i - P_o) \quad \textbf{(3.3.2)}$$

$$\sigma_{zz} = \frac{P_i a^2 - P_o b^2}{b^2 - a^2} + \frac{P}{\pi(b^2 - a^2)} = Constant \quad \textbf{(3.3.3)}$$

$$\epsilon_{rr} = \frac{1}{E}[\sigma_{rr} - v(\sigma_{\theta\theta} + \sigma_{zz})] \quad \textbf{(3.3.4)}$$

$$\epsilon_{\theta\theta} = \frac{1}{E}[\sigma_{\theta\theta} - v(\sigma_{rr} + \sigma_{zz})] \quad \textbf{(3.3.5)}$$

$$\epsilon_{zz} = \frac{1}{E}[\sigma_{zz} - v(\sigma_{rr} + \sigma_{\theta\theta})] \quad \textbf{(3.3.6)}$$

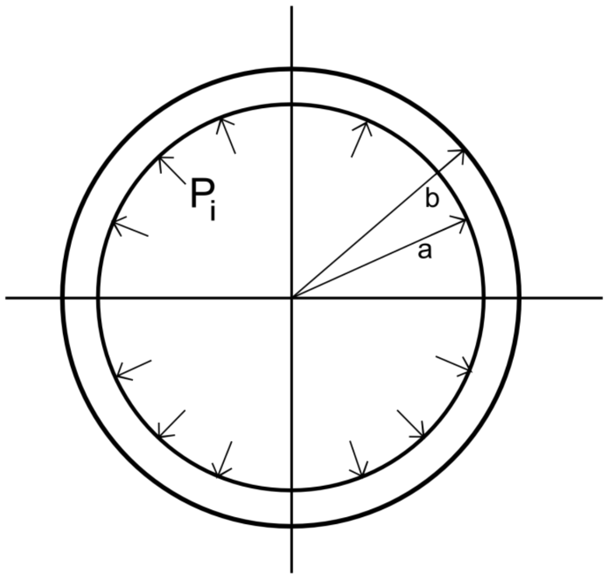


Figure 3.3.1.1: Distribution of internal pressure and variables a and b

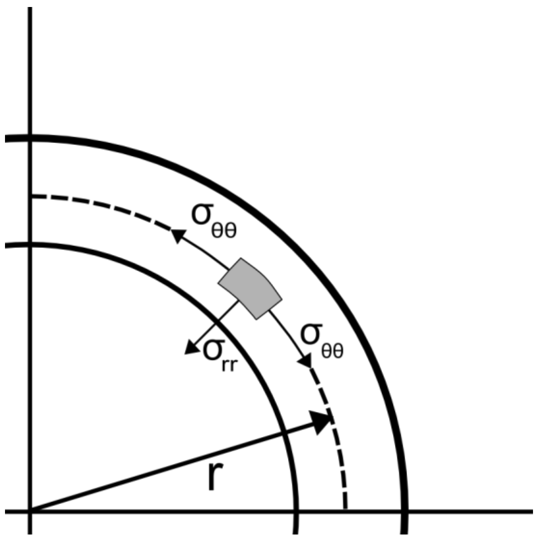


Figure 3.3.1.2: Detail illustration of stress element and variable r

3.3.2 Pressure effects

The first set of equations provides the pressure effects that we will measure in Stage 1 of the testing. The internal pressure has been set at 10355,7 psi (71,4MPa), the highest pressure seen in the testing of Stage 1. This has been done to allow easy comparison between these hand calculations and strain gauge data. There is however no tension or bending moment applied to the prototype during Stage 1 of testing.

The following calculations will be performed to calculate the stress at $r = b = 44{,}83\ \mathrm{mm}$ or at the outer radii surface of the titanium tube, ref equations 3.3.1-6.

$$\sigma_{rr} = \frac{71{,}4\ \mathrm{MPa} * (36{,}44\ \mathrm{mm})^2 - 0{,}1\ \mathrm{MPa} * (44{,}83\ \mathrm{mm})^2}{(44{,}83\ \mathrm{mm})^2 - (36{,}44\ \mathrm{mm})^2} - \frac{(36{,}44\ \mathrm{mm})^2 (44{,}83\ \mathrm{mm})^2}{(44{,}83\ \mathrm{mm})^2((44{,}83\ \mathrm{mm})^2 - (36{,}44\ \mathrm{mm})^2)} * (71{,}4\ \mathrm{MPa} - 0{,}1\ \mathrm{MPa}) = -100000\ \mathrm{Pa} = -0{,}1\ \mathrm{MPa}$$

$$\sigma_{\theta\theta} = \frac{71{,}4\ \mathrm{MPa} * (36{,}44\ \mathrm{mm})^2 - 0{,}1\ \mathrm{MPa} * (44{,}83\ \mathrm{mm})^2}{(44{,}83\ \mathrm{mm})^2 - (36{,}44\ \mathrm{mm})^2} + \frac{(36{,}44\ \mathrm{mm})^2 (44{,}83\ \mathrm{mm})^2}{(44{,}83\ \mathrm{mm})^2((44{,}83\ \mathrm{mm})^2 - (36{,}44\ \mathrm{mm})^2)} * (71{,}4\ \mathrm{MPa} - 0{,}1\ \mathrm{MPa}) = 277605219\mathrm{Pa} = \underline{277{,}6\ \mathrm{MPa}}$$

$$\sigma_{zz} = \frac{71{,}4\ \mathrm{MPa} * (36{,}44\ \mathrm{mm})^2 - 0{,}1\ \mathrm{MPa} * (44{,}83\ \mathrm{mm})^2}{(44{,}83\ \mathrm{mm})^2 - (36{,}44\ \mathrm{mm})^2} + \frac{0\ \mathrm{N}}{\pi((44{,}83\ \mathrm{mm})^2 - (36{,}44\ \mathrm{mm})^2)} = 138'752'609{,}5\mathrm{Pa} = \underline{138{,}75\ \mathrm{MPa}}$$

This provides the following strains on the surface of the material:

$$\epsilon_{rr} = \frac{1}{116{,}52\ \mathrm{GPa}}[-0{,}1\ \mathrm{MPa} - 0{,}33(277{,}6\ \mathrm{MPa} + 138{,}75\ \mathrm{MPa})] = -0{,}001144284\ {}^{\mathrm{m}}/_{\mathrm{m}} = \underline{-1144\ {}^{\mathrm{\mu m}}/_{\mathrm{m}}}$$

$$\epsilon_{\theta\theta} = \frac{1}{116{,}52\ \mathrm{GPa}}[277{,}6\ \mathrm{MPa} - 0{,}33(-0{,}1\ \mathrm{MPa} + 138{,}75\ \mathrm{MPa})] = 0{,}002001648\ {}^{\mathrm{m}}/_{\mathrm{m}} = \underline{2002\ {}^{\mathrm{\mu m}}/_{\mathrm{m}}}$$

$$\epsilon_{zz} = \frac{1}{116{,}52\ \mathrm{GPa}}[138{,}75\ \mathrm{MPa} - 0{,}33(-0{,}1\ \mathrm{MPa} + 277{,}6\ \mathrm{MPa})] = 0{,}000404866\ {}^{\mathrm{m}}/_{\mathrm{m}} = \underline{404\ {}^{\mathrm{\mu m}}/_{\mathrm{m}}}$$

3.3.3 Pressure and tension effects

The second set of equations provides the pressure effects and tension effects as we will measure in Stage 2 of the testing. Tension of 15600 Newton is applied. Only the results of equations 3.3.1-6 are displayed

$$\sigma_{rr} = \underline{-0{,}1\ \mathrm{MPa}}$$

$$\sigma_{\theta\theta} = \underline{277{,}6\ \mathrm{MPa}}$$

$$\sigma_{zz} = \underline{146{,}03\ \mathrm{MPa}}$$

$$\epsilon_{rr} = \underline{-1200\ {}^{\mu\mathrm{m}}/_{\mathrm{m}}}$$

$$\epsilon_{\theta\theta} = \underline{1969\ {}^{\mu\mathrm{m}}/_{\mathrm{m}}}$$

$$\epsilon_{zz} = \underline{467\ {}^{\mu\mathrm{m}}/_{\mathrm{m}}}$$

We will be able to apply strain gauges to the titanium tubes on the MSJ prototype to monitor both $\epsilon_{\theta\theta}\ and\ \epsilon_{zz}$. We are not however able to monitor ϵ_{rr} with strain gauges and we will rely on FEA data instead. The data from the hand calculations are compared with FEA and strain gauge data in Chapter 3.6.

3.3.4 Abaqus FEA

A 3-D model of the prototype MSJ was run through the FEA software Abaqus. This was performed during the fall of 2011 by AlTiSS Technologies. This model was used to find bending, tension and pressure capabilities and also to check small design changes. The Abaqus model was also central in the choice of material for the various parts in the MSJ. The model will not be explained in full detail as it was not generated nor operated by the author. There are however a few items that must be noted.

The modeling was performed with a half symmetric model. The key numbers for the model is presented in Table 3.3.4.1 below. The model was set up with bonded surfaces in the thread areas. This makes the areas much stiffer in the model then would be the case in real prototype testing. This simplification was performed due to time constraints. The bottom flange was modeled as fixed in all degrees of freedom (DOF).

Table 3.3.4.1: Abaqus FEA mesh and elements

Number of elements	56 344
Number of Unknowns	235 377
Instant mesh size	0,25”

The analysis was performed with a two step method. The first step included preloading the bolts on the swivel flange. Step two included adding the internal pressure, tension and bending force.

Figure 3.3.4.2 shows the stress contours for the prototype MSJ while under the influence of internal pressure 68,95MPa (10 000 psi), tension (15 600 N) and bending of 136,65mm (5,38 inches). The max stress is positioned inside of the first connector. The FEA data is compared with the hand calculations and strain gauge data in Chapter 3.6.

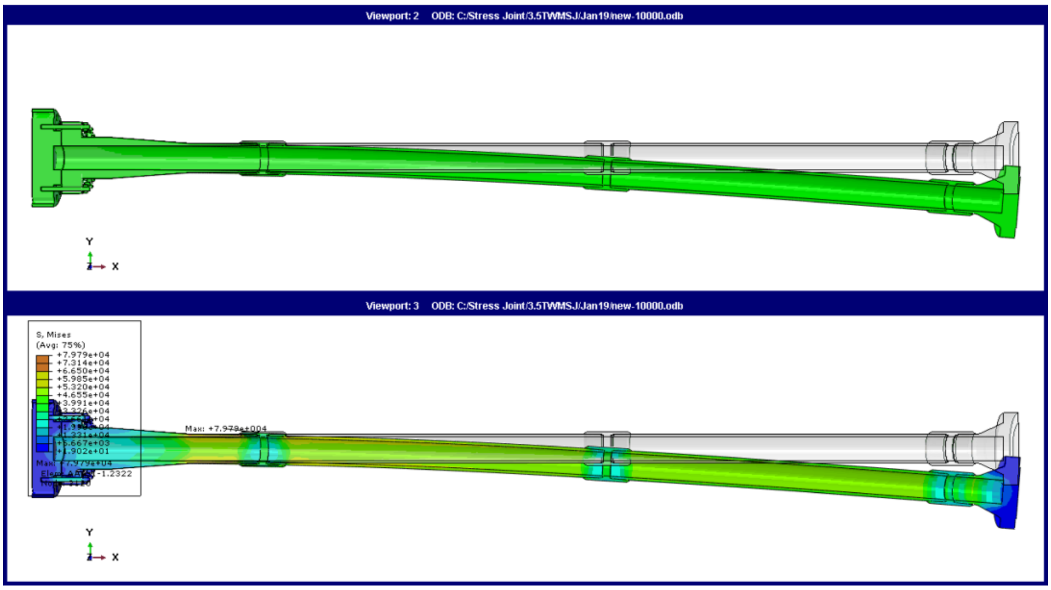


Figure 3.3.4.2: Stress contour of prototype MSJ generated in Abaqus.

3.4 Testing

3.4.1 Introduction

Testing of the prototype Modular Stress Joint will be performed to confirm the general design and function of the stress joint. The scale model is a 1/3 scale prototype of the full size MSJ design with a few exemptions. The final thread and seal design have not been chosen, the connections thread and seal design for this prototype will thus not represent the full size stress joint. The connections are designed for all forces it may see from the test procedure.

3.4.2 Testing procedure for prototype

The prototype is to be qualified for general design and functionality as noted above. The threads and sealing chosen shall allow us to perform all the required testing but is not representative of the modular stress joint final design.

The prototype will be tested in a rig set up at Subsea Technologies Ltd. in Aberdeen, Scotland. The test rig will allow the prototype to be fixed at its base flange while forces in both axial direction (tension) and deflection (vessel offset) may be applied at its top flange as seen in Figure 3.4.2.1. The prototype will also be pressure tested both individually and while under loading. This is done to investigate the prototype performance changes related to internal pressure, tension changes and bending.

Strain gauges will be applied to the MSJ prototype to monitor critical strain locations and a deflection measurement system will be installed. Curves of the prototype performance in deflection may then be generated by controlling internal pressure and forces in both directions.

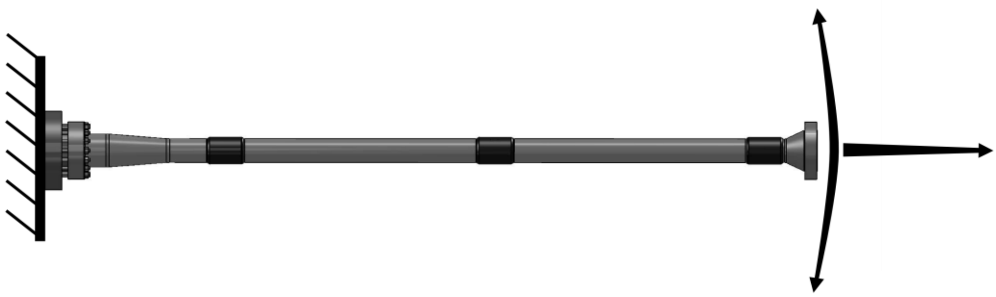


Figure 3.4.2.1: External forces to be applied to prototype through test procedure

3.4.3 Test Procedure Stages

The first stage will be a hydrostatic pressure test to investigate pressure integrity. Liquid leak-tightness test will be performed as stated in ISO13628-7:2006 Section 6.4.11.5. Standards also specify that testing shall be performed at 150% of working pressure. A test procedure for this is found in Appendix A, Stage 1. Strain gauges will be applied to monitor hoop strains during testing.

The second stage will be to induce external loading forces into the prototype. This includes gradually applying tension to check that the test rig and prototype perform within safe parameters. Strain gauges will be placed on the prototype to verify FEA work and to control strain in specific areas. Leak tightness testing should also be included to verify seals and connectors with variable tension forces. A test procedure is found in Appendix A, Stage 2.

The third stage will be to gradually include bending into the prototype. Strain gauges will be located at critical areas on the prototype to verify FEA data. Leak tightness test will also be included to verify seals in connectors. A test procedure is found in Appendix A, Stage 3.

The fourth stage will include both tension and bending. Both tension and bending will gradually be increased to verify the performance of the prototype. Strain gauges will be connected to the prototype to monitor the strains in the material. Leak tightness tests will be performed as the internal pressure is increased. A test procedure is found in Appendix A, Stage 4.

Conducting the full test procedure will result in a hydrostatic pressure test to test pressure and 12 cycles to working pressure. A summary of forces applied in the various testing stages are found in Table 3.4.3.1.

Table 3.4.3.1: Overview of testing stages for prototype MSJ

Yes/No	**Stage 1**	**Stage 2**	**Stage 3**	**Stage 4**
Pressure	**Y**	**Y**	**Y**	**Y**
Axial force	N	**Y**	N	**Y**
Bending force	N	N	**Y**	**Y**

3.4.4 Failure modes for testing

The Failure modes listed in Table 3.4.4.1 are found in ISO 13628-7:2006. They list a set of failure modes for general completion/workover riser systems and riser connectors. This list will be utilized as a checklist to look at the risks involved in the testing of the prototype stress joint. Some of these failure modes are relevant and some are not due to the controlled testing conditions compared to real world conditions. The likelihood and any risk reducing measures are set by reasoning and discussion by personnel involved in the testing. The "toolbox talk" (HSE meeting) before testing identified no further failure modes for the testing. All test equipment had certified high-pressure equipment.

Table 3.4.4.1: Failure modes for general C/WO riser system [ISO 13628-7:2006]

Failure mode	Likelihood	Risk reducing measures/Reason
Excessive yielding	Low	FEA and limited forces applied
Buckling	Very Low	Not possible in test rig
Fatigue	Very Low	Low forces, cycles and FEA
Brittle fracture	Low	Low forces, low Young`s modulus
Excessive deflection	Low	Limited deflection allowed
Leak-tightness	Low	Step-wise testing in test pit
Corrosion and wear	Very Low	New material
Sudden disengagement	Low	Step-wise testing in test pit
Mechanical function	Low	N/A

3.4.5 Testing in Aberdeen

The testing was performed in Aberdeen at Subsea Technologies Ltd's Donside facility. The testing was initially scheduled for February/March but was delayed until April 30th to May 4th. The prototype modular stress joint was assembled in a torque machine in Aberdeen on April 30th.

During this assembly, potential problems with the thread design were discovered. They were as follows: In one of the connectors the load shoulder did not shoulder with the steel connector as planned. It is believed that this was caused by machining error in the threads (threads not long enough). This caused one of the sealing surfaces to not contact properly.

Another problem was the heat generated from the high interference in the connections. This may have caused potential problems for later break-out of the connection (Galling or heat welding). This did not however effect the testing of the prototype and pressure integrity was maintained as is described below.

The test rig with the prototype MSJ installed may be seen in Figure 3.4.5.1. Stages 1 and 3 of the testing procedure were performed as intended but Stage 2 and 4 were omitted due to missing test equipment. The tools to apply tension were not available at testing time.

Figure 3.4.5.1: Test rig with prototype MSJ installed

3.4.6 Stage 1

Stage 1 comprised of a pressure test from atmosphere pressure to 10 000[3] psi. Stage 1 also included testing to the test pressure of 15 000[4] psi. The testing to 15 000 psi was postponed until all other testing was performed in case of any failures at test pressure. The test certificate with the pressure graph may be seen in Appendix B, test certificate 430. A picture of this setup may be seen in Figure 3.4.6.1.

The pressure was increased in 1000 [5] psi increments. Pressure integrity was confirmed by holding the work pressure of 10 000 psi for a total of 15 min. The pressure loss of 24psi is assumed to be in the hydraulic hoses and test equipment. It might be assumed that this pressure loss diminishes over time and that it levels off after approximately 30-60 minutes.

Figure 3.4.6.1: Pressure test to 10 000 psi with no bending or tension

3.4.7 Stages 2 and 4

Stage 2 and 4 comprised of putting the prototype into tension. The equipment for putting the prototype into tension was not available at the testing time. Stages 2 and 4 are for this reason left untested and recommended for future work.

[3] 10 000 psi = 68,95MPa = 689,5 Bar
[4] 15 000 psi = 103,42MPa = 1034,2 Bar
[5] 1000 psi = 6,895MPa = 68,95 Bar

3.4.8 Stage 3

Stage 3 comprised of putting the prototype into bending and testing it with pressure up to work pressure. The prototype was put into bending by tightening a ratchet between the prototype top flange and the test frame as seen in Figure 3.4.8.1-2. As it was not possible to measure force, the bending measurements were performed with measuring the deflection of the top flange. The bending was increased 1 inch at a time up to the max deflection of 5,5 inches(139,7mm). The internal pressure was raised in 2 500 psi increments to 10 000 psi. The pressure graphs may be seen in Appendix B, test certificates 437-440 and 442. Figures 3.4.8.1 and 3.4.8.2 on the following page shows the deflection at 139,7mm (5,5 inches).

Figure 3.4.8.1: Stage 3 bending of prototype. Max bending at 139,7mm (5,5 inches)

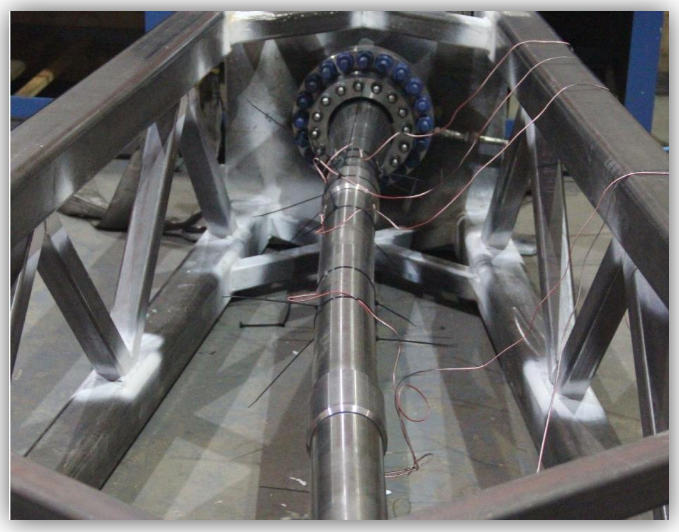


Figure 3.4.8.2: Stage 3 bending of prototype. Max bending at 139,7mm (5,5 inches)

3.4.9 Test Pressure

The test pressure was left as the last item to do. Test pressure is defined in ISO 13628-7 as 1,5 times the work pressure with no external forces applied. The prototype was left in its neutral position and the internal pressure was gradually increased to 15 000 psi. The pressure graph may be seen in Appendix B test certificate 444. It is important to note the pressure drop seen in the graph. Investigations into this pressure leak discovered a loose hose fitting on the test equipment. As no leaks were discovered below the prototype and that the pressure was held for over 15 minutes at test pressure, the test was categorized as successful. No further testing was performed on the prototype during the thesis period of spring 2012.

3.5 Strain gauges

3.5.1 Strain gauge placement and direction

The placement of the strain gauges was made after performing FEA to identify any large strain areas. The strain gauges are tactically located in these areas to verify the FEA data and if possible any other theoretical calculations. Their location is identified in Figure 3.5.1.1. For the pressure effects, we can monitor the hoop stresses with strain gauge 6. This can be checked against both FEA and theoretical calculations. The large strain areas in bending are identified around the area located between strain gauges 2 and 5. This area is of special interest due to change in geometry and material in this area. All strain gauges were installed to measure strain in axial direction except for strain gauge 6 which was installed in transverse direction to measure hoop strain.

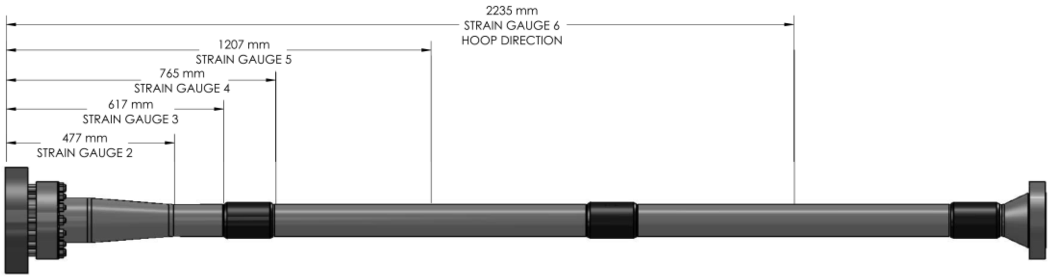


Figure 3.5.1.1: Strain gauge placement on prototype MSJ

3.5.2 Strain gauges

Strain gauges were acquired through Vishay. The key information is the grid resistance of $350{,}0 \pm 0{,}15\%\ \Omega$ and the gauge factor of $2{,}120 \pm 0{,}2\%$. This data was inserted into the strain gauge recording software to interpret the data being recorded. The strain gauges were attached to the prototype stress joint by engineer Cody DeHart from AlTiSS Technologies. The strain gauges were connected to the signal amplifier by the author in a setup named “Single strain gauge using three-wire connection with compensating resistor”. A schematic of this connection may be seen in Figure 3.5.2.1 and a picture of how the wires and compensating resistor were soldered may be seen in Figure 3.5.2.2.

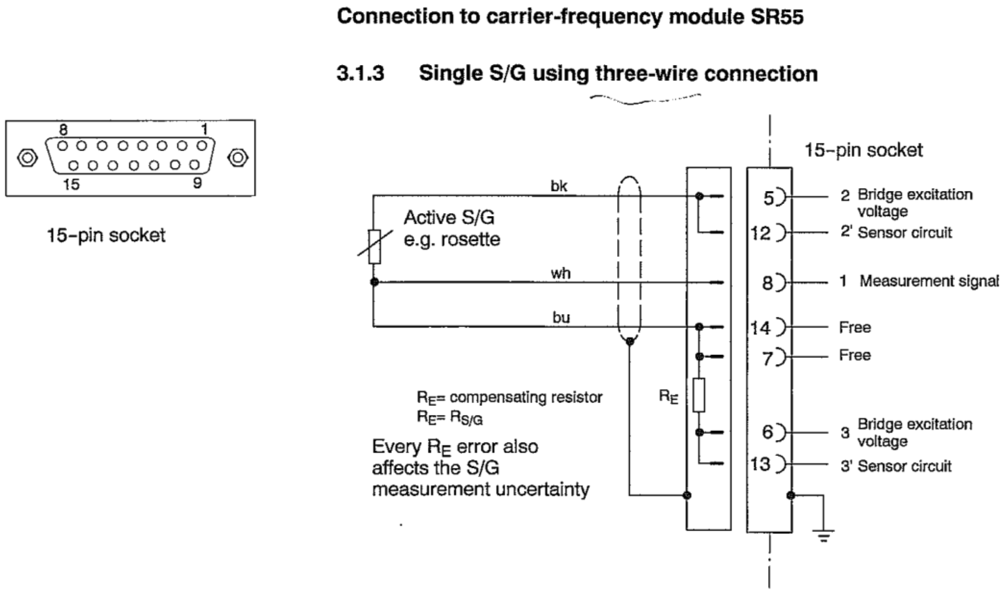


Figure 3.5.2.1: Schematics for connecting Strain gauge to Spider 8 (Hottinger Baldwin Messtechnik u.d.)

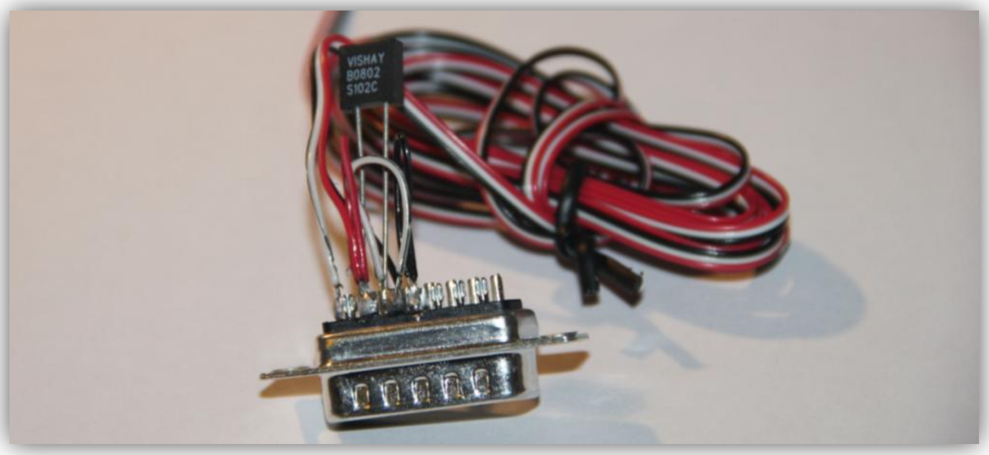


Figure 3.5.2.2: 15 pin connector with soldered on wires and compensating resistor

3.5.3 Signal amplifier and recording software

A system for recording data from all strain gauges was required. A Spider 8 system (SR55 module) from HBM was borrowed from the University of Stavanger. This signal amplifier was connected to a computer running "Catman 4.5" from the same vendor to setup and record the data input. This system allowed both continuous presentation and recording of all strain gauges attached during testing. The data was exported into excel files after each test. The data was further processed and analyzed in Chapter 3.6.

3.6 Test data

3.6.1 Stage 1 - Pressure testing

Figure 3.6.1.1 represents the strains measured in hoop and axial direction for testing performed in Stage 1. The figure represents the strains measured while increasing pressure from atmospheric to 10 000[6] psi from strain gauge 5 (hoop) and 6 (axial). The figure also includes values calculated by hand in Chapter 3.3 and through Abaqus FEA software. Test certificate 432 found in Appendix B shows the pressure profile.

Figure 3.6.1.1 clearly shows how the measured strain in hoop direction is higher than expected through modeling. Strains measured in axial direction however are slightly lower than expected. The hand calculated values in axial direction are covered by the strain gauge data. Table 3.6.1.2 shows the values and deviation between modeled results and actual strain gauge data.

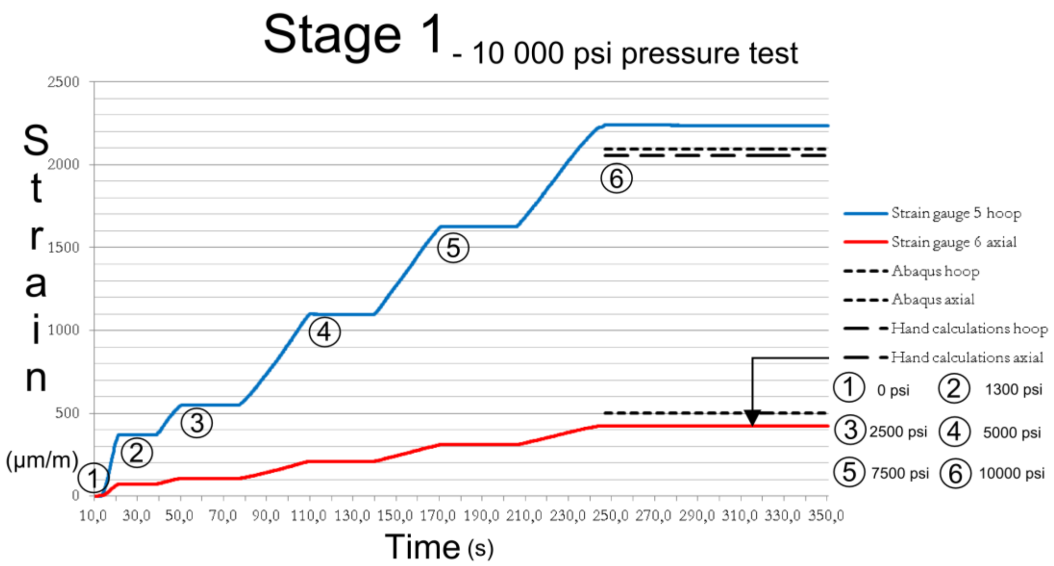


Figure 3.6.1.1: Stage 1 strain gauge measuring hoop strain pressure up sequence

[6] 10 000 psi = 68,95MPa = 689,5 Bar

Table 3.6.1.2: Comparison of strain gauge data and modeled results

	Strain Gauge	Abaqus	Hand calculation	% Deviation to Abaqus	% Deviation to Hand calc
Hoop	2239	2160	2002	-3,53%	-10,58%
Axial	424	500	404	15,2%	-4,7%

The deviations between the numbers are further commented in Chapter 3.7.

3.6.2 Stage 3 – Bending and pressure testing

Stage 3 involves bending and pressurizing the prototype. This includes 5 pressure sequences with deflection varying from 1 to 5,5 inches (25,4 - 139,7mm). The prototype is released from bending into its natural position after each test. A ratchet was used for applying the bending in the prototype. The sensitivity and sample rate of the strain gauge equipment make it possible to see the ratchet process in Figures 3.6.2.1-5. As this process was manual, the time axis if of no concern, it is not possible to correlate and compare the data with time.

The strains for strain gauges 2, 3, 4 and 5 can be seen in the application of bending and release back to neutral position. The procedure of releasing the prototype and strain gauges back to its neutral position is very important as any failures in the strain gauge or residual stresses in the material become visible.

Figure 3.6.2.1 (1 inch deflection) clearly shows how strain gauge 5 moves into a negative area (compression) after bending force is released. This abnormality may be caused by various factors. It is believed that this could have been the result of either slipping in the strain gauge or the result of residual stresses in the material from machining etc.

A similar abnormality is found in Figure 3.6.2.3 (3 inch) where strain gauge 5 again does not zero out after the bending force is released. As the strain gauge now shows a positive result, it is believed that the strain gauge is slipping on the material somehow. A decision was made on the mentioned abnormalities to define data from strain gauge 5 to be corrupt.

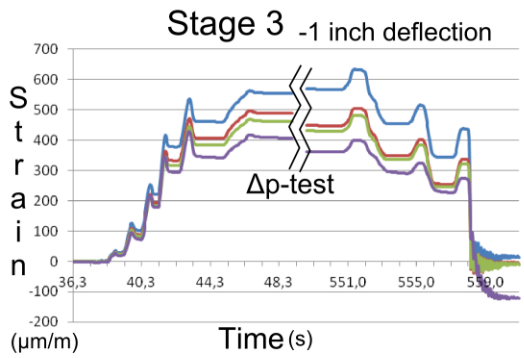


Figure 3.6.2.1: Strain - 25,4mm deflection

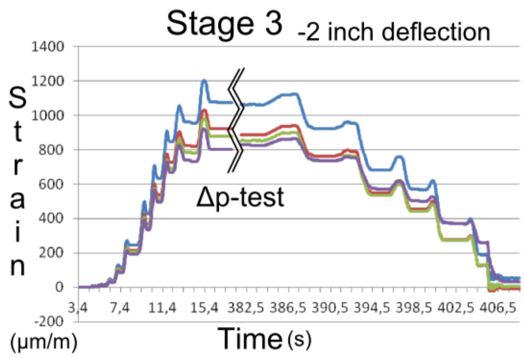


Figure 3.6.2.2: Strain - 50,8mm deflection

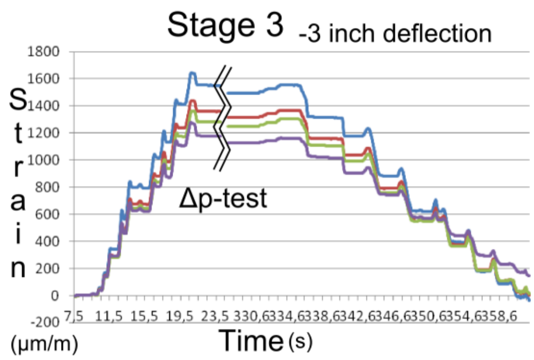


Figure 3.6.2.3: Strain - 76,2mm deflection

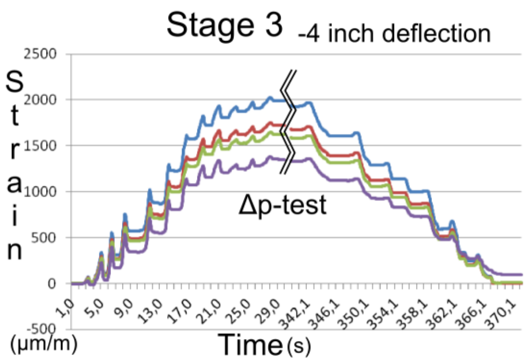


Figure 3.6.2.4: Strain - 101,6mm deflection

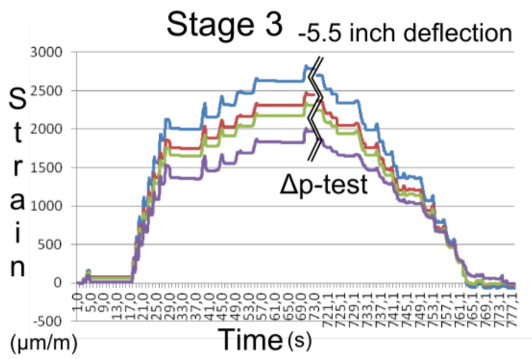


Figure 3.6.2.5: Strain – 139,7mm deflection

After the MSJ prototype had been put into bending, the internal pressure was gradually increased to a max of ~10 000 psi[7]. This was done in pressure increments of 2500 psi except for the first 1300 psi delivered from a low pressure reservoir. The test certificates with pressure graphs may be seen in Appendix B.

Figure 3.6.2.6 shows the strains in axial direction with 1 inch deflection with the strain gauges in tension. The data from strain gauge 5 has been removed as this data was classified as corrupt due to slipping of the strain gauge. It is clearly seen that there is a deviation between modeled results in Abaqus and the measured strain gauge data. Strain gauge 2 is close to modeled data while strain gauge 3 and 4 has larger deviation. The following Figures 3.6.2.6-10 shows the measured and modeled strains with increasing deflection. The data from these graphs were used to generate Table 3.7.1 in Chapter 3.7.

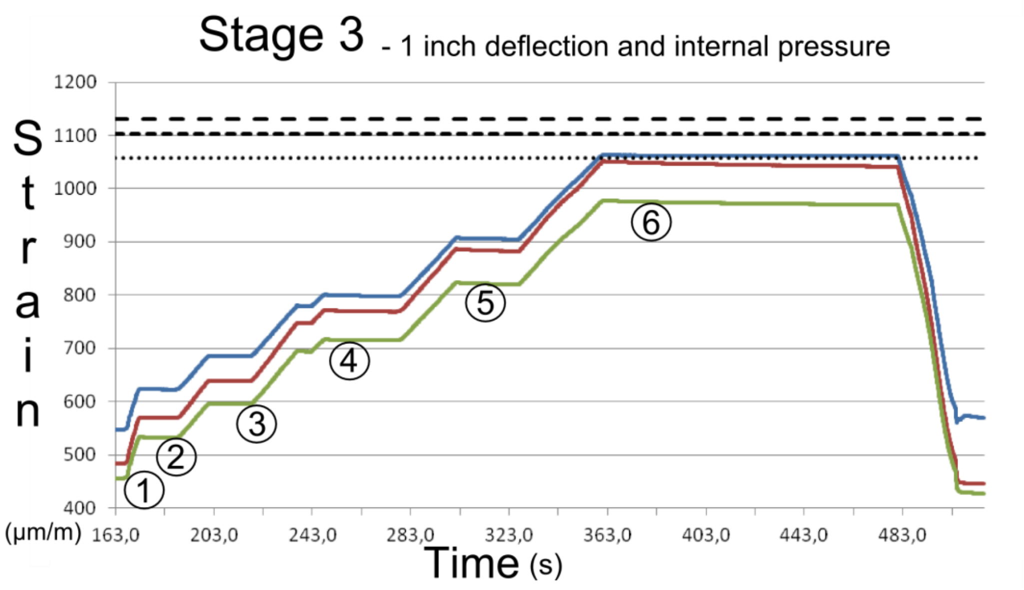
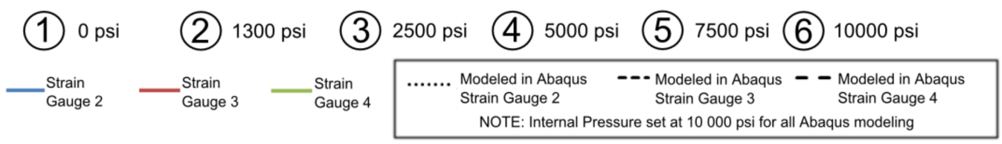


Figure 3.6.2.6: Strains for Stage 3 – 25,4mm deflection – 10 000 psi (68,95MPa)



[7] 10 000 psi = 68,95MPa = 689,5 Bar

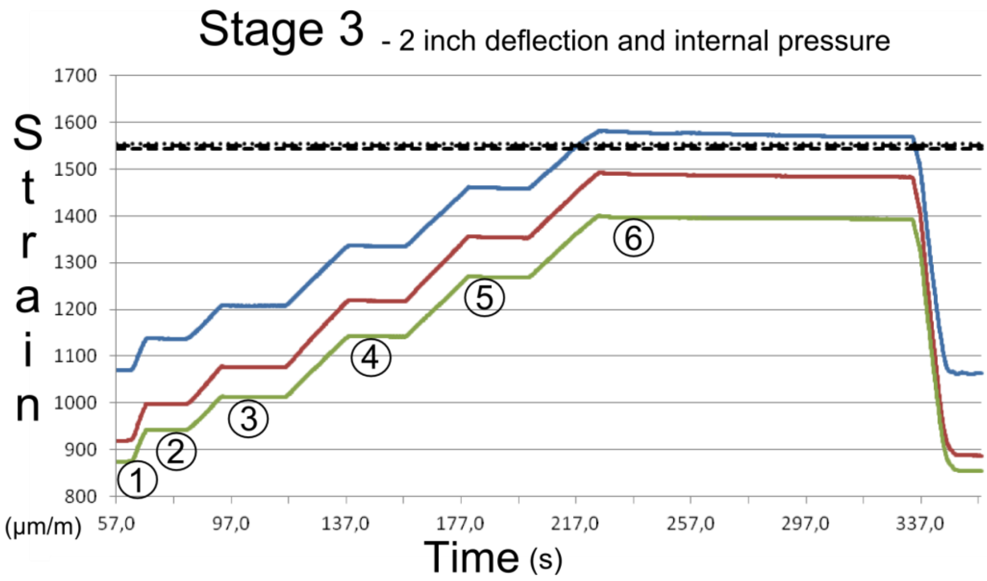


Figure 3.6.2.7: Strains for Stage 3 – 50,8mm deflection – 10 000 psi (68,95MPa)

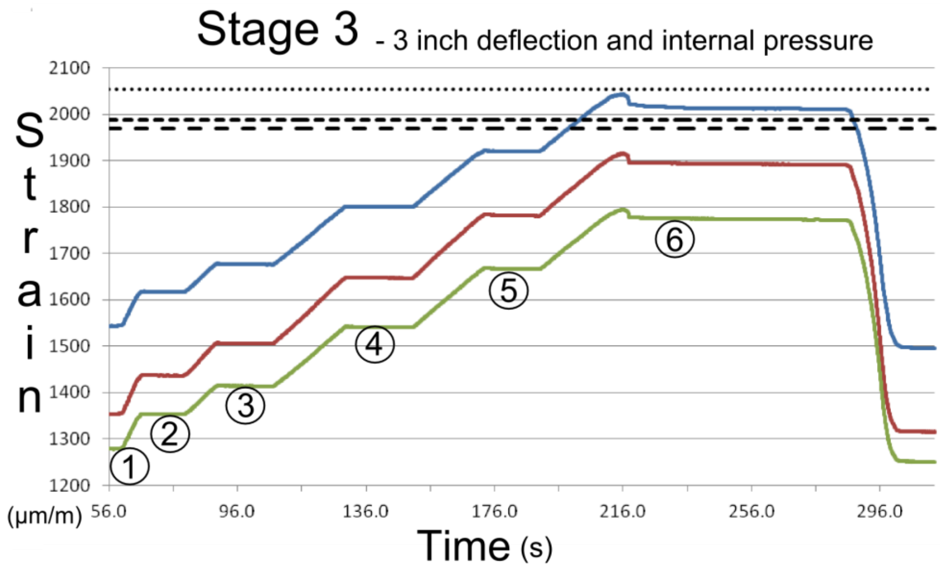
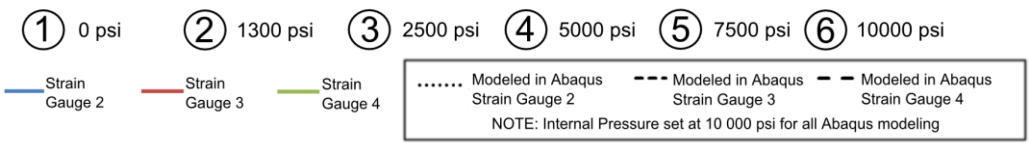


Figure 3.6.2.8: Strains for Stage 3 – 76,2mm deflection – 10 000 psi (68,95MPa)



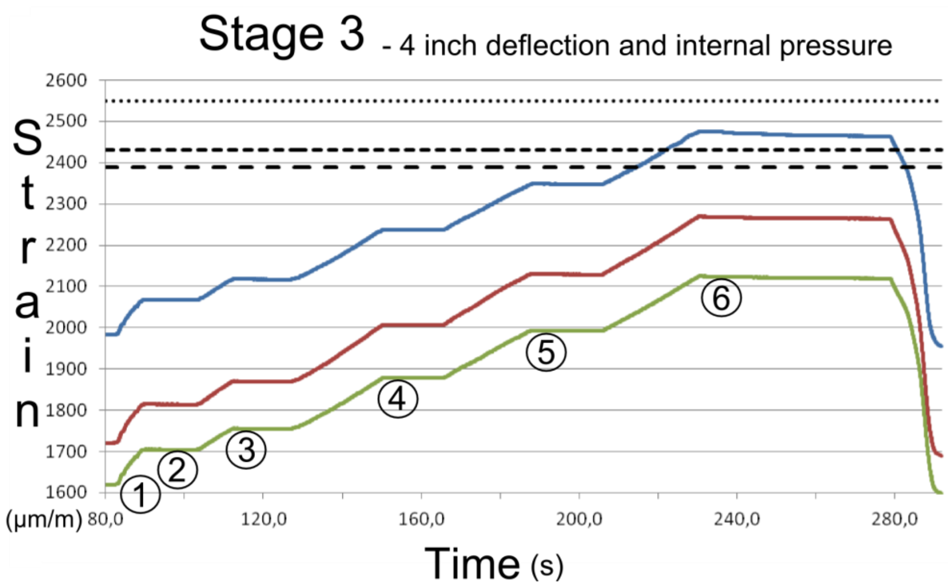


Figure 3.6.2.9: Strains for Stage 3 – 101,6mm deflection – 10 000 psi (68,95MPa)

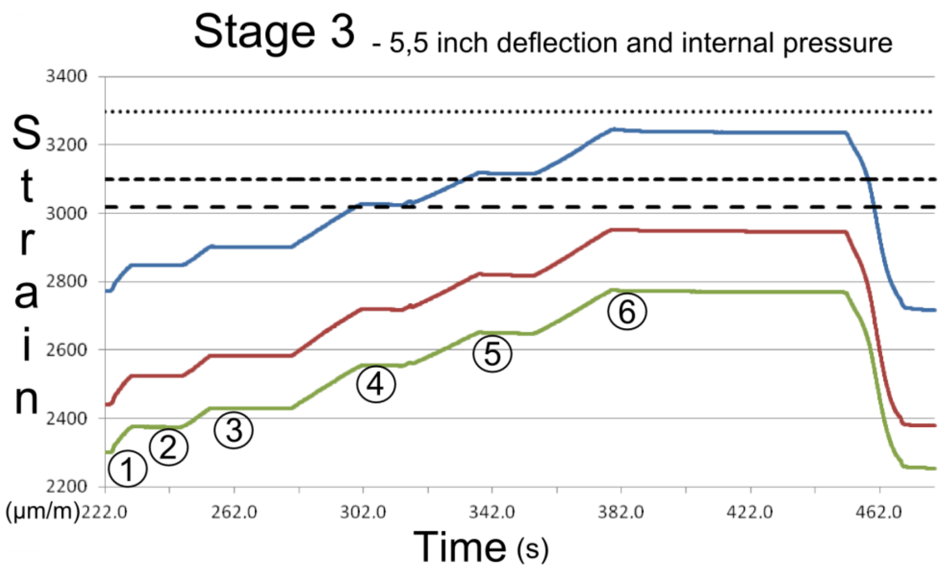
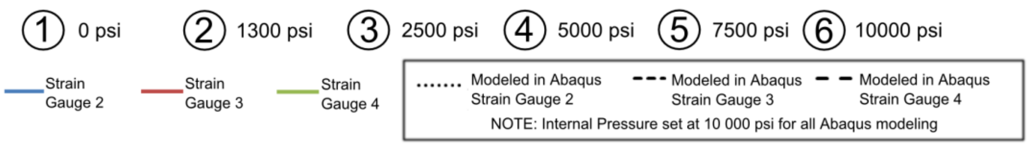


Figure 3.6.2.10: Strains for Stage 3 – 139,7mm deflection – 10 000 psi (68,95MPa)



3.7 Conclusion from prototype test data

The data from strain gauge 2 shows a close correlation between the modeled results and the actual measured data as seen in Table 3.7.1 below. The deviation between the model and prototype varies between +0,5% and -3,0%. Industry practice and the expert opinion of FEA expert Dr Young-Hoon Han at AlTiSS Technologies say that a deviation of less than 10% is usually acceptable. The deviations in this case are accepted because of a reasonable explanation for the deviation was presented.

The deviation is believed to be found in how the FEA model was set, run and how FEA software models threads in bending. The setup and input for the FEA work is noted in Chapter 3.3. The FEA was performed with bonded surfaces in the threaded areas and with the minimum yield for the material properties.

Table 3.7.1: Stage 3 strain gauge and FEA modeled results

	Deflection [**inch**]	**Strain Gauge** [$\mu m/m$]	**Abaqus** [$\mu m/m$]	**Deviation** [%]
Strain Gauge 2	1	1063	1057	0,56 %
Strain Gauge 3	1	1051	1102	-4,6 %
Strain Gauge 4	1	977	1131	-13,6 %
Strain Gauge 2	2	1581	1555	1,64 %
Strain Gauge 3	2	1491	1545	-3,5 %
Strain Gauge 4	2	1398	1551	-9,86 %
Strain Gauge 2	3	2042	2053	-0,53 %
Strain Gauge 3	3	1914	1988	-3,72 %
Strain Gauge 4	3	1793	1970	-8,98 %
Strain Gauge 2	4	2474	2551	-3,0 %
Strain Gauge 3	4	2268	2432	-6,74 %
Strain Gauge 4	4	2123	2390	-11,17 %
Strain Gauge 2	5,5	3245	3298	-1,6 %
Strain Gauge 3	5,5	2951	3098	-4,74 %
Strain Gauge 4	5,5	2774	3019	-8,11 %

It is important to note that (almost) all strain gauge measurements were displaying lower strains then expected from the FEA model. This shows that the model depicts worst case conditions. It is also worth mentioning that the prototype deflections were measured with a tape measurement and some variance in the actual bending of the prototype compared with modeled results is expected.

The testing of the prototype is classified as successful but with a note, as necessary test equipment to perform Stages 2 and 4 was not available at the scheduled test date. The main objective of the testing was to make sure that the prototype behaved like the models predicted and that we had pressure integrity during the test procedure. This was achieved and the subsequent testing to test pressure shows that the design holds test pressure after it was subjected to numerous bending operations.

As the MSJ will be subject to a cyclic and highly variable load in real world conditions, fatigue testing is required. This is scheduled to start the summer of 2012 and continue for an extended period of time. This will be used to show that pressure integrity is still maintained after extended periods of cyclic loading.

4. FEA of full scale MSJ

4.1 Introduction

This chapter concentrates on the use of the Modular Stress Joint (MSJ) in real world environment conditions. This chapter is the next step in the development of the MSJ and the natural analysis to perform after prototype testing reported in Chapter 3. This chapter consists of the analysis of a full size modular stress joint. The dimensions of the MSJ were chosen to match a steel Tapered Stress Joint (TSJ). The steel TSJ was designed for operations in a 80m water depth situation, which has governed the testing and modeling parameters of the MSJ to allow for a comparison with the newly designed and prototype tested MSJ.

The analysis will be based on Abaqus FEA data and Orcaflex global riser modeling and analysis. The prototype analyzed in Chapter 3 provides confidence that the FEA data is good and actually provides higher stresses than expected in real life testing. The FEA data of the MSJ may thus be used as input and operational limits in the Orcaflex model and analysis.

The MSJ in this analysis was developed to match the steel TSJ in bore, length and the design basis in all other variables. This similarity in outer geometry allows easy comparison of performance between the steel TSJ and the MSJ. The dimensions chosen for this analysis do not necessarily represent the dimensions of the finished product as the development of the full size MSJ is still at an early stage. Material selection is equal to prototype MSJ. The relevant mechanical properties and testing parameters are listed below in Tables 4.1.1 and 4.1.2.

Table 4.1.1: Mechanical properties of titanium MSJ

Symbol	Description	Imperial	SI
L	Length of MSJ	308 inch	7,823 meter
D_o	Outside diameter of titanium tube	9 inch	228,6 mm
D_i	Inside diameter of titanium tube	7,375 inch	187,33 mm
t	Wall thickness of titanium tube	0,81 inch	20,63 mm

Table 4.1.2: Testing parameters for stress joints

Symbol	Description	Imperial	SI
$P_{i,test}$	Internal pressure at test	15 000 psi	103,4 MPa
$P_{i,work}$	Internal pressure at work	10 000 psi	68,95 MPa
$F_{tension}$	Force in tension	22 000 lbs	~ 98,1 kN
$F_{bending}$	Force in bending	Max 20 000 lbs	~ Max 89 kN

4.2 The model

The global analysis of the riser system is performed with an Orcaflex model developed by Principal Analysis Engineer Ashley Bird at Subsea Technologies Ltd in Aberdeen, Scotland. The actual use of the model, data acquisition and small modifications during use were performed by the author.

The model consists of a generic vessel that would be used in intervention related operations. The vessel is equipped with a heave compensation system for the connection to the riser. The model includes a steel riser connected to the stress joint and the subsea equipment. The subsea equipment included is the EDP, LMRP, X-mas tree and wellhead. This is a generic layout for intervention related operations on a subsea well. The location of this model is in the North Sea.

4.2.1 Analysis

Modeling and analysis of a global riser system with software like Orcaflex becomes complicated and quickly time consuming. The continuous hurdle is that one wishes to model all contributing effects but still would like to keep computing time at a reasonable level. Simplifications are then introduced to speed up the analysis work and make the work more manageable.

The analysis time is largely controlled by the duration chosen for the simulation. A 3 hour duration was initially chosen for these simulations. As the complexity of the model increased, the computational time went from 4-5 hours into several days and even weeks in some cases. A decision was made to run the 3 hour simulations to identify the largest contributing effects and to use those in a much shorter duration simulation for the model. This greatly reduced the computational time from several days to less than an hour. This is further commented in Chapter 4.4.

4.2.2 Environment

The environmental variables in Orcaflex may be set to model virtually any conditions, like waves, current, air and water pressure, salinity, humidity etc. Some of these variables have been manually set based on the conditions we are trying to match while others are left as standard values. The wave conditions used in this analysis are based on JONSWAP (JOint

North Sea WAve Project) conditions. JONSWAP wave conditions are widely recognized as the best approximation to real world conditions in the North Sea (K.Chakrabarti 2005).

The significant wave height (H_S) discussed is the average height of the top third largest measured waves over a period. The relationship for a 3 hour storm as we are using here is:

$$H_{MAX} = 1{,}86 * H_S \quad \textbf{(4.2.1)}$$

Figure 4.2.2.1 below represents the wave profile used to identify the highest wave in a 3 hour storm condition. The parameters used to generate this are found in Table 4.2.2.2 below. The other parameters in the JONSWAP wave were calculated by Orcaflex based on these inputs.

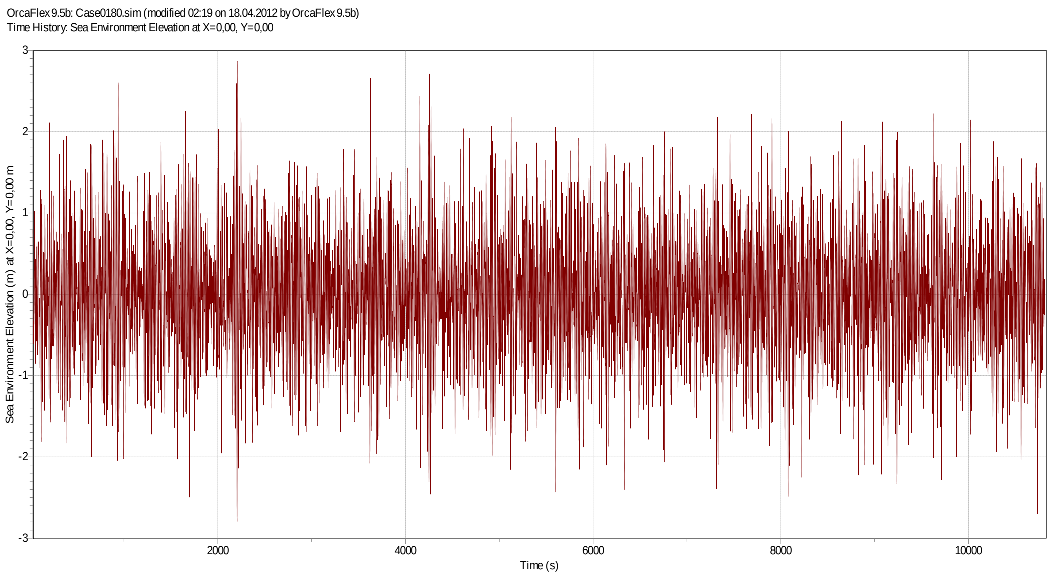


Figure 4.2.2.1: Wave profile of 3 hour duration JONSWAP conditions, parameters: Table 4.2.2.2

Table 4.2.2.2: JONSWAP parameters in Orcaflex 3 hour conditions

Direction	180	Deg
$\boldsymbol{H_S}$	3,0	Meter
$\boldsymbol{T_Z}$	12,0	Sec
$\boldsymbol{\gamma}$	2,000	
$\boldsymbol{f_m}$	0,0622	Hz
$\boldsymbol{T_P}$	16,0764	Sec

Figure 4.2.2.3 represents a zoom of the wave profile of Figure 4.2.2.1. It shows the largest wave in that 3 hour simulation. The actual trough and crest points of that wave are found in Table 4.2.2.4 below.

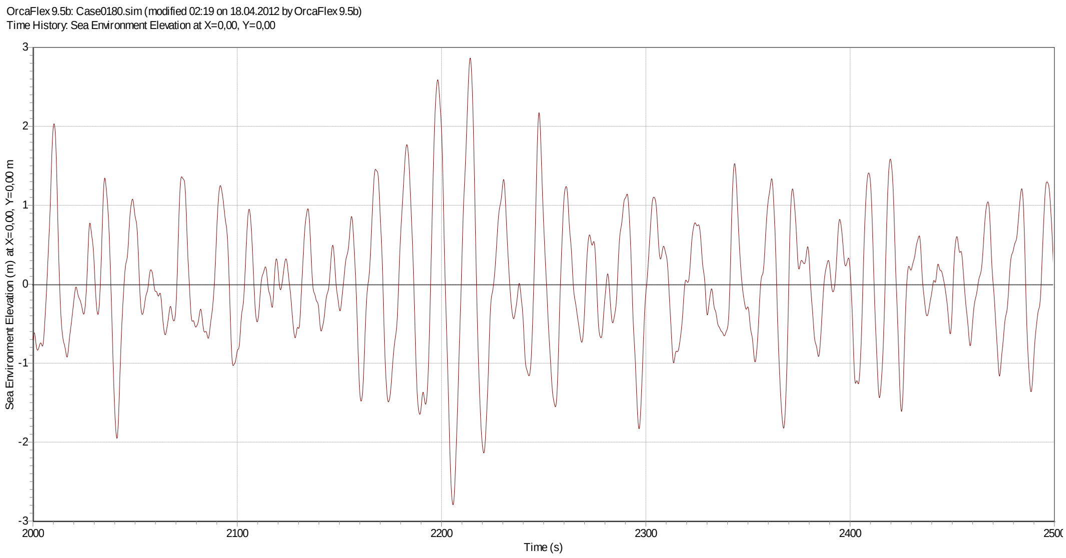


Figure 4.2.2.3: Wave profile zoom from Figure 4.2.2.1 (2000 sec – 2500 sec)

Table 4.2.2.4: Largest wave in 3 hour simulation

Location	Simulation time	Amplitude
Trough	2205,6 s	-2,79m
Crest	2214 s	2,86m

Calculating for $H_{MAX} = 1{,}86 * 3 \text{ m} = 5{,}58 \text{ m}$ and comparing with modeled wave $H_{MAX} = 2{,}86 + 2{,}79 = 5{,}65 \text{ m}$ shows that the results are very close. This simplification is further used in Chapter 4.4 to reduce computing time.

A current was also introduced in the analysis. The current was set to linearly decrease from 1m/s at the surface to 0,2 m/s at the sea floor. This data is not field specific and is set in the same direction as the wave train, directly towards the bow of the vessel.

Wind was not introduced into the analysis due to the large variable of vessel superstructures used for such projects. It was assumed that any wind would be in the wave direction and should thus not cause any large roll motion.

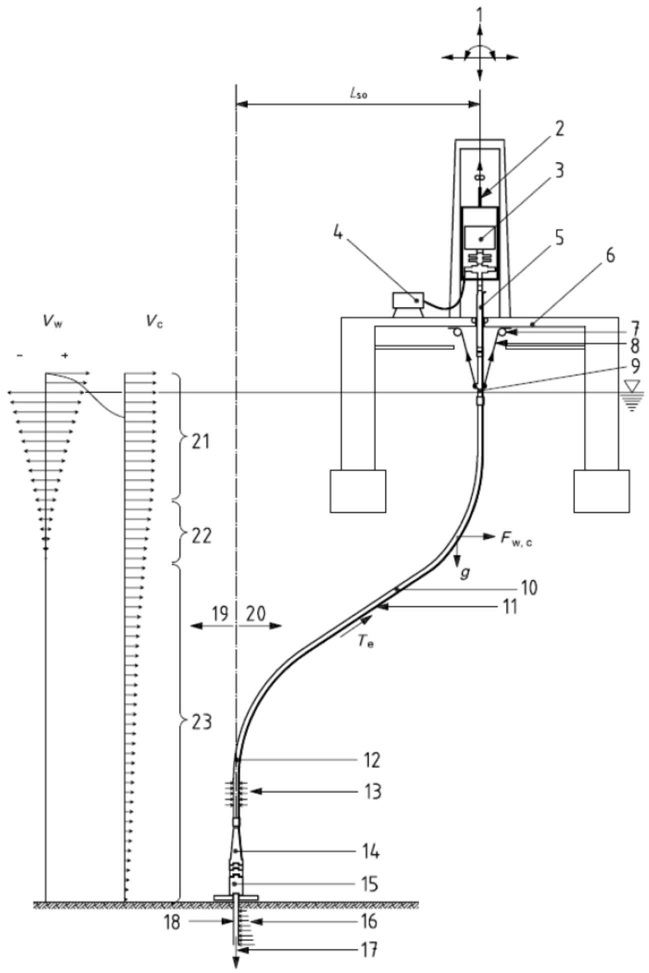
4.2.3 The Subsea Equipment

The model includes a set of subsea equipment that would be in place for intervention operations on subsea wells. The equipment is modeled as 6-D objects in Orcaflex. Starting at the seabed is the Wellhead and the total height of this stack is 10,2 meter above the seabed.

- The wellhead is connected to the seabed and the X-MAS tree
- The X-MAS tree is connected to the LMRP (Lower Marine Riser Package)
- The LMRP is connected to the EDP (Emergency Disconnect Package)
- The EDP is connected to the bottom flange of the stress joint
- The top of the stress joint is connected through an adapter joint to the riser

All equipment except for the EDP were given generic values for size, weight, drag etc. The EDP chosen for this analysis is developed by Subsea Technologies Ltd. A new hydraulic operated connector, the XR Connector™, has been developed as an EDP connector that will allow disengagement at very high angles and bending moments. As this ability is limited with other EDP designs, the combination of the MSJ and XR Connector™ appears like a good fit.

An illustration of the entire global model is presented in Figure 4.2.3.1 on the following page. The illustration is found in (ISO 13628-7 2006).



Key

1 rig motions due to first-order wave motions
2 draw works tension and stroke
3 surface equipment
4 surface pressure (choke or mud-pump)
5 slick joint
6 drill floor
7 tensioner sheaves
8 tensioner tension and stroke
9 tensioner joint
10 outside diameter
11 riser joints
12 bending stiffness
13 external pressure
14 stress joint
15 subsea equipment
16 soil restraint
17 tool
18 conductor bending stiffness
19 upstream
20 downstream
21 excitation zone
22 shear zone
23 damping zone

$F_{w,c}$ wave and current forces
g gravitational force
T_e effective tension
V_w wave velocity
V_c current velocity
L_{so} vessel offset (+)

Figure 4.2.3.1: Principal parameters in the design of completion and workover risers. Found in (ISO 13628-7 2006)

4.2.4 The Vessel

The Vessel used in the model is nameless due to the confidential nature of the information that will be listed in this master thesis. The vessel is however a good representation of an intervention style vessel used in the North Sea today. The vessel has a length of 143m, width of 29,5m and draft of 5,5m. The vessel weight is approx 23400tons. The Response Amplitude Operator (RAO) specifics for this vessel are listed below in Figure 4.2.4.1.

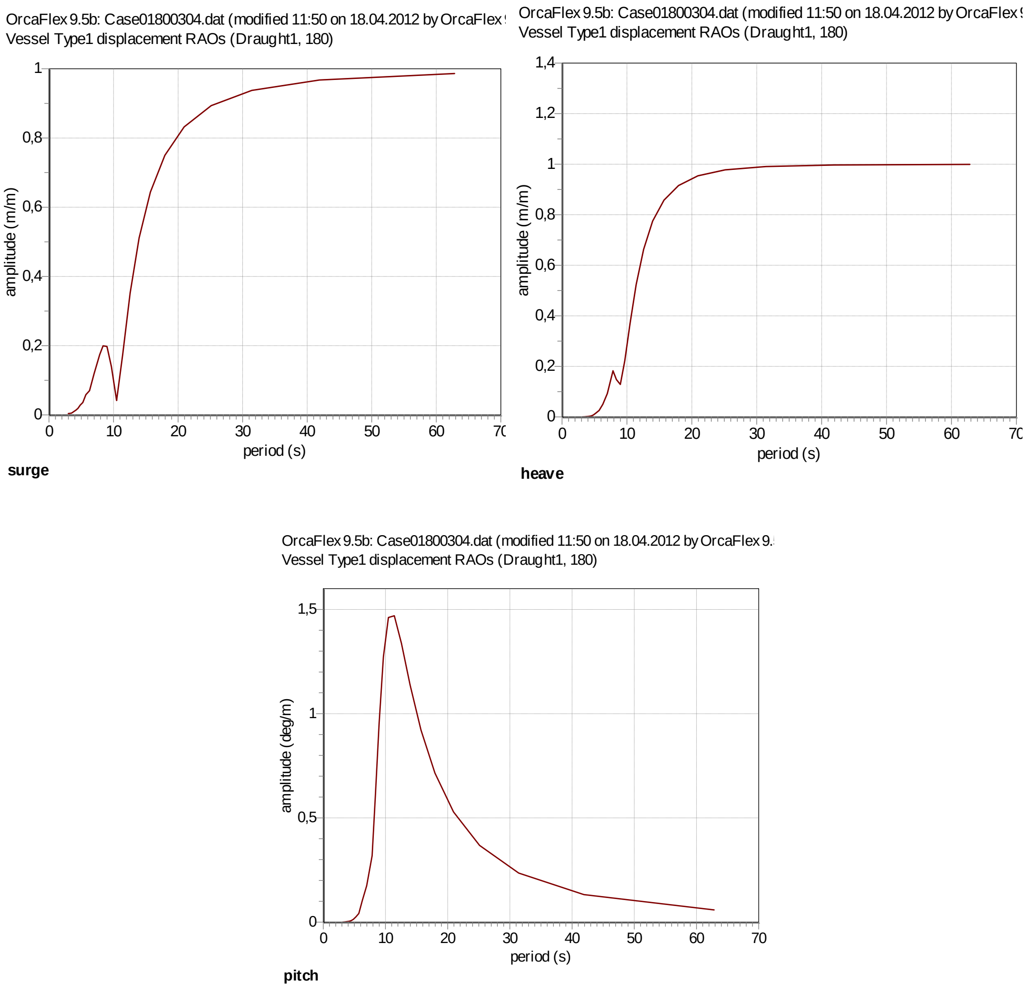


Figure 4.2.4.1: RAO in Surge, Heave and Pitch for intervention vessel

The Figures for roll, sway and yaw are not included as they are set equal to zero for waves moving directly towards the bow of the vessel.

4.2.5 The Riser

The riser used in the analysis consists of several items that are listed below in Table 4.2.5.1. Starting on the top we can see several surface joints at various lengths. These are used to get the length of the riser just right to match it with the heave compensation system installed on the vessel. The surface joints are connected through an adapter joint to the actual steel drilling riser. This is again connected through an adapter joint to the stress joint. The use of these adapter joints allow for a large variety of steel risers to be used with the same surface joints and stress joint. Further info and location of the different joints may be found in the illustration Figure 1.1.1 in Chapter 1.1.

Table 4.2.5.1: Dimension of riser sections used in Orcaflex model

Part	Length	OD	ID
Surface joint 1	2,5 m	193 mm	140 mm
Surface joint 2	2,5 m	193 mm	140 mm
Surface joint 3	8 m	193 mm	140 mm
Surface joint Adapter	3 m	193 mm to 168 mm	139 mm
Riser	56 m	168 mm	139 mm
Adapter joint	2,5 m	168 mm to Profile 1	139 mm
Stress joint	8 m	"Profile 1"	139 mm

"Profile 1" is a variable of the outside dimensions for the stress joint. It is generated with straight lines along the length of the stress joint. A model of both the steel TSJ and the Modular Stress Joint has been generated and will be compared later in the analysis chapter. Further information about items like torsional stiffness, Poisson ratio, Young's Modulus, bending and axial stiffness for parts in the riser stack can be found in the simulation files in the attached CD.

4.2.6 The Model

The Orcaflex models are included on the attached CD. The two models are similar in all respects except for stress joint geometry and materials. The illustration found below in Figure 4.2.6.1 represents both models and clearly depicts the vessel, heave compensation system, riser and subsea equipment. The heave compensation system has been exaggerated in physical size to allow easier visualization of its use.

The generic 3-hour simulation Orcaflex model files are included as ".dat" files on the attached CD due to file size limits. ".dat" files require a valid Orcaflex key to simulate results. There are however included ".sim" files of shorter duration simulations that can be run without a license with demo software from the Orcaflex website. A video of the simulations is also included to allow visualization of the simulated results.

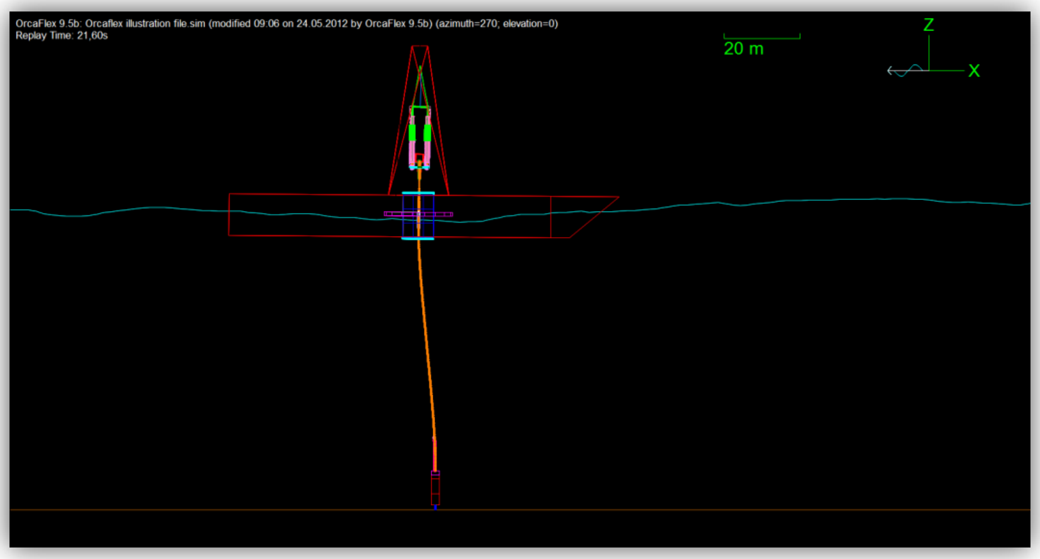


Figure 4.2.6.1: Orcaflex model with JONSWAP waves (H_S = 3m)

The following figures show the modeled equipment on the vessel and subsea in the Orcaflex model. Figure 4.2.6.2 on the following page show the topside of the vessel with the heave compensation system and top of the riser. Figure 4.2.6.3 show the subsea equipment included in the Orcaflex model.

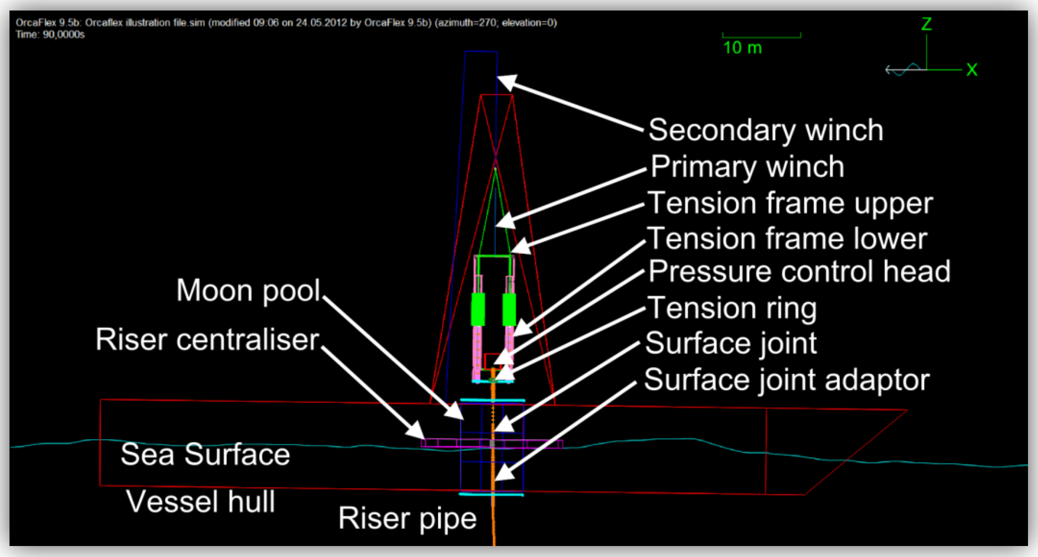


Figure 4.2.6.2.: Topside equipment in Orcaflex model

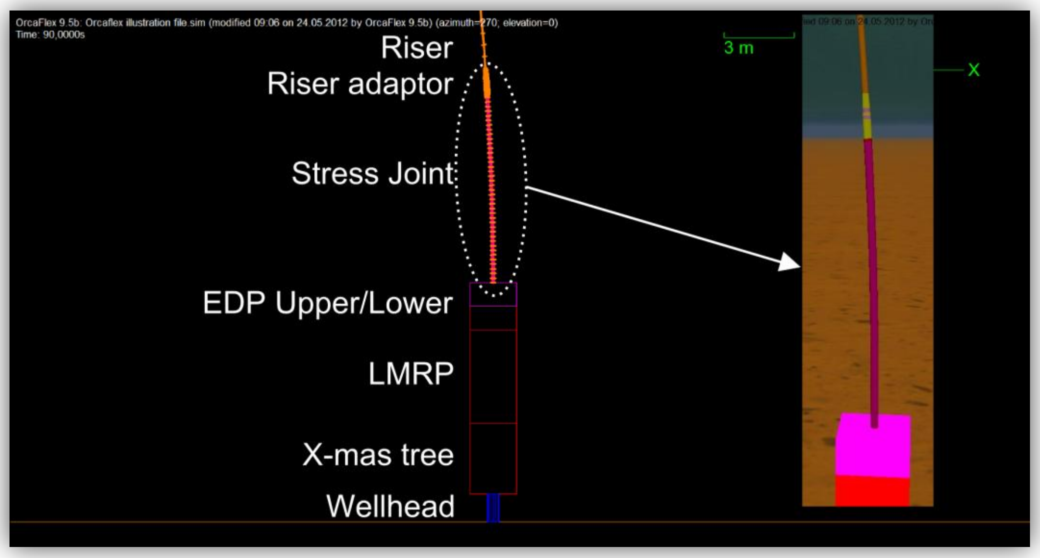


Figure 4.2.6.3: Subsea equipment in Orcaflex model

4.3 Stress Joint FEA

Presenting data on the performance of the MSJ becomes difficult without a way of comparing it with other products or designs. The company Subsea Technologies Ltd, developed a steel stress joint a few years ago. This steel stress joint was used in the design of the full size MSJ to match length, bore and end flanges. This was done so that data between the two designs could easily be compared.

4.3.1 Steel Tapered Stress Joint

The steel Tapered Stress Joint (TSJ) was modeled in Abaqus by a former colleague Matt Petty at AlTiSS Technologies in Houston during the summer of 2011. The Abaqus analysis used the same boundary conditions as with the modeling of the MSJ. That is a fixed bottom flange in all directions. The loads applied were internal pressure, effective tension and a side load to create bending. The effective tension and the side load were concentrated on a point at the top face of the top flange. An illustration of the steel TSJ is included below in Figure 4.3.1.1. Note that non-essential features such as the bolt holes in flanges have been removed to simplify analysis.

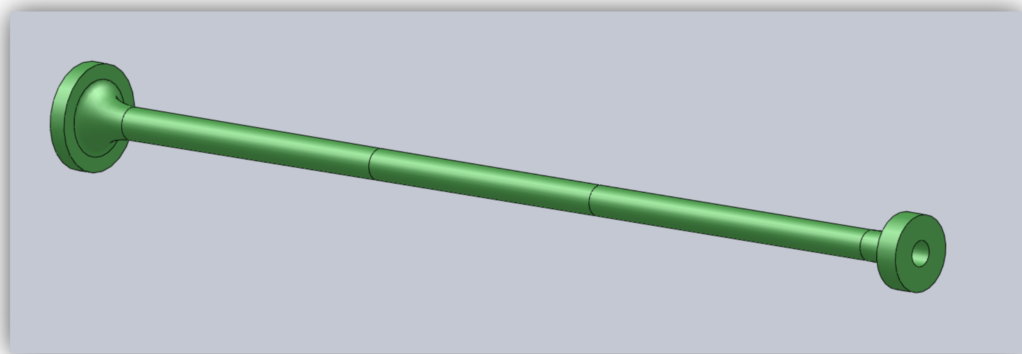


Figure 4.3.1.1: FEA model of steel TSJ

The material properties used in the analysis of the steel TSJ and the FEA mesh data are listed below in Tables 4.3.1.2 and 4.3.1.3.

Table 4.3.1.2: Material properties used in Abaqus FEA for steel TSJ

	Imperial	SI
Young's Modulus	30 000 000 psi	206,8 GPa
Poisson Ratio	0,3	0,3
Yield Strength	80 ksi	551,58 MPa

Table 4.3.1.3: Mesh and elements used in FEA of steel TSJ

Number of elements	20941
Number of nodes	36841
Number of DOF	111786

Testing parameters for the Finite Element Analysis is found in Table 4.1.2 in Chapter 4.1. The analysis was performed with increasing bending load and internal pressure until the max von Mises stress reached the materials yield strength of 80ksi[8]. The results for the maximum permissible deflection based on varying the bending force and internal pressure is showed below in Table 4.3.1.4.

Table 4.3.1.4: FEA results from steel TSJ

Internal pressure	$F_{bending}$	Max von Mises stress	Deflection [inch]
0 psi	53,8 kN	552,09 MPa	562,51 mm [22,15]
5000 psi (34,47 MPa)	75,6 kN	546,08 MPa	464,6 mm [18,29]
10 000 psi (68,95 MPa)	44,5 kN	555,32 MPa	163,6 mm [6,44]

The information in Table 4.3.1.4 shows that the possible deflection before reaching yield stress in the material is greatly affected by the internal pressure. The stress contour over the FEA mesh is illustrated in Figure 4.3.1.5 on the following page.

[8] 551,58 MPa

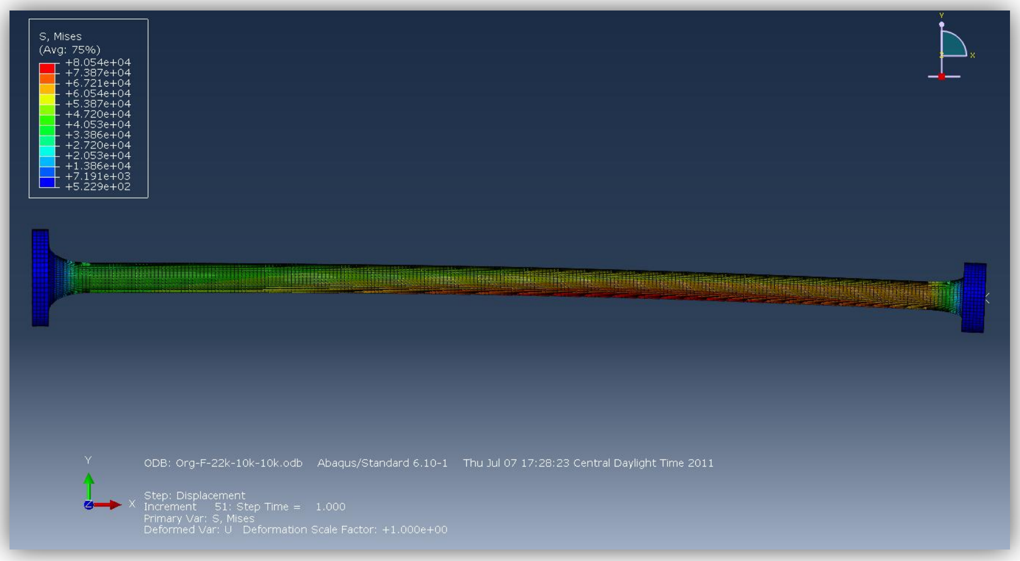


Figure 4.3.1.5: Von Mises Stresses - 10 000 psi (68,95 MPa) internal pressure

4.3.2 Modular Stress Joint

The titanium MSJ was analyzed in Abaqus by Dr. Young-Hoon Han at AlTiSS technologies. The boundary conditions and testing conditions were kept the same as for the modeling of the steel TSJ. The materials used have also been kept the same as for the prototype MSJ analyzed in Chapter 3. The yield strength of the titanium used in the analysis has been lowered compared to the prototype. This has been done due to the difficulty in achieving uniform material properties in material of larger dimensions. The material properties for the titanium used in the MSJ are listed below in Table 4.3.2.1

Table 4.3.2.1: Material properties used in Abaqus FEA for titanium MSJ

	Imperial	SI
Young's Modulus	16 500 000 psi	113,4 GPa
Poisson Ratio	0,33	0,33
Yield Strength	120 ksi	827,4 MPa

The following table, 4.3.2.2, was created with the data from the FEA analysis. It shows the max von Mises stress in the material at increasing deflections

Table 4.3.2.2: Max von Mises stress versus deflection in titanium MSJ

Deflection [inch]	Max von Mises stress
65,22 mm [2,568]	441,73 MPa
130,4 mm [5,134]	441,72 MPa
228 mm [8,977]	475,2 MPa
<u>373,7 mm [14,713]</u>	<u>552,8 MPa</u>
518 mm [20,395]	660,73 MPa
644,7 mm [25,382]	760,76 MPa

A decision was made to use a 2/3 yield criteria to maintain high fatigue life in the MSJ. This results in a max stress allowed in the material of: $2/3 * 827{,}4\ MPa = 551{,}6\ MPa$. The resulting deflection is seen to be 373,7 mm.

The stress contour is shown below in figure 4.3.2.3. The largest stress is found in the middle connector. This FEA analysis is showing a worst case condition. Based on experience from the MSJ prototype, the stresses are assumed to be lower because of the bonded conditions used in modeling of the threads.

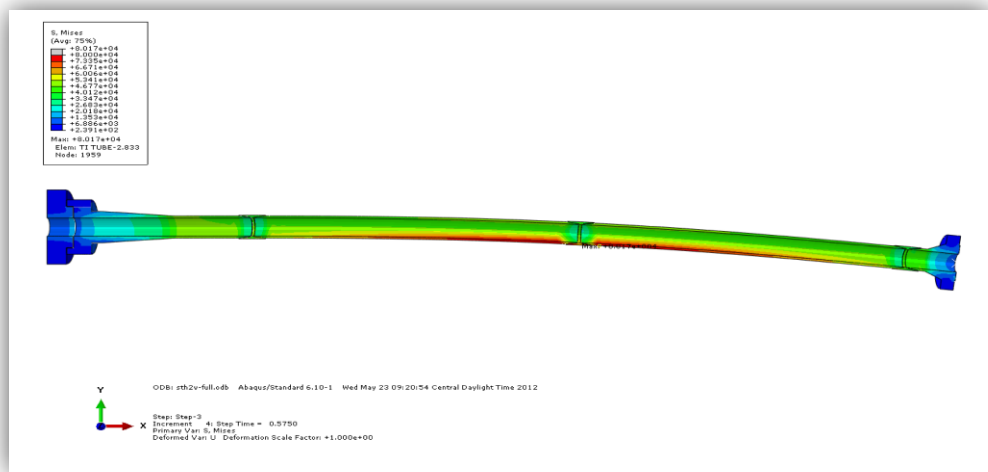


Figure 4.3.2.3: Von Mises stresses - 10 000 psi (68,95 MPa) internal pressure

4.3.3 Comparison of data

The use of the 2/3 yield criteria in the analyzing of the titanium MSJ results in both stress joints being analyzed with equal max stress of 551,6 MPa. The numbers are summarized in Table 4.3.3.1 below. It must be noted that the boundary conditions and loading conditions were kept equal in both cases. The improvement in deflection may be attributed to the design and lower Young's modulus of titanium versus steel.

Table 4.3.3.1: Deflection of steel TSJ vs titanium MSJ

steel TSJ	titanium MSJ	% Increase
163,6 mm	373,7 mm	259 %

4.4 Analysis

The analysis of the modular stress joint has been performed through two different types of software. The local Finite Element Analysis of the stress joint was performed with Abaqus. This data is then used as the limits to how the stress joints may operate in the global Orcaflex model. The reason for using two different types of software is the limited FEA capabilities in Orcaflex. Orcaflex will calculate stresses in components such as the stress joint but the limited ability to draw complex shapes makes the data less accurate than tailor made FEA software. There are a few assumptions in the Orcaflex simulations and analysis. They are listed below.

Assumption 1: All equipment except for the stress joint is given assumed information and properties. This includes properties like weight, material and operational parameters for equipment such as the heave compensation system, riser and subsea equipment.

Assumption 2: The stress joint is assumed to be the weakest link in the riser and the limiting factor in the marine operation. It also assumes than any decision to disconnect from the well will be based on the forces in the stress joint.

To save time in the rough analysis, the 3-hour storm conditions were replaced with shorter single Airy waves. As the largest wave has been identified in a 3-hour conditions to be $H_{MAX} = 1{,}86 * H_S$, this was used to create a shorter simulation with only deterministic H_{MAX} waves. The simulation time was chosen to be 240 sec (4 min) to allow any vibrations in the simulation to damp out. The wave profile is seen on the following page in Figure 4.4.1 for a wave with $H_{MAX} = 9{,}3m$ to represent the worst wave in a 3 hour storm with $H_S = 5m$.

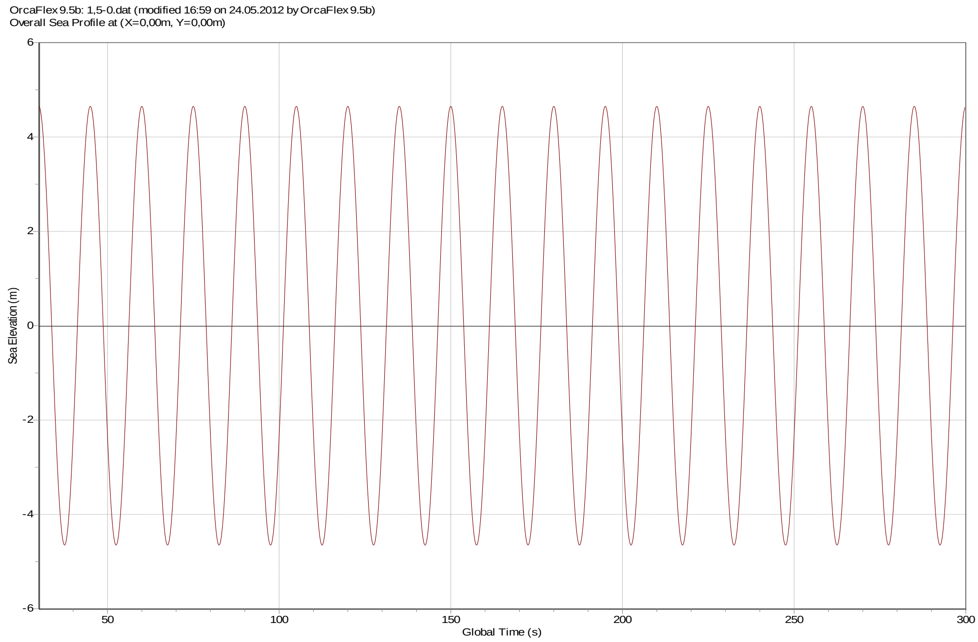


Figure 4.4.1: Wave profile for single Airy wave with $H_{MAX} = 9,3m$

Several load cases were run with varying the wave height and the vessel offset to the wellhead. The wave height and vessel offset were increased until the top of the stress joint passed through its maximum deflection as noted in Table 4.3.3.1 in Chapter 4.3.3. A total of 237 iterations were run to identify the conditions that would lead to maximum deflection. To confirm the data, full 3 hour JONSWAP storm conditions were performed and analyzed to make sure that the maximum deflection did not pass through the set maximum values. The values may be seen for the steel TSJ in Table 4.4.1 and titanium MSJ in Table 4.4.2 on the following page. Numbers are listed for both the 3-hour storm and the 4 minutes with continuous H_{MAX} waves.

The deviation between the maximum deflection in the two cases are small. This shows that the analysis of the 4 min window with H_{MAX} waves served as a good approximation and a good tool to use in the iterations process.

Table 4.4.1: Deflection of steel TSJ in Orcaflex model, all values in meter

H_S	Vessel offset	Deflection 3-hr storm	H_{MAX}	Deflection 4 min with H_{MAX}
5,4	0	0,1587	10	0,1636
4,6	1	0,1631	8,5	0,163
3,8	2	0,1613	7	0,1629
3	3	0,1701	5,6	0,1682
2,2	4	0,1692	4,1	0,1731
1,15	5	0,1699	2,2	0,1743
0,27	6	0,1793	0,5	0,1787

Table 4.4.2: Deflection of titanium MSJ in Orcaflex model, all values in meter

H_S	Vessel offset	Deflection 3-hr storm	H_{MAX}	Deflection 4 min with H_{MAX}
8,3	0	0,3703	15,4	0,3703
7,6	1	0,3651	14,1	0,368
7	2	0,3706	13	0,376
6,2	3	0,3691	11,5	0,3727
5,3	4	0,3679	9,8	0,3748
4,3	5	0,3648	8	0,3691
3,2	6	0,3614	6	0,3659
2,2	7	0,3627	4	0,3621
1,3	8	0,368	2,4	0,3735
0,4	9	0,3771	0,7	0,3651

The 3-hour models (.dat-files) have been included on the attached CD with the following file name system:

CASE_STEEL_TSJ_Hs("value")_X("Offset value")

CASE_TITANIUM_MSJ_Hs("value")_X("Offset value")

The numbers for significant wave height and vessel offset was used to generate Figure 4.4.3. It must be noted that the values in this graph are greatly affected by the assumptions stated in the beginning of this chapter. The performance characteristics of the stress joints may thus not be used as real figures due to the uncertainties in other equipment in the riser string. What the graph does show is the general improvement in operational window between the steel TSJ and the titanium MSJ.

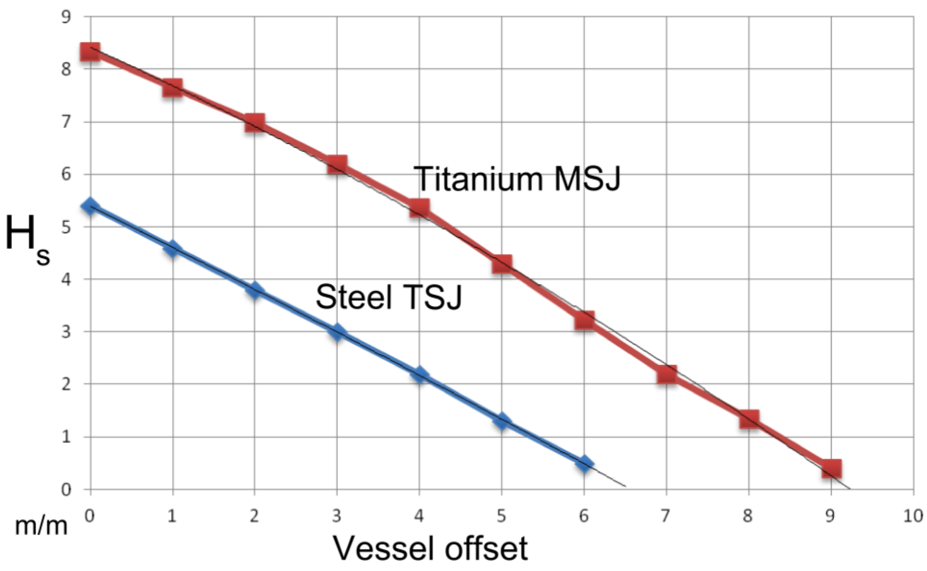


Figure 4.4.3: Vessel offset versus Significant wave height from Orcaflex model

The data generated in this analysis is further commented in Chapter 5, Conclusions.

5. Conclusions

The development of the Modular Stress Joint (MSJ) started with a set of design objectives. The set of design objectives defined in the design basis were, manufacturing cost, production lead time, transportability and versatility. The four objectives were incorporated in the design at an early stage with the result being an unproven design of the MSJ. The design was tested and analyzed to the point where the next step was building a prototype. The prototype was manufactured, assembled and tested as a part of the thesis.

The MSJ prototype was fitted with attached strain gauges during the pressure end bending tests performed and reported in Chapter 3. The prototype was subjected to a rigorous testing procedure with the exception of Stages 2 and 4. Those two Stages of testing will be performed in combination with a long term fatigue testing during the summer of 2012 to complete the testing procedure.

The results provided confidence in the design of the MSJ. Some issues were discovered in this process but they were related to the areas of the connector and this is still an area of development. New connectors will be developed to control the problems discovered during the prototype testing assembly.

The strain gauge data from the prototype MSJ as reported in Chapter 3 provided confidence in the results from the FEA analysis on the full size MSJ reported in Chapter 4. In Chapter 4, a full size Orcaflex model of a riser system with a vessel was used to test the design in simulated real world conditions. The Orcaflex model was used to compare the design of a steel Tapered Stress Joint (TSJ) and the titanium MSJ. The results found in Chapter 4.4 showed an improvement in the operability of the titanium MSJ versus the steel TSJ. This improvement is largely caused by the lower Young's modulus of titanium versus steel.

The use of titanium in offshore applications is nothing new. The use of titanium in contact with sea water and sour conditions are proven solutions. The use of titanium in risers and stress joints are also proven solutions. The unproven solution is the connections and seal surfaces in this area of high bending and stress. The connectors provided leak tightness during the testing of the prototype. No problems were anticipated in this short test procedure. Further fatigue testing is required to identify any problems over time with cyclic loading of the MSJ.

Titanium is a more expensive material than steel and thus used less in the industry. The superior corrosion capabilities and strength/weight relation is not enough to simply replace steel. One of the design ideas of the MSJ is to increase the use of titanium by increasing the versatility. The characteristics of the MSJ may be altered by adding or removing titanium tubes to get the required bending for each field. As a result of the increased versatility, the impact of the manufacturing cost is lowered.

Part of the conclusion is also related to potential improvements in the design of the MSJ. One item that should be addressed in a full size model of the MSJ is the stress distribution. The current design shows high stresses moving into the base piece just below the first connector. The assumed outcome of the fatigue testing scheduled for the summer of 2012, is that the MSJ will fail after a high number of cycles. The failure point will usually be in an area of high stress where the materials "fatigue life" is used up. This area should be located in a titanium tube that may be easily replaced and not in the more expensive base piece.

Based on the assumptions listed in Chapter 4, the analysis provided a comparison of capabilities between the steel TSJ and the titanium MSJ. The capabilities of the stress joints were visualized in Figure 4.4.3 in Chapter 4.4. This figure is included again below as Figure 5.1.

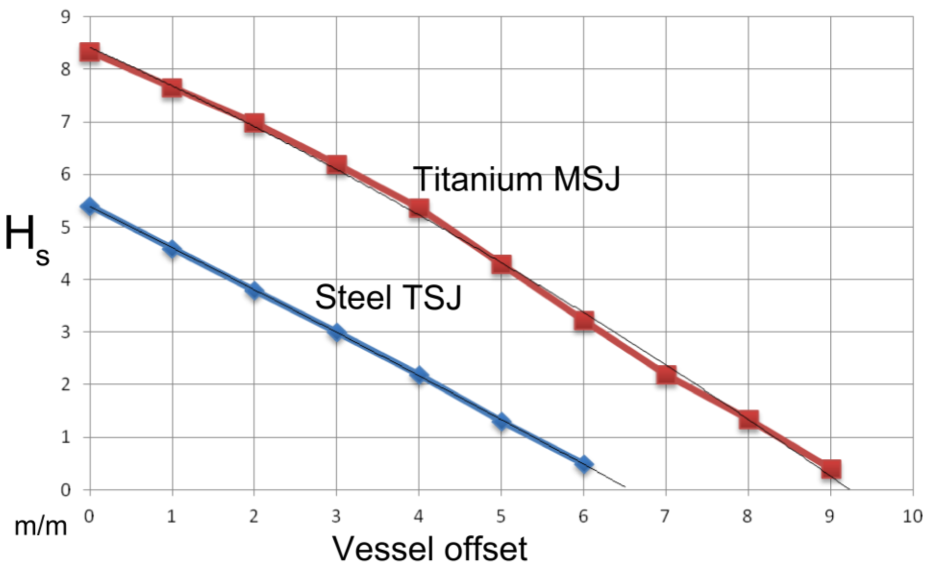
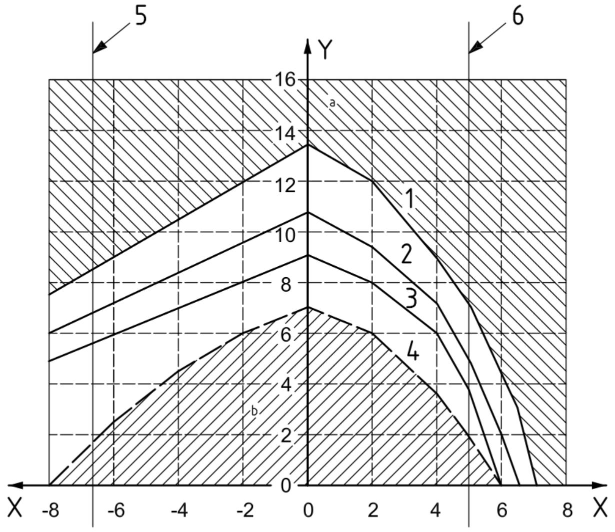


Figure 5.1: Vessel offset versus Significant wave height from Orcaflex model

Figure 5.1 shows how the significant wave height may increase by an average of 2,5-3m from the steel TSJ to the titanium MSJ. This increase in the significant wave height and vessel offset increases the operational envelope for the marine operation. The information provided in Figure 5.1 would be correlated with operational parameters from other equipment into an operating envelope as illustrated in Figure 5.2 on the following page. This illustration is found in (ISO 13628-7 2006).



Location: Field ....
Vessel: Vessel name ....
Riser: Designation ....
Mode: Tree mode ....
Top configuration: Tension frame.
Top tension:
Motion compensator: ....kN (lbf)
Tension ring: ....kN (lbf)
Bottom tension:
Bottom of EDP: ....kN (lbf)
Pressure: ....MPa (psi)
Fluid density: .... s.g.
Wave direction: Head ....
T_p range: 90 % confidence interval
Current: .... year return period
Current direction: Head ....
Wellhead inclination: ....°

Key

X vessel static offset, L_{so}, from wellhead extension, measured as percentage of water depth, positive in direction of current

Y significant wave height, H_s

1 strength limit: accidental

2 strength limit: extreme

3 strength limit: normal

4 stroke limit: motion compensator

5 EDP angle limit: upstream

6 EDP angle limit: downstream

[a] Unsafe operating area.

[b] Safe operating area.

Figure 5.2: Typical operating envelope – Tree mode (ISO 13628-7 2006)

Pending development of a new connector and successful fatigue testing of the prototype and a full size MSJ, the next step is to generate such an operating envelope as seen in figure 5.2 with all relevant information and only limited assumptions. It is not expected that the titanium MSJ will outperform a custom designed titanium stress joint. It is expected that the titanium MSJ will be an almost as good solution with lower cost, lead time and higher versatility. This would make the titanium MSJ a competitive solution to the custom made titanium and steel TSJ.

6. References

Abadi, Parviz, and Inc ABB Vetco Gray. USA Patent US6659690. 2003.

Avery, D, D Byrd, T Whitlock, D Cole, and N Bhathena. "Wyman-Gordon Forgings." *Integrally extruded flaged titanium stress joint for a workover riser, OTC7922.* May 1-4, 1995. http://www.onepetro.org/mslib/app/Preview.do?paperNumber=OTC-7922-MS&societyCode=OTC (accessed March 29, 2012).

Brett, Paul, Karim Jan, and Simon Luffrum. "Subsea Riser Products Ltd." *Why Shrink-Fit Steel Flanges to Titanium Pipe?* February 2010. http://www.subseariserproducts.com/documents/papers/9602-1003-02-Shrink-fitting-of-Steel-Flanges-to-Titanium-Pipe-DOT-Houston-Feb-2010-SUBMITTED.pdf (accessed March 30, 2012).

Conoco, Inc. USA Patent US4256417. 1978.

Det Norske Veritas. *Design of Titanium Risers.* Høvik: DNV, October 2002.

Dziekonski, Mitch. USA, Texas, Houston Patent Patent Pending #13/506352. 2012.

Groves, Steve, Mike Hogan, Scott Moses, Ralph Dean, and Frank Hartley. 2010. http://www.pennenergy.com/index/petroleum/display/272386/articles/offshore/volume-66/issue-9/production/flexible-joint-improvements-may-aid-reliability.html (accessed February 21, 2012).

Hottinger Baldwin Messtechnik. "Operating Manual Spider-8."

ISO 13628-7. *BS EN ISO 13628-7:2006; Petrolem and natural gas industries - Design and operation of subsea priduction systems - Part 7: Completion/workover riser systems.* British Standards, 2006.

K.Chakrabarti, Subrata. "Handbook of Offshore Engineering." In *Volume I*, 106-117. Plainfield Illinois, USA: Elsevier, 2005.

Oil States Industries. "FlexJoint." *For Drilling and Production Risers.* 2004. http://www.oilstates.com/_filelib/FileCabinet/_R/G/_27G/Brochures/Offshore/Floating_Production_Systems/Flexjoint.pdf?FileName=Flexjoint.pdf (accessed February 21, 2012).

Peacock, David. *Why is Titanium used in Offshore Applications.* December 1996. www.azom.com/article.aspx?articleID=638 (accessed March 29, 2012).

Stainless Steel World. *Fact File Titanium > Offshore.* 2010. http://www.stainless-steel-world.net/titanium/ShowPage.aspx?pageID=170 (accessed March 29, 2012).

Appendix

List of Appendixes

Appendix A: Test Procedure for prototype MSJ

This test procedure was used to setup the testing and to prepare the test personnel for any dangerous activities. …

Stage 1

The Pressure test shall be performed with the following procedure and any deviations shall be noted in the test form for pressure testing with no external loads as attached.

1. The working pressure for the prototype is set at 10 000 psi or 690 BAR.
2. The test pressure for the prototype is set at 15 000 psi or 1035 BAR.

   NOTE:

   - The pressure SHALL NOT go beyond the test pressure set at 15 000 psi or 1035 BAR at any time during testing.
   - The test pressure range between 10 000 and 15 000 psi (690 BAR and 1035 BAR) is only valid for pressure testing with NO external loading.

3. The prototype shall be assembled according to written specifications.
4. A visual inspection of the prototype shall be performed before any testing. Specific attention on any damages on the material and to the connections.
5. Inspect the test port for leak tightness.
6. Verify that all connected pressure equipment is certified to the pressure the test will achieve.
7. Apply a preliminary test pressure of 50 psi/3,5 BAR. Close inlet valve and inspect for any leaks indicated visually on prototype or by pressure drop. Hold for 10 minutes.
8. The test pressure may then be increased in accordance with test form: "Stage 1: Pressure testing of prototype with no external loads".
9. Information regarding actual pressure, pressure drop and minutes held at pressure shall be recorded on the test form.
10. After the test program is finished, the test pressure shall be slowly decreased. No more than 10 000 psi / 700 BAR/per minute.

STAGE 1

Stage 1: Pressure testing with NO external loading

Equipment tested: 1/3 Scale model of Titanium Modular Stress Joint

Test technician:__________________________ Date:______________

Pressure testing medium: ______________________________________

Test Parameter / Load case	Test pressure [psig]	Test pressure [BARg]	Vertical force [N]	Tension Force [N]	Acctual pressure [psi]/[BAR]	Pressure drop [psi]/[BAR]	Vertical Deflection [cm]	Load hold minimum [minutes]	Comments and Data Recording ID:	Signature	Procedure
Load case 1	0	0							Inspection		
Load case 2	200	14						10			
Load case 3	500	34						10			
Load case 4	2500	172									
Load case 5	5000	345									
Load case 6	7500	517									
Load case 7	10000	689									
Load case 8	12500	862									
Load case 9	15000	1034									
Load case 10	10000	689									
Load case 11	7500	517									
Load case 12	5000	345									
Load case 13	2500	172									
Load case 14	500	34									
Load case 15	0	0									
Finish	0	0	0	0					Inspection		

Prepared by Thomas Sola Larsen for Titanium Engineers AS	Form approved by:______________________________
Date: 27.02.2012	Testing approved by: ______________________________

Stage 2

1. The working pressure for the prototype is set at 10000 psi or 690 BAR.
2. The test pressure for the prototype is set at 15000 psi or 1035 BAR.

   NOTE:

   - The pressure SHALL NOT go beyond the test pressure set at 15000 psi or 1035 BAR at any time during testing.
   - The test pressure range between 10000 and 15000 psi (690 BAR and 1035 BAR) is only valid for pressure testing with NO external loading.

3. The testing shall follow the general rules for pressure testing from Stage 1 with the following exemption.

   EXEMPTION:

   - The times set for pressure hold in Stage 1 may be disregarded if Stage 1 testing showed no pressure leak.

4. Tension and pressure shall be increased while following the test form: "Stage2: Pressure testing with variable tension loading applied".

   NOTE:

   - Applied tension SHALL NOT go beyond the test force of 15600 N
5. Information regarding actual pressure, pressure drop and load hold shall be documented in the same test form.

STAGE 2

Stage 2: Pressure testing with variable tension loading applied

Equipment tested: 1/3 Scale model of Titanium Modular Stress Joint

Test technician:__________________________ Date:______________

Pressure testing medium: _____________________________________

Test Parameter / Load case	Test pressure [psig]	Test pressure [BARg]	Vertical force [N]	Tension Force [N]	Acctual pressure [psi]/[BAR]	Pressure drop [psi]/[BAR]	Vertical Deflection [cm]	Load hold minimum [minutes]	Comments and Data Recording ID:	Signature	Procedure
Load case 1	0	0	0	2500					Inspection		
Load case 2	2500	172	0	2500							
Load case 3	5000	345	0	2500							
Load case 4	7500	517	0	2500							
Load case 5	10000	689	0	2500							
Load case 6	0	0	0	5000							
Load case 7	2500	172	0	5000							
Load case 8	5000	345	0	5000							
Load case 9	7500	517	0	5000							
Load case 10	10000	689	0	5000							
Load case 11	0	0	0	10000							
Load case 12	2500	172	0	10000							
Load case 13	5000	345	0	10000							
Load case 14	7500	517	0	10000							
Load case 15	10000	689	0	10000							
Load case 16	0	0	0	15600							
Load case 17	2500	172	0	15600							
Load case 18	5000	345	0	15600							
Load case 19	7500	517	0	15600							
Load case 20	10000	689	0	15600							
Finish	0	0	0	0					Inspection		

Prepared by Thomas Sola Larsen for Titanium Engineers AS	Form approved by:________________________________
Date: 27.02.2012	Testing approved by: _____________________________

Stage 3

1. The working pressure for the prototype is set at 10000 psi or 690 BAR.
2. The test pressure for the prototype is set at 15000 psi or 1035 BAR.

   NOTE:

   - The pressure SHALL NOT go beyond the test pressure set at 15000 psi or 1035 BAR at any time during testing. .
   - The test pressure range between 10000 and 15000 psi (690 BAR and 1035 BAR) is only valid for pressure testing with NO external loading.

3. The testing shall follow the general rules for pressure testing from Stage 1 with the following exemption.

   EXEMPTION:

   - The times set for pressure hold in Stage 1 may be disregarded if Stage 1 testing showed no pressure leak.

4. Vertical forces and pressure shall be increased while following the test form: "Stage3: Pressure testing with variable vertical loading applied".

   NOTE:

   - Applied force for deflection SHALL NOT go beyond 4900 N

5. Information regarding actual pressure, pressure drop, vertical deflection and load hold shall be documented in the same test form.

STAGE 3

Stage 3: Pressure testing with vertical loading applied

Test technician:__________________________ Date:______________

Equipment tested: 1/3 Scale model of Titanium Modular Stress Joint

Pressure testing medium: _____________________________________

Test Parameter / Load case	Test pressure [psig]	Test pressure [BARg]	Vertical force [N]	Tension Force [N]	Acctual pressure [psi]/[BAR]	Pressure drop [psi]/[BAR]	Vertical Deflection [cm]	Load hold minimum [minutes]	Comments and Data Recording ID:	Signature	Procedure
Load case 1	0	0	1000	0					Inspection		
Load case 2	2500	172	1000	0							
Load case 3	5000	345	1000	0							
Load case 4	7500	517	1000	0							
Load case 5	10000	689	1000	0							
Load case 6	0	0	2000	0							
Load case 7	2500	172	2000	0							
Load case 8	5000	345	2000	0							
Load case 9	7500	517	2000	0							
Load case 10	10000	689	2000	0							
Load case 11	0	0	3000	0							
Load case 12	2500	172	3000	0							
Load case 13	5000	345	3000	0							
Load case 14	7500	517	3000	0							
Load case 15	10000	689	3000	0							
Load case 16	0	0	4900	0							
Load case 17	2500	172	4900	0							
Load case 18	5000	345	4900	0							
Load case 19	7500	517	4900	0							
Load case 20	10000	689	4900	0							
Finish	0	0	0	0					Inspection		

Prepared by Thomas Sola Larsen for Titanium Engineers AS	Form approved by:__________________________________
Date: 27.02.2012	Testing approved by: _______________________________

Stage 4

1. The working pressure for the prototype is set at 10000 psi or 690 BAR.
2. The test pressure for the prototype is set at 15000 psi or 1035 BAR.

   NOTE:

   - The pressure SHALL NOT go beyond the test pressure set at 15000 psi or 1035 BAR at any time during testing. .
   - The test pressure range between 10000 and 15000 psi (690 BAR and 1035 BAR) is only valid for pressure testing with NO external loading.

3. The testing shall follow the general rules for pressure testing from Stage 1 with the following exemption.

   EXEMPTION:

   - The times set for pressure hold in Stage 1 may be disregarded if Stage 1 testing showed no pressure leak.

4. Vertical forces, tension and pressure shall be increased while following the test form: "Stage4: Pressure testing with both variable vertical loading and variable tension loading applied".

   NOTE:

   - Applied tension SHALL NOT go beyond the test force of 15600 N
   - Applied force for deflection SHALL NOT go beyond 4900 N
5. Information regarding actual pressure, pressure drop, vertical deflection and load hold shall be documented in the same test form.

STAGE 4

Stage 4: Pressure testing with variable tension and vertical loading

Equipment tested: 1/3 Scale model of Titanium Modular Stress Joint

Test technician:____________________ Date:__________

Pressure testing medium: ______________________________

Test Parameter / Load case	Test pressure [psig]	Test pressure [BARg]	Vertical force [N]	Tension Force [N]	Acctual pressure [psi]/[BAR]	Pressure drop [psi]/[BAR]	Vertical Deflection [cm]	Load hold minimum [minutes]	Comments and Data Recording ID:	Signature	Procedure
Load case 1	0	0	1000	2500					Inspection		
Load case 2	2500	172	1000	2500							
Load case 3	5000	345	1000	2500							
Load case 4	7500	517	1000	2500							
Load case 5	10000	689	1000	2500							
Load case 6	0	0	2000	5000							
Load case 7	2500	172	2000	5000							
Load case 8	5000	345	2000	5000							
Load case 9	7500	517	2000	5000							
Load case 10	10000	689	2000	5000							
Load case 11	0	0	3000	10000							
Load case 12	2500	172	3000	10000							
Load case 13	5000	345	3000	10000							
Load case 14	7500	517	3000	10000							
Load case 15	10000	689	3000	10000							
Load case 16	0	0	4900	15600							
Load case 17	2500	172	4900	15600							
Load case 18	5000	345	4900	15600							
Load case 19	7500	517	4900	15600							
Load case 20	10000	689	4900	15600							
Finish	0	0	0	0					Inspection		

Prepared by Thomas Sola Larsen for Titanium Engineers AS	Form approved by:____________________
Date: 27.02.2012	Testing approved by: ____________________

Appendix B: Test Certificates for pressure testing

STL

Testcertificate

Certificate no. : 430

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

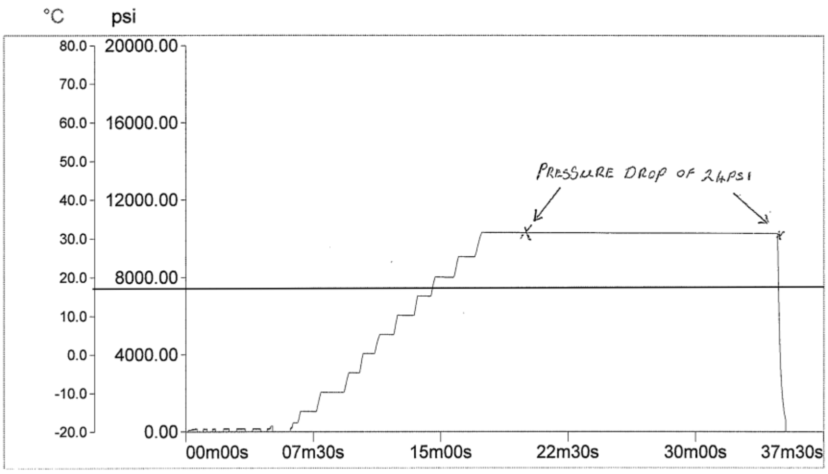
Description	:	stress joint
	:	_
Test Info	:	Job Number
		_
I.D.	:	0.0 mm
Length	:	0 mm
Working pressure	:	10000.0 psi
Test pressure	:	15000.0 psi
Burst pressure	:	0.0 psi

REMARKS

: Reference Info
: 15 min test_
: Test No 1_
: _

TEST DATA

Peak pressure	:	10334.0 psi
Test time	:	00h35m40s
Test medium	:	Water



Tested by	:	E Spence_
Test date and time	:	01.05.2012 16:26
Order no.	:	Customer Name

Approved by :

Testcertificate

Certificate no. : 432

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

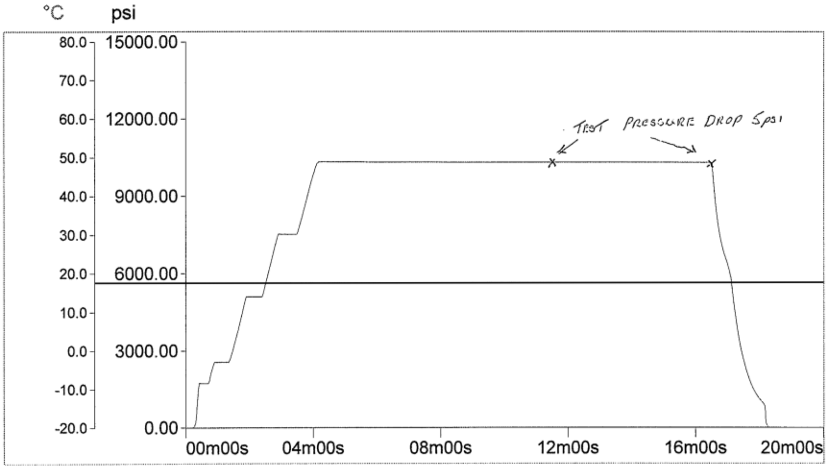
Description	:	Serial Number
	:	Stress joint_
Test Info	:	Job Number
		_
I.D.	:	0.0 mm
Length	:	0 mm
Working pressure	:	0.0 psi
Test pressure	:	10000.0 psi
Burst pressure	:	0.0 psi

REMARKS

: Reference Info
: Test No 2_
: 5 min test_
: No side load_

TEST DATA

Peak pressure	:	10355.7 psi
Test time	:	00h18m33s
Test medium	:	Water



Tested by	:	E Spence_
Test date and time	:	02.05.2012 14:23
Order no.	:	Customer Name

Approved by :

Testcertificate

Certificate no. : 437

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

REMARKS

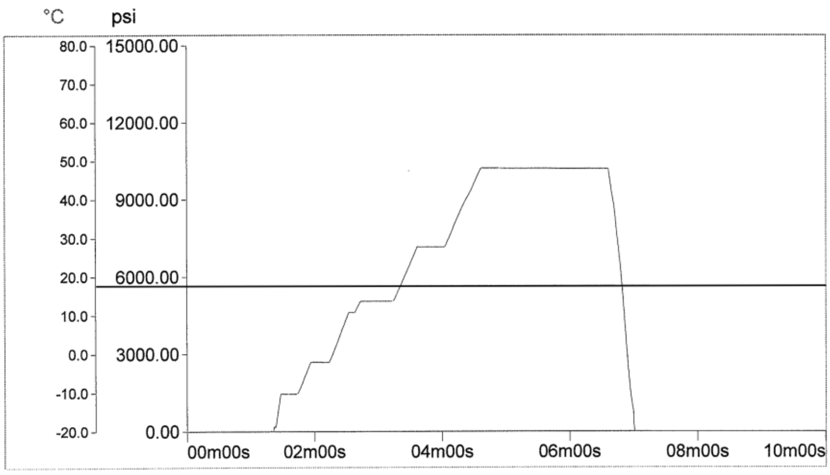
: Reference Info
: Test No a 1_
: 5 min test_
: 1" side load_

EQUIPMENT DATA

Description	: Serial Number	
	: Stress joint_	
Test Info	: Job Number	
	_	
I.D.	: 0.0 mm	
Length	: 0 mm	
Working pressure	: 0.0	psi
Test pressure	: 10000.0	psi
Burst pressure	: 0.0	psi

TEST DATA

Peak pressure	: 10223.4	psi
Test time	: 00h07m11s	
Test medium	: Water	



Tested by : E Spence_
Test date and time : 02.05.2012 17:05
Order no. : Customer Name

Approved by :

Testcertificate

Certificate no. : 438

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

Description : Serial Number
: Stress joint_
Test Info : Job Number
_

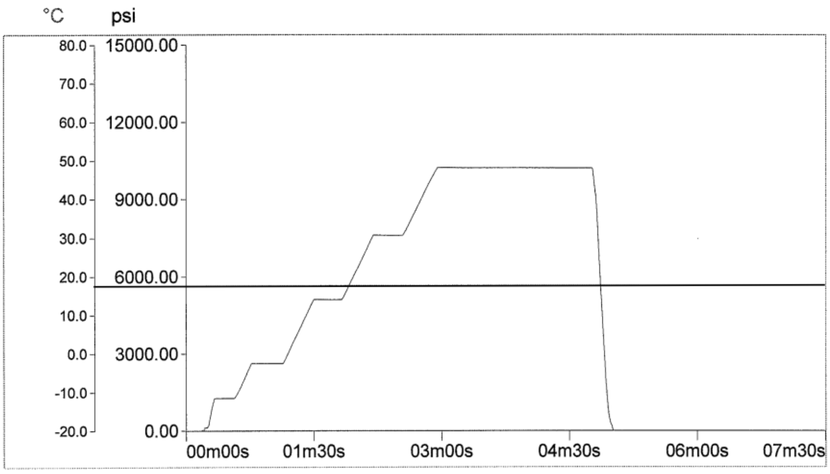
I.D. : 0.0 mm
Length : 0 mm
Working pressure : 0.0 psi
Test pressure : 10000.0 psi
Burst pressure : 0.0 psi

REMARKS

: Reference Info
: Test No a 2_
: 5 min test_
: 2" side load_

TEST DATA

Peak pressure : 10240.2 psi
Test time : 00h05m14s
Test medium : Water



Tested by : E Spence_
Test date and time : 02.05.2012 17:14
Order no. : Customer Name

Approved by :

Testcertificate

Certificate no. : 439

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

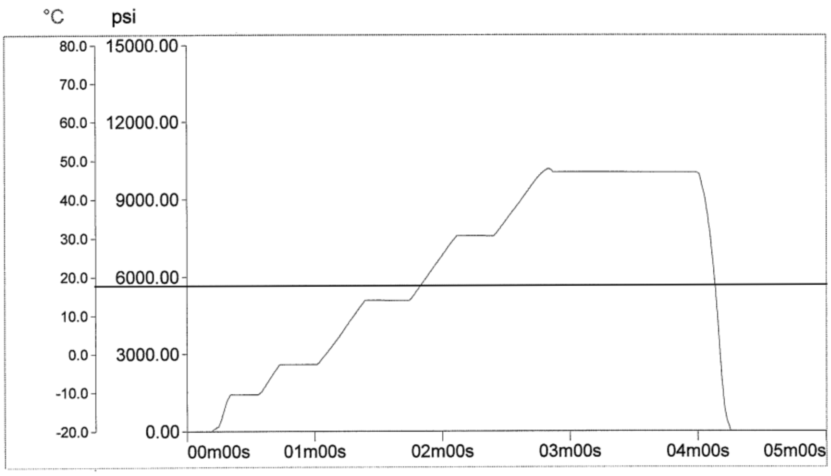
Description	:	Serial Number
	:	Stress joint_
Test Info	:	Job Number
		_
I.D.	:	0.0 mm
Length	:	0 mm
Working pressure	:	0.0 psi
Test pressure	:	10000.0 psi
Burst pressure	:	0.0 psi

REMARKS

: Reference Info
: Test No a 3_
: 5 min test_
: 3" side load_

TEST DATA

Peak pressure	:	10190.9 psi
Test time	:	00h04m27s
Test medium	:	Water



Tested by : E Spence_
Test date and time : 02.05.2012 17:22
Order no. : Customer Name

Approved by :

STL

Testcertificate

Certificate no. : 440

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

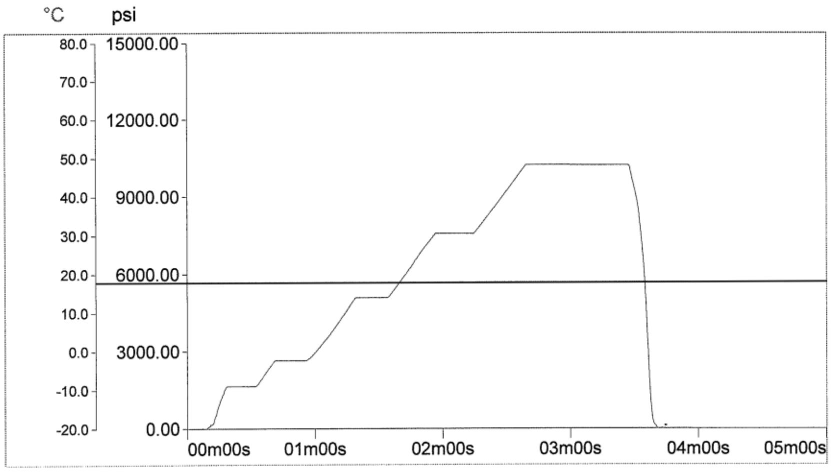
Description	:	Serial Number
	:	Stress joint_
Test Info	:	Job Number
		_
I.D.	:	0.0 mm
Length	:	0 mm
Working pressure	:	0.0 psi
Test pressure	:	10000.0 psi
Burst pressure	:	0.0 psi

REMARKS

: Reference Info
: Test No a 4_
: 5 min test_
: 4" side load_

TEST DATA

Peak pressure	:	10266.7 psi
Test time	:	00h03m51s
Test medium	:	Water



Tested by	:	E Spence_
Test date and time	:	02.05.2012 17:31
Order no.	:	Customer Name

Approved by :

Testcertificate

Certificate no. : 442

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

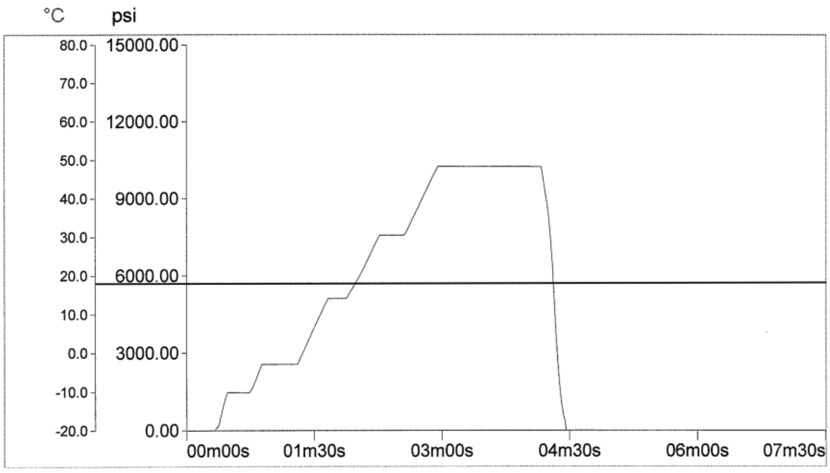
Description : Serial Number
: Stress joint_
Test Info : Job Number
_
I.D. : 0.0 mm
Length : 0 mm
Working pressure : 0.0 psi
Test pressure : 10000.0 psi
Burst pressure : 0.0 psi

REMARKS

: Reference Info
: Test No a 5
: 5 min test_
: 5.50" side load_

TEST DATA

Peak pressure : 10272.7 psi
Test time : 00h04m44s
Test medium : Water



Tested by : E Spence_
Test date and time : 02.05.2012 17:43
Order no. : Customer Name

Approved by :

STL

Testcertificate

Certificate no. : 444

CUSTOMER DATA

Titanium Engineers
MSJ Prototype
Partner

Order no. customer : _

EQUIPMENT DATA

Description	:	Serial Number
	:	Stress joint_
Test Info	:	Job Number
		_
I.D.	:	0.0 mm
Length	:	0 mm
Working pressure	:	0.0 psi
Test pressure	:	15000.0 psi
Burst pressure	:	0.0 psi

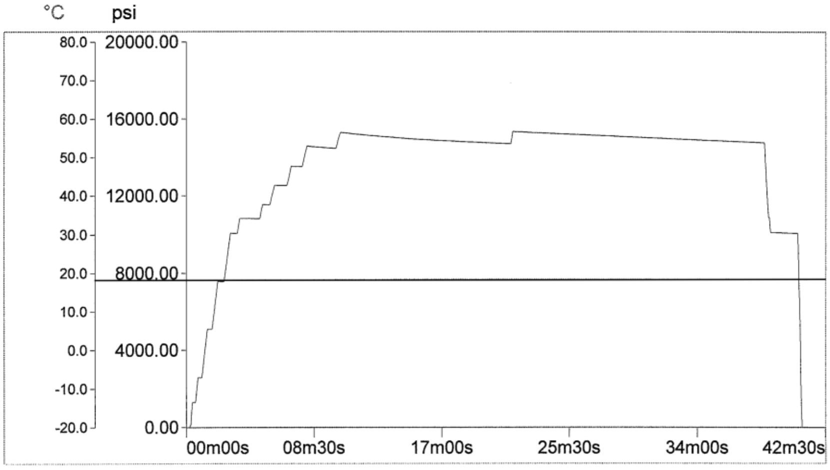
REMARKS

: Reference Info
: Test No a 6
: 15 min test_
: No side load_

LEAK AT HOSE FITTING

TEST DATA

Peak pressure	:	15368.2 psi
Test time	:	00h41m14s
Test medium	:	Water



Tested by	:	E Spence_
Test date and time	:	02.05.2012 17:57
Order no.	:	Customer Name

Approved by :

Appendix C: Material certificates for prototype MSJ

Special Testing Ltd
Bacon Lane, Sheffield S9 3NH
Tel: 0114 244 1061
Fax: 0114 244 0444 Works
Fax: 0114 244 5566 Acc/Admin
www.specialtesting.co.uk

UKAS TESTING 0046

Test Certificate 188400

AUTHENTICATION CODE 677E432E21E1
ROLLS ROYCE APPROVED LABORATORY
Approval No. 11191

CUSTOMER
SPECIAL STEELS LIMITED
BACON LANE
ATTERCLIFFE
SHEFFIELD S9 3NH
DATE RECEIVED 27 September 2011
TEST DATE 28 September 2011

CUSTOMER ORDER NO	260195A BR	TEST NUMBER	188400	IDENTITY	260195A BR	CAST	T29183
ALL REQUIREMENTS MIN UNLESS STATED							
SPECIFICATION	ESS415C REV E			SIZE OF TEST PIECE	110mm dia TEST PIECE		

TENSILE: ASTM A370 2010						0.2% PROOF	UTS			ELONG ON	%	R OF A %	COMMENTS
O	TEMP	SECTION	AREA SQ	UNIT	REQUIREMENTS	125000 - 140000	>=140000				>=14	>=40	
L	RT	12.72mm	127.08mm²	PSI	RESULTS	136000	155000			50.8mm	19.0	58.0	1/4 T

IMPACT: ASTM A370 2010	O	TEMP	SECTION	NOTCH	UNIT	REQUIREMENTS	SINGLE 20 AVE 27			LAT EXPANSION MM	SHEAR %	
CHARPY	L	-32°C	10 X 10 MM	2MM V	JOULES	RESULTS	70	78	78	N/A	N/A	1/4T
CHARPY NOMINAL VALUE 300J												

IMPACT: ASTM A370 2010	O	TEMP	SECTION	NOTCH	UNIT	REQUIREMENTS	SINGLE 32 AVE 42			LAT EXPANSION MM	SHEAR %	
CHARPY	L	-10°C	10 X 10 MM	2MM V	JOULES	RESULTS	81	83	81	N/A	N/A	1/4T
CHARPY NOMINAL VALUE 300J												

IMPACT: ASTM A370 2010	O	TEMP	SECTION	NOTCH	UNIT	REQUIREMENTS	SINGLE 47 AVE 54			LAT EXPANSION MM	SHEAR %	
CHARPY	L	RT	10 X 10 MM	2MM V	JOULES	RESULTS	88	84	88	N/A	N/A	1/4 T
CHARPY NOMINAL VALUE 300J												

HARDNESS: ASTM E10 2010	DESIGNATION	REQUIREMENTS	RESULTS
BRINELL (1/4T-Test Piece)	HBW 10/3000	285 - 341	318
BRINELL (Surface)	HBW 10/3000	285 - 341	321

METALOGRAPHY	RESULTS
GRAIN SIZE: ASTM E112-10	TO FOLLOW

REPRESENTING
Engineering Special Steels
S033597 02433/PRD
11 off 110MM DIA X 26FT LONG APPROX - BATCH M2050

HEAT TREATMENT Customer	TEMP °C	SOAK	QUENCH	START	FINISH
Harden	870	3:13 Hrs	Water	19	28
Temper	590	6:10 Hrs	Air	0	0

No copy is authentic without the embossed seal and signature. This certificate can be authenticated at www.specialtesting.co.uk with the code 677E432E21E1

For and on behalf of
SPECIAL TESTING LTD
A Member of the Special Steel Group of Companies

Steve Pearce
Certification Manager

Test results do not in any way confer approval of the quality of manufacture of the material. All results are subject to uncertainty of measurement budgets, details available upon request.

Page 1 of 2

Special Testing Ltd
Bacon Lane, Sheffield S9 3NH
Tel: 0114 244 1061
Fax: 0114 244 0444 Works
Fax: 0114 244 5566 Acc/Admin
www.specialtesting.co.uk

UKAS TESTING 0046

Test Certificate 188400

AUTHENTICATION CODE 677E432E21E1
ROLLS ROYCE APPROVED LABORATORY
Approval No. 11191

CUSTOMER
SPECIAL STEELS LIMITED
BACON LANE
ATTERCLIFFE
SHEFFIELD S9 3NH
DATE RECEIVED 27 September 2011
TEST DATE 28 September 2011

CUSTOMER ORDER NO	260195A BR	TEST NUMBER	188400	IDENTITY	260195A BR	CAST	T29183
ALL REQUIREMENTS MIN UNLESS STATED							
SPECIFICATION	ESS415C REV E			SIZE OF TEST PIECE	110mm dia TEST PIECE		

CLEANNESS: ASTM E45-2010 €1	TO FOLLOW
FERRITE COUNT (STANDARD): N/A	TO FOLLOW

END OF CERTIFICATE

REPRESENTING
Engineering Special Steels
S033597 02433/PRD
11 off 110MM DIA X 26FT LONG APPROX - BATCH M2050

HEAT TREATMENT Customer	TEMP °C	SOAK	QUENCH	START	FINISH
Harden	870	3:13 Hrs	Water	19	28
Temper	590	6:10 Hrs	Air	0	0

No copy is authentic without the embossed seal and signature. This certificate can be authenticated at www.specialtesting.co.uk with the code 677E432E21E1

For and on behalf of
SPECIAL TESTING LTD
A Member of the Special Steel Group of Companies

Steve Pearce
Certification Manager

Test results do not in any way confer approval of the quality of manufacture of the material. All results are subject to uncertainty of measurement budgets, details available upon request.

Page 2 of 2

Special Testing Ltd.
Metallurgical Investigation Section

Mechanical Testing, Metallography
Corrosion and Failure Investigation Specialists

Bacon Lane, Sheffield S9 3NH
Tel : 0114 244 1061
Fax : 0114 244 0444 (Works)
Fax : 0114 244 5566 (Admin/Accounts)

TEST REPORT

Met Page No. 1 of 2

CUSTOMER : SPECIAL STEEL Co. LTD.
ADDRESS : BACON LANE
SHEFFIELD S9 3NH.
CUSTOMER ORDER No : 260195A
STL TEST No. : ***188400***
SAMPLE IDENTITY : Ref. S033597 02433/PRD. CAST No. T29183.

TEST CONDITIONS
MATERIAL SPECIFICATION : ESS415C Rev E.
SECTION : LONGITUDINAL.
SAMPLE AREA EXAMINED : 160mm ². **FIELD SIZE :** 0.50mm².
EXAMINED AT : x100.

TEST RESULTS TO ASTM E45:2010^{e1} METHOD A (Plate 1-r)

WORST-FIELD RATING							
TYPE A		TYPE B		TYPE C		TYPE D	
T	H	T	H	T	H	T	H
½	0	0	0	0	0	½	0

SIGNED : A D Morton **DATE COMPLETED :** 29 / 09 / 2011

A.D.Morton (Metallurgical Laboratory Manager) - For and on behalf of Special Testing Ltd.

UKAS TESTING 0046

ONLY AUTHENTIC IF EMBOSSED BY STL
A Member of the Special Steel Group of Companies

ROLLS ROYCE APPROVED LABORATORY
Approval No. 11191

Special Testing Ltd.
Metallurgical Investigation Section

Mechanical Testing, Metallography
Corrosion and Failure Investigation Specialists

Bacon Lane, Sheffield S9 3NH
Tel : 0114 244 1061
Fax : 0114 244 0444 (Works)
Fax : 0114 244 5566 (Admin/Accounts)

TEST REPORT

Met Page No. 2 of 2

CUSTOMER : SPECIAL STEEL Co. LTD.
ADDRESS : BACON LANE
SHEFFIELD S9 3NH.
CUSTOMER ORDER No : 260195A
STL TEST No. : *188400*
SAMPLE IDENTITY : Ref. S033597 02433/PRD. CAST No. T29183.

TEST CONDITIONS

MATERIAL SPECIFICATION : ESS415C Rev E.
SECTION : LONGITUDINAL.
ETCHANT'S USED : 40% NaOH (Electrolitically) & HCl / H_2O_2.
EXAMINED AT : x100 & x200.

TEST RESULTS

% FERRITE : ASTM E562:2008	No evidence of ferrite was present in the fields rated. 0%.
GRAIN SIZE : ASTM E112: 2010 (COMPARISON METHOD).	Range - 6 to 8. Average - 7.

SIGNED : A D Morton **DATE COMPLETED :** 29 / 09 / 2011

A.D.Morton (Metallurgical Laboratory Manager) - For and on behalf of Special Testing Ltd.

UKAS TESTING 0046

ONLY AUTHENTIC IF EMBOSSED BY STL
A Member of the Special Steel Group of Companies

ROLLS ROYCE
APPROVED LABORATORY
Approval No. 11191

STW (Non-Destructive) Ltd

STW (Non-Destructive) Ltd
Bacon lane,
Sheffield S9 3NH
England
Tel . +44 (0) 114 2441061
Fax. +44 (0) 114 2440444
www.specialtesting.co.uk
sales@specialsteelgroup.com

ULTRASONIC TESTING REPORT FORM

Date: 03/10/2011 **N.D.T Ref No:** CB2997

Customer: ENGINEERING SPECIAL STEELS Ltd.

Contract No: 260195A **Your Ref:** SO33597
02433/PRD

Description of Item: 11 BARS 110mm∅ X 26' LONG (5355Kgs)

Cast No: T29183

Material: 4145 **Surface Condition:** HEAT TREATED

Code of Acceptance: ESS 415C Rev. E
API 6A PSL 3

Flaw Detector Type: KK USN 58 L

	Coverage of Scan	Crystal			Angle (°)	Couplant	Sensitivity
		Probe	Type	FQY			
1	180° x Full Length	SEB2	TWIN	2 MHz	0	PASTE	1/8" FBH DAC +6dBs

Calibration Blocks: IIW A2 BLOCK, 1/8" FBH BLOCK

Result of Test: NO RECORDABLE INDICATIONS WERE LOCATED
MATERIAL IS ACCEPTABLE TO SPECIFICATION.

Signed: CBoreham

Operator: C. BOREHAM

Operator Approvals: PCN / SNT LEVEL II

Certificate No. Q06199

A Member of The Special Steel Group of Companies

Registered Number 1573493

ISSUED BY

Page: 1 of 1

Number:2011/08/001447-PER

SOCHOROVÁ VÁLCOVNA TŽ

E.S.S. BATCH No: M2050

SOCHOROVÁ VÁLCOVNA TŽ, a.s. / Průmyslová 1000/ 739 70 Třinec - Staré Město / Czech republic/
výrobní závod: 272 01 Kladno-Dříň

Special Steels Co Ltd

Bessemer Road
S9 3XN Sheffield
United Kingdom

Ihr Auftrag - Your order:
KL10503
8760
Aviso - Advise note:
oder/or
EU-Lieferschein Nr.- Delivery note:
11/08/000597/03 04.08.2011

Werks Nr. - Our order:
7000314657 / 000010
0041170482 / 306
Waggon No:

3154-0807462-5

Abnahmeprüfzeugnis 3.1- Inspection certificate 3.1 - Certificat de réception 3.1, EN 10204:2004

Lieferung - Delivery - Livraison	Gewicht - Weight -Poids	Güte -Grade -Qualité	Norm - Standard - Norme
HOT ROLLED ROUND PEELED BARS AR QUALITY: AISI 4140H MOD./AISI 4145H MOD. ORDER NO. KL10503/8760 110 mm 6-7 m	5355 kg	AISI4140H/ 4145Hmod natural	SAE J1268:1995

CHEMISCHE ZUSAMMENSETZUNG - CHEMICAL ANALYSIS - COMPOSITION CHIMIQUE (%)

Schmelze Heat No Charge No	Gewicht Mass [kg]								
T29183	5355	C	Mn	Si	P	S	Cr	Ni	Cu
BO- cc blooms		0.43	0.96	0.30	0.010	0.006	1.07	0.22	0.03
		Mo	Al	Sn	V	Ti	N	H	
		0.310	0.028	0.004	0.006	0.0022	0.0065	1.40 ppm	

MECHANISCHE WERTE - MECHANICAL VALUES - QUALITES MECHANIQUES

Schmelze Heat No Charge No	Streckgrenze Yield point Limité d'elast. R [MPa]	Zugfestigkeit Tensile strength Resistance Rm [MPa]	Bruchdehnung Elongation Allongem A5 [%]	Brucheinschn. Contraction Contraction Z [%]	Schlagarbeit. Energy of impact Resilience KV [J]	Reinheitsgrad nach Microcleaness accord.to DIN 50 602 K4	Harte Hardness HB	
T29183	1126	1238	15,4	49,2			278 -U.	

```
Microcleanliness Rating ASTM E - 45/05:
           TYPE  A      TYPE  B      TYPE  C      TYPE  D
          THIN THICK   THIN THICK   THIN THICK   THIN THICK
   max:    2,0  2,0    0,5 0,5      0,5  0,5     1,5  0,5
Grain size ASTM E 112 - 5-8
Test sample D 10 mm  quenched and tempered-BD.
Temperatures: quenching:870°C,tempering:540°C
Ultrasonic inspection results ASTM A388/M API PSL LEVEL 3  - satisfactory
Spectrotest  100 %
Vacuum degassed practice.
Rolling reduction ratio =13,9
Mode of production oxygen converter
```

DIE RADIOISOTOPISCHEAKTIVITÄT MAX. 100 BQ/KG WURDE EINGEHALTET.
RADIO - ISOTOPIC ACTIVITY WAS CONTROLLED IN LADLE SAMPLE, IT WAS NO HIGHER THAN 100 BQ/KG.

Dieses Dokument wurde mit elektronischer Unterschrift im Einklang mit dem Gesetz Nr. 227/2000Slg. versehen.
This document was electronically according to Law No. 227/2000Coll undersigned.

ENGINEERING SPECIAL STEELS LTD
CERTIFIED TRUE COPY
OF THE ORIGINAL
CHECKED BY Q.A. DEPT.

Die obengenannten Erzeugnisse entsprechen den Bestellungsvorschriften - Products conform with the prescription of order.

Der Werksachverständige - Expert: Eva Perglerová,

Kladno:05.08.2011

Der Sachbearbeiter der Qualitätskontrolle fur Freigabe und Atteste, unabhängiger berechtigter Vertreter
Officer of Quality Inspection of Realising and Attesting, independent authorized agent

Tata Steel UK Limited
Stocksbridge
Sheffield
S36 2JA
United Kingdom
Telephone: +44 (0) 114 2882361
Fax: +44 (0) 114 2832079
Website: www.tatasteel.com

E.S.S. BATCH NO: M2442

INSPECTION CERTIFICATE/TYPE 3.1 to BS EN10204

ENGINEERING SPECIAL STEELS LIMITED
VICTORIA WORKS
31 CATLEY ROAD
DARNALL
SHEFFIELD

Cast No. **B4011A**

Works Order No. CB831704
Customer Order No. 008893

Date of Issue 19-DEC-2011
Certificate No. 00353023/1
Page No. 1 of 2

SPECIFICATIONS ASSOCIATED WITH THIS ORDER -
4145 TO ESS415A REV X
ENGINEERING SPECIAL STEELS ESS415A REV X

PRODUCT INFORMATION -
SIZE - 0310.000 MM DIA
QUANTITY/WEIGHT - As per advice note
CONDITION OF MATERIAL - COLD STR, FINALLY SMOOTH TURN, H&T
STEELMAKING PROCESS/PROCESS OF MANUFACTURE - Electric VDG Ingot

HEAT TREATMENT OF MATERIAL - W61390
Harden . at 880° C; for 02:00; time to 04:52; total time 06:52; Water Quenched from 12° C.
Temper at 620° C; for 03:00; time to 05:01; total time 08:01; Air Cooled.

ANALYSIS -

Cast No	C	Si	Mn	P	S	Cr	Mo	Ni	Cu	Sn	Al
Cast Analysis											
B4011A	.47	.27	1.10	.010	.010	1.18	.34	.23	.15	.015	.029

Mechanical Test - Tested to ASTM A370 - 11 + ASTM E8/E8M - 09

	Test No.	Ingot Id	Ingot Pos.	Sample Pos.	Orient	Temp.	0.2% PS	U.T.S	Elong 4D	R of A	Hardness
Units						C	PSI	PSI	%	%	HBW10/3000
Result	1786200		M	1" Below Surface	Lo	23	130000	152000	18	55	321

Impact Test - Tested to ASTM A370 - 11 + ASTM E23 - 07a (E1)

	Test No.	Ingot Id	Ingot Pos.	Sample Pos.	Geometry	Orient	Temp.	Imp Mean	Imp	Imp	Imp
Units							C	J	J	J	J
Result	1786201		M	1" Below Surface	CH2MMV	Lo	-10	66	65	69	65
	1786202		M	1" Below Surface	CH2MMV	Lo	-32	51	52	58	42

MacQuaid Ehn Grain Size - Tested to ASTM E112 - 10

	Test No.	Ingot Id	Ingot Pos.	G.Size	G.Size
Result	1786199	15	M	6	8

ENGINEERING SPECIAL STEELS LTD
CERTIFIED TRUE COPY
OF THE ORIGINAL
CHECKED BY Q.A. DEPT.

Tata Steel UK Limited
Stocksbridge
Sheffield
S36 2JA
United Kingdom
Telephone: +44 (0) 114 2882361
Fax: +44 (0) 114 2832079
Website: www.tatasteel.com

E.S.S. BATCH NO: M2442

INSPECTION CERTIFICATE/TYPE 3.1 to BS EN10204

Cast No. **B4011A**	Works Order No. CB831704	Date of Issue 19-DEC-2011
	Customer Order No. 008893	Certificate No. 00353023/1
		Page No. 2 of 2

Surface Hardness -

	Test No.	Ingot Id	Ingot Pos.	Hardness	Hardness
Units				HBW10/3000	HBW10/3000
Result	1786205			285	285

Ultrasonic Test - Tested to ASTM A388/A388M - 10
Satisfactory to 12.0mm FBH
ULTRASONIC MEETS API6A CLAUSE 7.4.2.3.15 PSL 3
"100% U/S TESTED TO ASTM A388" - REF 7UF-RLSTD

MISCELLANEOUS INFORMATION -
REDUCTION RATIO: 3.64
Material type tested satisfactorily
Method of Analysis
- Elemental Analysis (combustion/fusion) C S
- Atomic Emission Spectroscopy (OES) Si Mn P Cr Mo Ni Cu Sn Al

Our quality management system is accredited to the following standards : ISO9001, ASEN9100 and ISO17025.
Our environmental management system is accredited to ISO14001.

AUTHORISED SIGNATURE -
Certified by our Stocksbridge Works that, unless otherwise stated above, the whole of the above mentioned materials have been manufactured, tested & inspected in accordance with the terms of the acknowledged contract/order applicable thereto and conform fully to the standards/specifications quoted hereon.
Approved Signatory - Lee Ibbitson - Certification and Accreditation Manager

Signed......................................

For Tata Steel UK Limited

End Of Certificate

ENGINEERING SPECIAL STEELS LTD
CERTIFIED TRUE COPY
OF THE ORIGINAL
CHECKED BY Q.A. DEPT.

Titanium material certificates

Material certificates for the High Strength Titanium have been redacted due to a pending patend application. The high strength titanium used in the prototype MSJ have properties equal or higher than the reported typical values from Table 3.1.2 in Chapter 3.1.