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ABSTRACT

The purpose of this work is to discuss the structural limitations of the unbounded flexible pipe technology, used to support offshore production activities in high pressure fields.

In addition, the thesis aims to describe and analyse state-of-art design solutions and elaborates on a development of a novel simplified methodology for early fatigue severity indicator (based on the ongoing discussions carried out by Technip).

The work contributes to the internship at the FlexiFrance/Technip, Product Engineering Division, Le Trait, France. The internship took place from 19th of January to 29th of May 2015.

As modern drilling seeks for deeper waters and high pressure/high temperature submarine formations, the producing companies require higher standards of the production systems, increasing the operating pressure ratings up to 20 000 psi. This provokes usage of more robust materials, increasing challenges for the flexible pipe fatigue.

In order to address the whole range of the flexible pipe applications within a production circuit the following basic design drivers are considered in the project: a) rough/smooth bore structures, b) sour/sweet services, c) dynamic/static applications.

The work is based on Technip’s expertise; however it broadens the referential, on the independent intiative and view of the author, base to the experiences of other companies, SPE & OTC publications, UiS courses, books e.t.c..

The following methods are incorporated in the project:

- Use of data simulated with Technip software (with following analysis);
- Overview of the existing studies on the subject, both public and private;
- Communication with UiS faculties and Technip onsite-personnel;
- Review of information available on specified libraries (internet, specific literature).

The most prolific results of the study are:

- Overview (learning) of the state-of-art for flexible pipes use in offshore production systems;
- Problem stipulation (high pressure performance, fatigue analysis);
- Research for the different operational limitations (scope of assumptions for the research);
- Alternatives for the flexible pipe design/future design improvements;
- Proposal of a novel simplified methodology for early fatigue severity indicator (based on the structural capacities and not on the designated local fatigue analysis/finite element analysis).

Major findings and conclusions are the definitions of the operating conditions for the flexible pipe in high pressure offshore areas, fatigue assessment methodology based on the results of the internship.

The student has however contributed the work with:

- Optimizing a Technip solution by selecting materials and structural elements of flexible pipes;
- Designing the flexible pipes with Technip in-house software;
- Presenting results and structural limitations;
- Synthetyzing fatigue analysis reports from Technip global units;
- Performing fatigue design of the flexibles based on data collected;
- Proposal of a novel indicative methodology for the fatigue performance the fatigue performance.

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ACRONYMS

CFA - Carbon Fiber Armour;

CFC - Carbon Fiber Composite;

DAF - Dynamic Amplification Factor;

DCC - Critical curvature;

DRAPS - Drilling & Refining Applications Division;

FAT - Factory Acceptance Test;

FPSO - Floating Production Storage Offloading;

FSHR - Free Standing Hybrid Riser;

GOM - Gulf of Mexico;

HDPE - High-Density Polyethylene;

HIC - Hydrogen Induced Cracking;

HPHT - High Pressure/High Temperature;

JIP - Joint Industry Project;

LF - Low Frequency;

LNG - Liquefied Natural Gas;

LRFD - Load And Resistance Factor Design;

LTMU - Le Trait Manufacturing Unit;

PSA - Petroleum Safety Authorities;

PVDF – Polyvinyldiflouride;

SCF - Stress Concentration Factor;

SLPM - Service Life Prediction Model;

SMYS - Specified Minimum Yield Strength;

SSC - Sulfide Stress Cracking;

SSC - Sulfide Stress Cracking;

TDP - Touch Down Point;

TLP - Tension Leg Platform;

Tpe - Polyolefin Based Polymer;

UF - Utilization Factor;

UTS - Ultimate Tensile Stress;

VIV - Vortex Induced Vibrations;

WD - Water Depth;

WF - Wave Frequency;

WSD - Working Stress Design.

ACKNOWLEDGMENTS

This thesis aims to conclude the author’s education at the University of Stavanger (Master of Science in Marine and Subsea Technology). The thesis work has mainly been performed at the Technip Product Engineering Division in France. Several people have entusiastically contributed to the project work.

First and foremost, I would like to express my very deep appreciation of time Prof. Ove Tobias Gudmestad has devoted during the thesis preparation. With his encouragment I was able to apply my engineering skills in a unique ambient of FlexiFrance. He from the very beginning believed in the success of the venture and has relentlessly helped me throughout the process. His support and valuable comments are therefore gratefully acknowledged. Moreover, I would like to admit his genuine passion for innovations and a company visits’ initiative. Without his guidlines the study process would be impossible.

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I would like to thank Mr. Philippe Secher, Mr. Pascal. Estrier and Mr. Laurent Paumier for providing unique experties, their supportivenes and cheerfullness during the stay at Flexi/France.

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Last but not least, I would like to thank my parents (Valentina and Igor) and Ksenia Galkina for their direct support and encouragments.

INTRODUCTION

Recently and particularily in the Gulf of Mexico and in the North Sea, high pressure offshore discoveries were made. While the drilling rigs and the well intervention vessels are traditionally fitted with 15 kpsi equipments, many rigs are now being upgraded to operate beyond this value.

Technip has long been providing 15 kpsi rated flexible pipes and is now moving toward 20 kpsi ratings [10].

In this thesis, the structural limitations of the unbounded flexible pipe technology with emphasis on high pressure applications will be shown. The reader will be provided with state-of art-solutions and challenges, that are being widely discussed among the industry.

The document will also cover a development of a novel simplified methodology for early fatigue severity indicator.

This paper contains the following chapters:

Chapter 1 discovers the challenges arising with the use of flexible pipe in high pressure projects. In addition, it gives a brief introduction to a flexible pipe technology industrial limitations and future trends.

Chapter 2 elaborates on basic design considerations for the flexible pipe engineer. Special emphasis is made on high pressure design criteria and failure modes.

Chapter 3 introduces an upgraded pressure reinforsement construction and gives a detailed analysis of structural capacities' calculations. This new improvement, however, causes the need for a detailed fatigue study.

Chapter 4 establishes fatigue performance indicators and gives a synthesis of fatigue behaviour of flexible pipes used in different regions of the world. Chapter also suggests a novel methodology for an early fatigue properties indicator. Reader will also be provided with the comprehensive introduction to basic fatigue design principles.

Finally, Chapter 5 concludes the study with the most prolific results and indicates the areas of further interest.

Project involved various activities, such as: learning of Technip best design practices and the industial know-how, applying those practices in a design engineering, collaboration with different global units and departments in France. Moreover, project required a lot of analysis and creativity from the mentors and the student himself.

As some part of the information taken is from a private company, it is censored, wherever required (with the assumptions stipulated respectively). The information from SPE & OTC publications, UiS courses and books is assumed to be trustworthy, unless other assumptions are mentioned.

CHAPTER 1 FLEXIBLE PIPE TECHNOLOGY. HIGH PRESSURE HORIZONS

1.1 Introduction

Development of offshore projects today applies several unprecedented challenges to oil and gas companies, such as:

1. Deeper waters;
2. Petroleum reservoirs with higher temperature and pressure;
3. Fields located in regions with a colder climates (Artic);
4. Installation of subsea processing equipment;
5. Long distance from infrastructure.

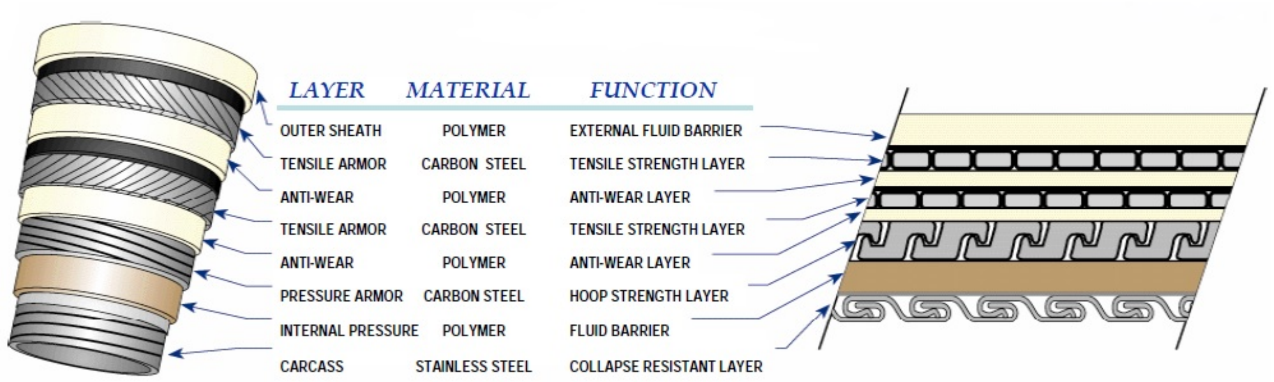
New challenges arising with the use of flexible pipe (riser or jumper) in High pressure, High temperature projects will be described in the following chapter. In addition, reader will have a brief introduction to a flexible pipe technology and will also have a chance to meet its current capability with track records and future trends [15].

1.2 Flexible pipe structure

A typical flexible pipe structure, for high pressure applications is shown in Figure 1.1 below.

This construction is known as a “unbonded” structure.

From the inside out it is composed of the following:



LAYER	MATERIAL	FUNCTION
OUTER SHEATH	POLYMER	EXTERNAL FLUID BARRIER
TENSILE ARMOR	CARBON STEEL	TENSILE STRENGTH LAYER
ANTI-WEAR	POLYMER	ANTI-WEAR LAYER
TENSILE ARMOR	CARBON STEEL	TENSILE STRENGTH LAYER
ANTI-WEAR	POLYMER	ANTI-WEAR LAYER
PRESSURE ARMOR	CARBON STEEL	HOOP STRENGTH LAYER
INTERNAL PRESSURE	POLYMER	FLUID BARRIER
CARCASS	STAINLESS STEEL	COLLAPSE RESISTANT LAYER

Figure 1.1 - Layers of the flexible pipe [7]

1.2.1 Carcass

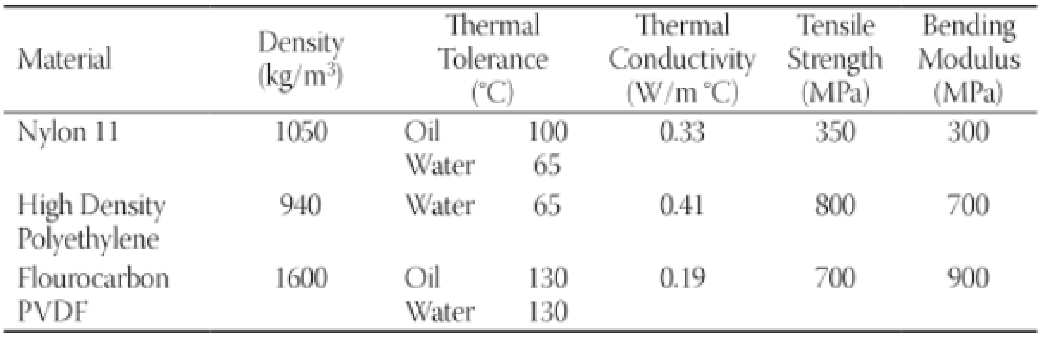
The innermost layer is the carcass, which is an interlocked flexible steel structure providing collapse resistance to the pipe against external hydrostatic pressure and crushing loads. As the carcass is not a leak-tight structure a polymeric inner liner (pressure sheath) is extruded around it [8].

1.2.2 Thermoplastic inner liner

The thermoplastic inner liner makes the pipe leak-proof. This layer limits the upper service temperature of the line and the chemical compatibility to the various fluids which may be transported through the line. Various plastic materials are used to manufacture the liner, depending upon the service application of the line. This liner can be reinforced by aramid tapes depending on the application.

The liner contains the hydrocarbon fluids. Liners are fabricated from high-density polyethylene, nylon and fluorinated polymers. The factor that determines the service life of the flexible line is its degradation that occurs as a result of reaction with components in the hydrocarbon stream. The operating temperature is a major factor in this rate of degradation and consequently the choice of polymer depends on the service temperature. For low temperatures or low water content fluids, high-density polyethylene (HDPE) and polyamide (nylon) liners are used. These materials are suitable to about 65 °C and 95 °C, respectively, though the precise limit depends on manufacturing details and should always be confirmed by the manufacturer. At the higher temperatures (to 130 °C) and high-water cut fluids, a more thermally stable liner is required. Suitable polymer solution is the polyvinyldiflouride (PVDF). The minimum commonly considered temperatures for these materials are -50 °C for HDPE and -20 °C for nylon and the fluorinated polymers. Table 1.1 lists the mechanical properties of liner materials [31].

Table 1.1 – Typical properties of thermoplastic liner materials [31]



Material	Density (kg/m^3)	Thermal Tolerance (°C)		Thermal Conductivity (W/m °C)	Tensile Strength (MPa)	Bending Modulus (MPa)
Nylon 11	1050	Oil Water	100 65	0.33	350	300
High Density Polyethylene	940	Water	65	0.41	800	700
Flourocarbon PVDF	1600	Oil Water	130 130	0.19	700	900

1.2.3 Interlocked pressure armour layer

This layer takes the hoop stress due to internal pressure and external crushing loads.

Interlocking of the pressure armour layer (typically Zeta shape) is a key parameter to define the minimum allowable bending radius of the line. The loss of interlocking (unlocking) due to excessive curvature is a severe and irreversible damage to the flexible line creating conditions for loss of leak-proofness and leakage of the flexible pipe.

1.2.4 Metallic reinforcement of the interlocked pressure armour layer

If necessary to extend pressure capability of the pipe, the Zeta layer is reinforced by a flat steel layer which is not interlocked. In this paper the effect of two flat steel spirals will be discussed.

1.2.5 Anti-wear thermoplastic layer

A thin anti-wear layer (tape or sheath) is added in the flexible pipe construction for dynamic applications in order to prevent wear between layers. This layer is not leak-proof.

1.2.6 Double cross-wound steel tensile armour

The double cross-wound steel tensile armour wires will resist axial load caused by internal pressure, or external axial loads. It is also this layer which provides the flexible line with its resistance to torsion.

1.2.7 Thermoplastic outer sheath

This layer is leak-proof. It will protect the armour wires against corrosion from seawater [35].

1.2.8 End-fitting

Each of the structural flexible pipe layers must be individually terminated to maintain fluid-tight integrity and to sustain the imposed loads. That is the role of the end fittings which are carbon steel force-resistant parts designated to terminate the ends of each of the layers of the flexible pipe, maintain the integrity of the pipe structure and transfer the loads. In particular, the end terminations include seals to ensure a reliable fluid-tight seal to the internal thermoplastic layer and the tight seal to outer thermoplastic layer (please see Figure 1.2, Figure 1.3). End-fittings are internally and externally coated for corrosion protection purpose [20].

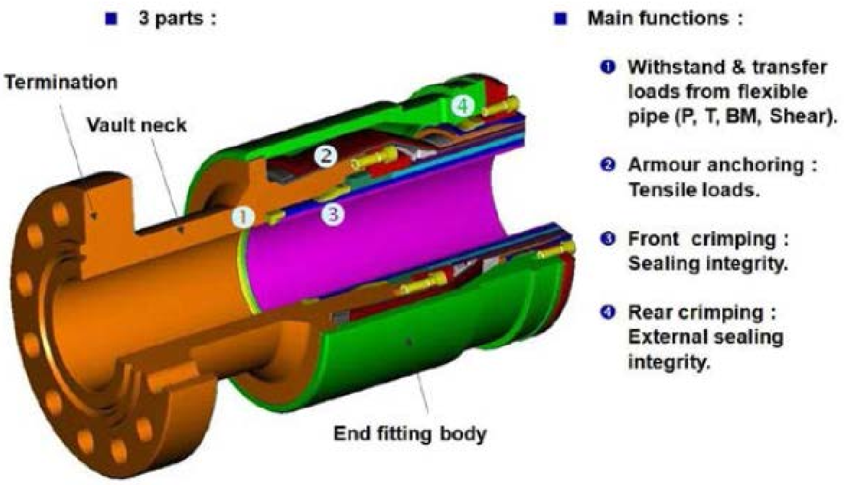


Figure 1.2 – Schematic 3D view of the end-fitting and a description of its main functions [31]

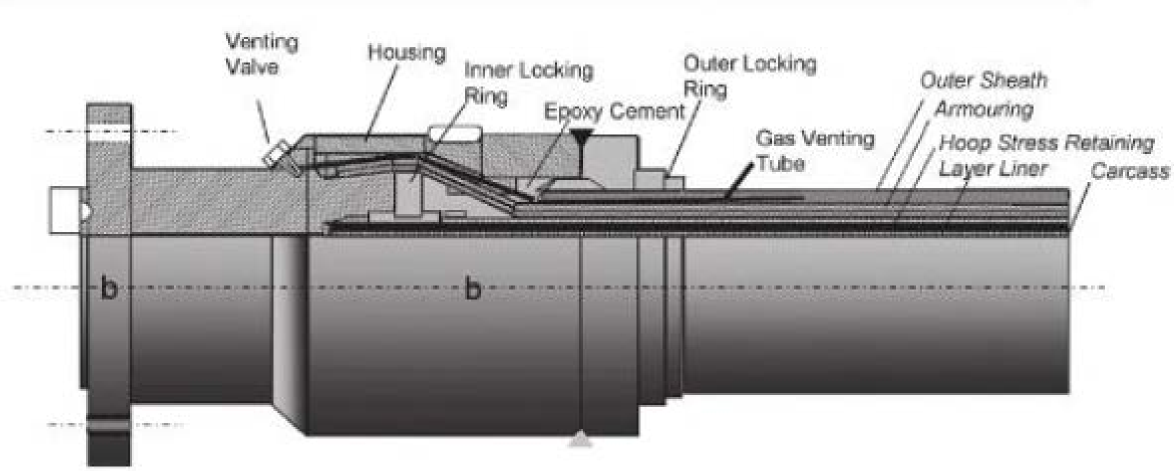


Figure 1.3– Sheath pressure relief systems for flexible pipelines [31]

However, depending on the application, flexible pipes can be subdivided into two big categories (rough and smooth bore), see Figure 1.4.

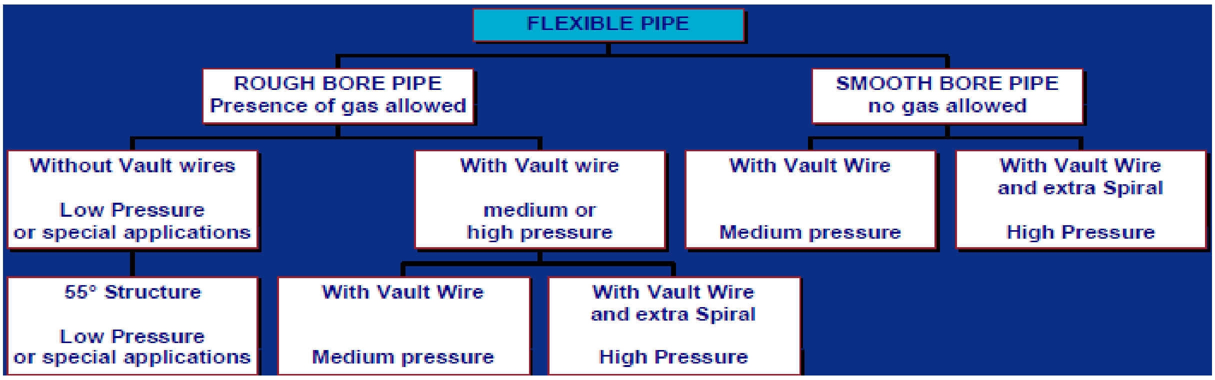


Figure 1.4– Flexible pipe structure layers [18]

Rough bore pipes with vault wire and extra spirals will be elaborated upon in the course of this project.

1.3 Flexible pipe manufacturers

The main flexible pipe manufacturers in the world are as follows:

NKT Flexibles (today is a part of the National Oilwell Varco group);

Wellstream (today is a part of General Electric Oil and Gas group);

Technip (formerly Coflexip).

NKT flexibles are classified as follows:

1. Low-pressure smooth bore;
2. Low-pressure rough bore;
3. High-pressure smooth bore;
4. High pressure rough bore.

Wellstream typical products are:

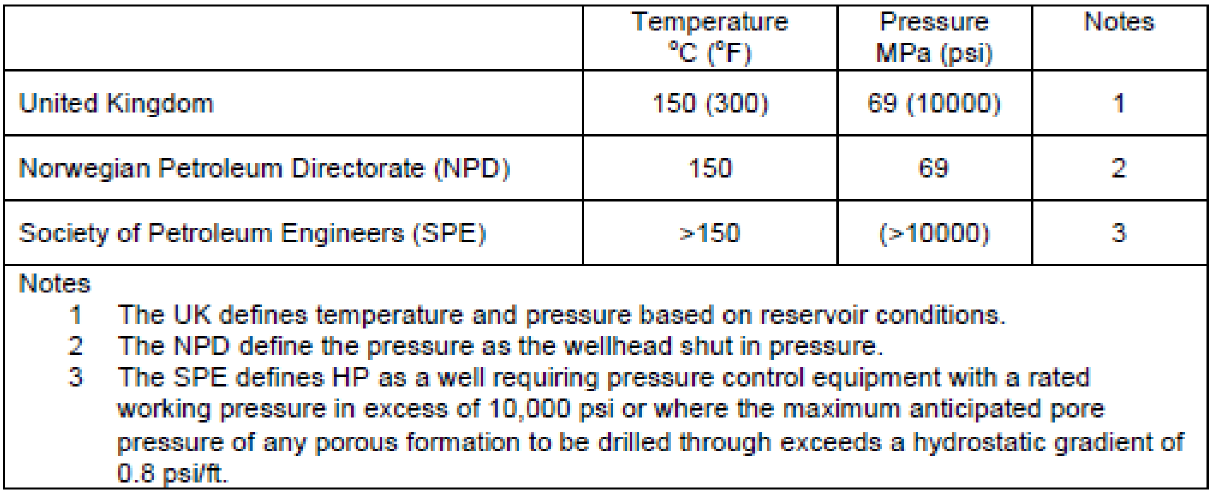
1. Low internal pressure;
2. Thermal resistance;
3. High external pressure;
4. For corrosive internal fluids;
5. Prevention of external abrasion.

Technip flexible pipe has the largest market share in the world for this product. Technip has designs to cover all aspects of deepwater applications including corrosion resistance, high temperature, and pressure (liquefied natural gas (LNG) applications and actively heated flexible pipes) [21].

1.4 High pressure, high temperature definitions

High pressure/high temperature (HPHT) fields are defined primarily by their characteristic reservoir pressure and temperature. They are typically gas/condensate fields, though rare exceptions apply. The typical numerical definitions for high pressure and high temperature applied in the UK and Norwegian sectors are given in Table 1.2[25].

Table 1.2 - Definition of the HPHT development [25]



	Temperature °C (°F)	Pressure MPa (psi)	Notes
United Kingdom	150 (300)	69 (10000)	1
Norwegian Petroleum Directorate (NPD)	150	69	2
Society of Petroleum Engineers (SPE)	>150	(>10000)	3

Notes

1 The UK defines temperature and pressure based on reservoir conditions.
2 The NPD define the pressure as the wellhead shut in pressure.
3 The SPE defines HP as a well requiring pressure control equipment with a rated working pressure in excess of 10,000 psi or where the maximum anticipated pore pressure of any porous formation to be drilled through exceeds a hydrostatic gradient of 0.8 psi/ft.

1.5 High pressure, high temperature for flexible pipes

Absolute internal Design Pressure (P in psi) is only one indicator of the High Pressure domain. It dictates for example the material choice. However, it is also very important to consider the $P \times internal\,diameter\,(ID)$ factor (expressed in psi.inch) which has a direct effect on the pipe construction and is the driving parameter for mechanical sizing of pressure and tensile armours.

Although there are no defined limits, we usually speak of High Pressure when Design Pressure is above 10 000 psi and Very High Pressure when Design Pressure is above 15 000 psi. High Design

Pressure mainly influences the choice of polymers for pressure sheaths and anti-wear layers. This parameter is then leading the design of small diameter pipes.

For large diameter pipes, Design Pressure may be considered as High when large quantity of steel is needed in order to get mechanical resistance. In this case, High Pressure domain can be considered, when $P \times ID$ factor is above 60 000 psi.inch and Very High Pressure domain is considered for the factor above 80 000 psi.inch.

1.6 Trends in flexible pipe technology

Over the past 40 years, Technip has been delivering a large variety of flexible pipes, the vast majority being designed and manufactured for given specific field applications. Figure 1.5 below presents on a Design Pressure versus Internal Diameter graph the track record of flexible pipes that have been manufactured by Technip.

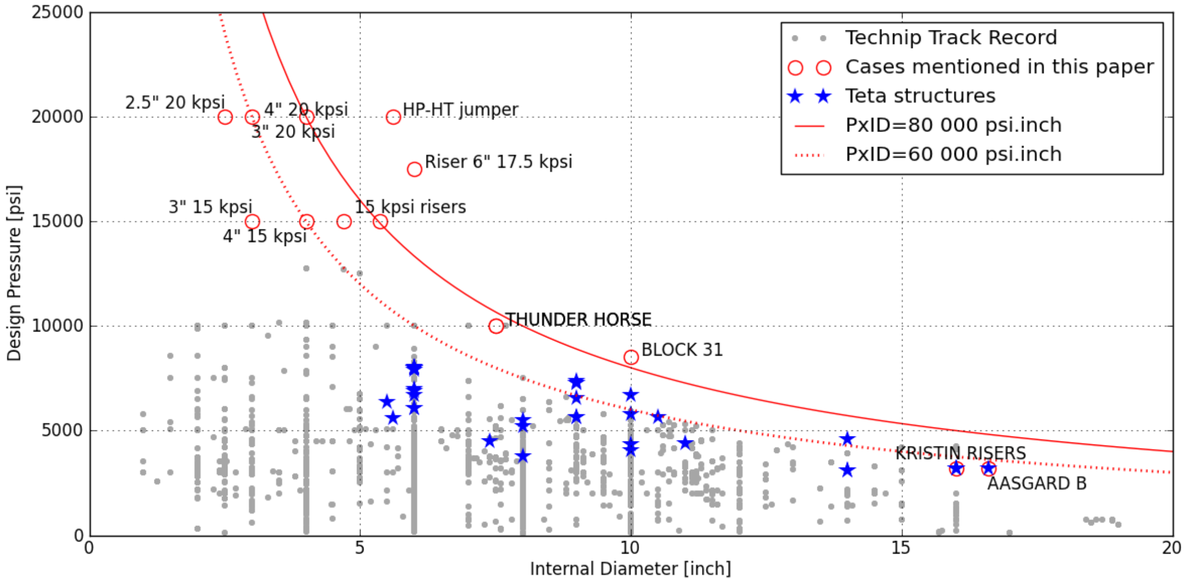


Figure 1.5– Design pressure versus internal diameter [23]

Thunder Horse project, with a $P \times ID$ of 75 000 psi.inch (7.5” × 10 000 psi water injection lines), was a reference project for the GoM. Kristin project in the Norwegian North Sea was also a very challenging project at that time with very high design temperature of 132 °C. For West Africa, the 10” ID water injection pipes with 8 500 psi Design Pressure for Block 31 FPSO are the pipes with the highest $P \times ID$ in operation in this region [10].

Maximum design pressure capacities for the sweet flexible risers delivered by one of Technip’s main competitor – Wellstream are presented below, Figure 1.6. However, the difference of the Maximum Design pressure and Design pressure will be discussed in further chapters. On the chart, Riser Max Design Pressure curve represents today’s limitation for the high pressure applications. One of the main tasks of the current study is to extend these pressure envelope and outline the limiting factors in the pipe structural design.

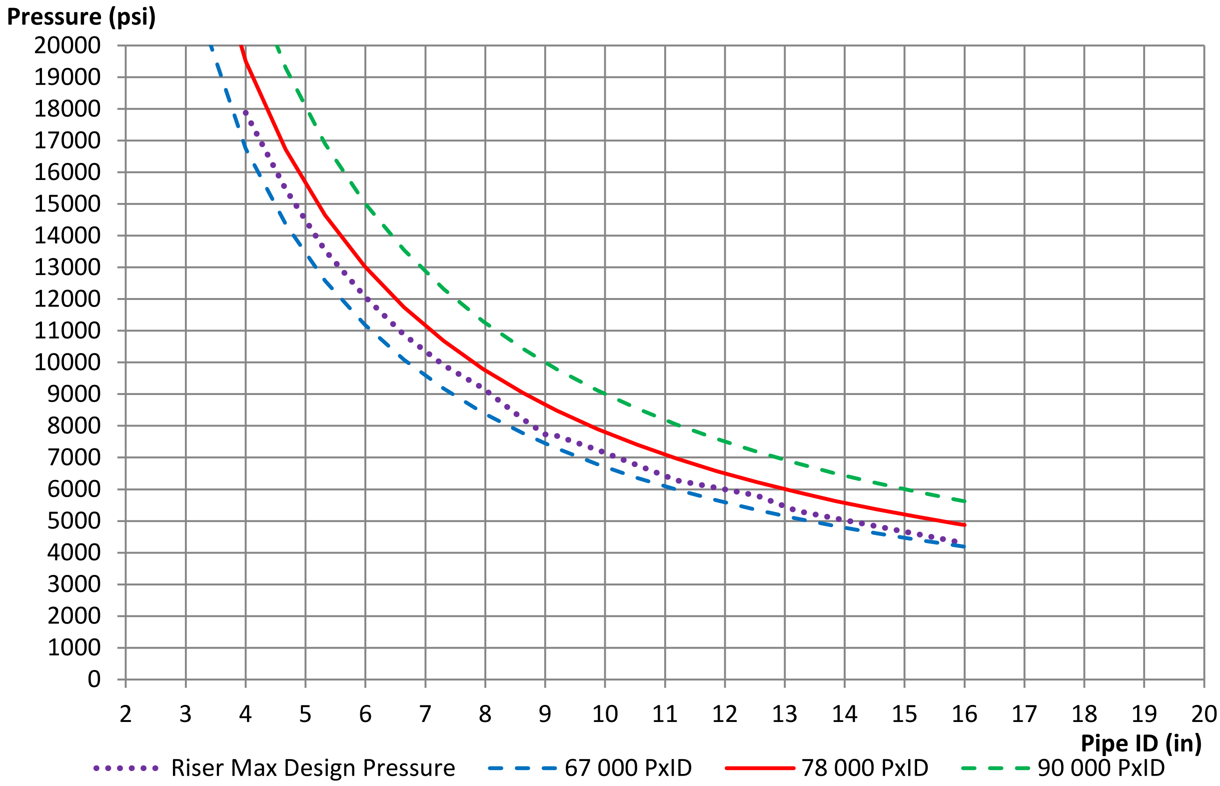


Figure 1.6 – Static and dynamic load resistance (modified) [4]

1.6.1 Composite materials for armour wires

The conventional structure of flexible pipe is facing extremely severe conditions in terms of dynamic loading, high H_2S and CO_2 contents and temperature. Design innovation is required to extend the envelopes of performance during the service production.

Indeed, when exposed to high tension variations in corrosive environment, the mechanical performance of steel components is restricted, especially in the presence of H_2S. Current state of art is to design more complex riser configurations such as lazy waves that require a large amount of buoyancy. This tends to generate extra costs and to slow down the flexible pipe installation, thereby adding cost to the overall as-installed solution.

In this situation, composite material will then contribute to improve the performances of the flexible pipes when it is exposed to fatigue and corrosion.

The performance of the composite material has already been presented in international conferences. It was shown that the fatigue behavior of carbon fiber composites (CFC) is exceptional since some of the fatigue tests on armours were carried out until 20 millions of cycles without failure at more than 50% of ultimate tensile stress (UTS) (Stress ratio R=0.3, Frequency 3Hz).

The main change suggested within a flexible pipe structure consists only in replacing conventional tensile armours, steel wires, by carbon fiber armours (CFA) [36].

1.6.2 Anti H_2S layer

In the recent years a new layer has been developed by Technip in order to stop the diffusion of H_2S from the bore to the annulus. With this new layer it is possible to select sweet service steel grades even for transportation of fluid containing H_2S. This layer is called the 'Anti H_2S layer', Figure 1.7.

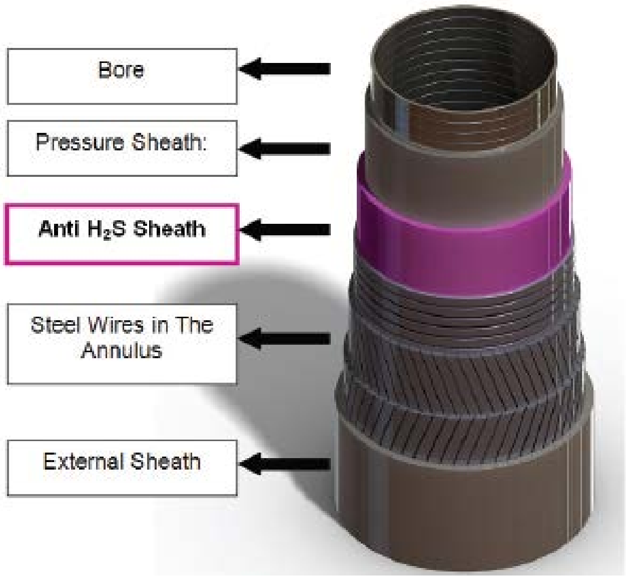


Figure 1.7– Flexible pipe with 'anti- H_2S layer' [17]

As this layer prevents the risks of sulfide stress cracking (SSC) or hydrogen induced cracking (HIC) in the annulus, it is possible to select sweet service steel grades with higher mechanical properties than sour service steel grades. This difference in mechanical properties of the steel grades selected for the structural layers can impact the overall design of the pipe. Figure 1.8 presents a schematic view of the positive impact of the use of the 'Anti H_2S' sheath on a flexible pipe design. On the pressure vault layer, the use of sweet service layer with higher mechanical properties will allow the use of wires with thinner dimensions. In some cases, it will also permit to avoid the use of a spiral layer between the pressure vault and the armour layer. On the armour layer, the same way of thinking applies: the use of high strength steel allows the use of thinner armour wires and in certain cases. It may even eliminate the requirement for a second pair of armours.

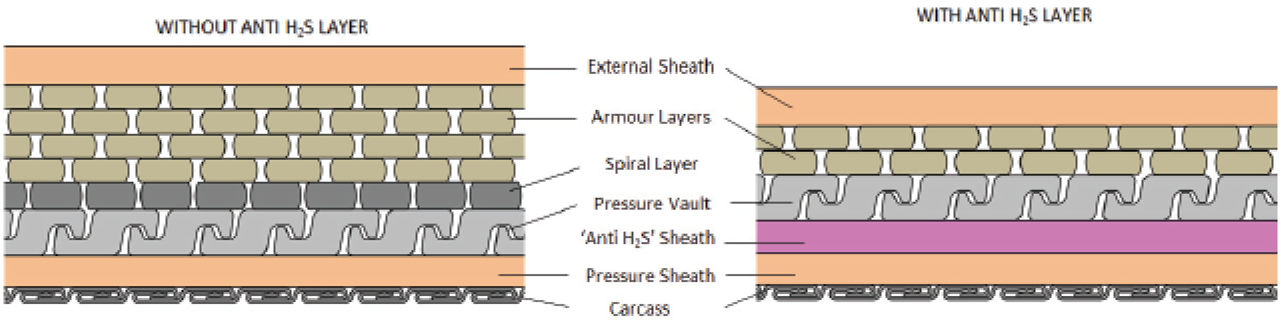


Figure 1.8– Example of the positive impact of the 'anti H_2S' layer on a flexible pipe design [17]

In addition to the decrease in the cost of the flexible pipe, all these differences have an impact on the weight of the product. Such advantages can lead to a reduction in the weight of the pipe up to 25% [17].

1.7 Overview of the chapter

These are the main subjects discussed in the Chapter 1:

1. Challenges in oil and gas industry with emphasis on HPHT developments;
2. Layer-by-layer description of the structural components of the flexible pipe;
3. Main manufacturers of the flexible pipe in the world;
4. Examples of future trends in the flexible pipe technology (utilization of the composite materials and anti H_2S layer).

CHAPTER 2 DESIGN CRITERIA FOR HIGH PRESSURE FLEXIBLE PIPES

2.1 Introduction

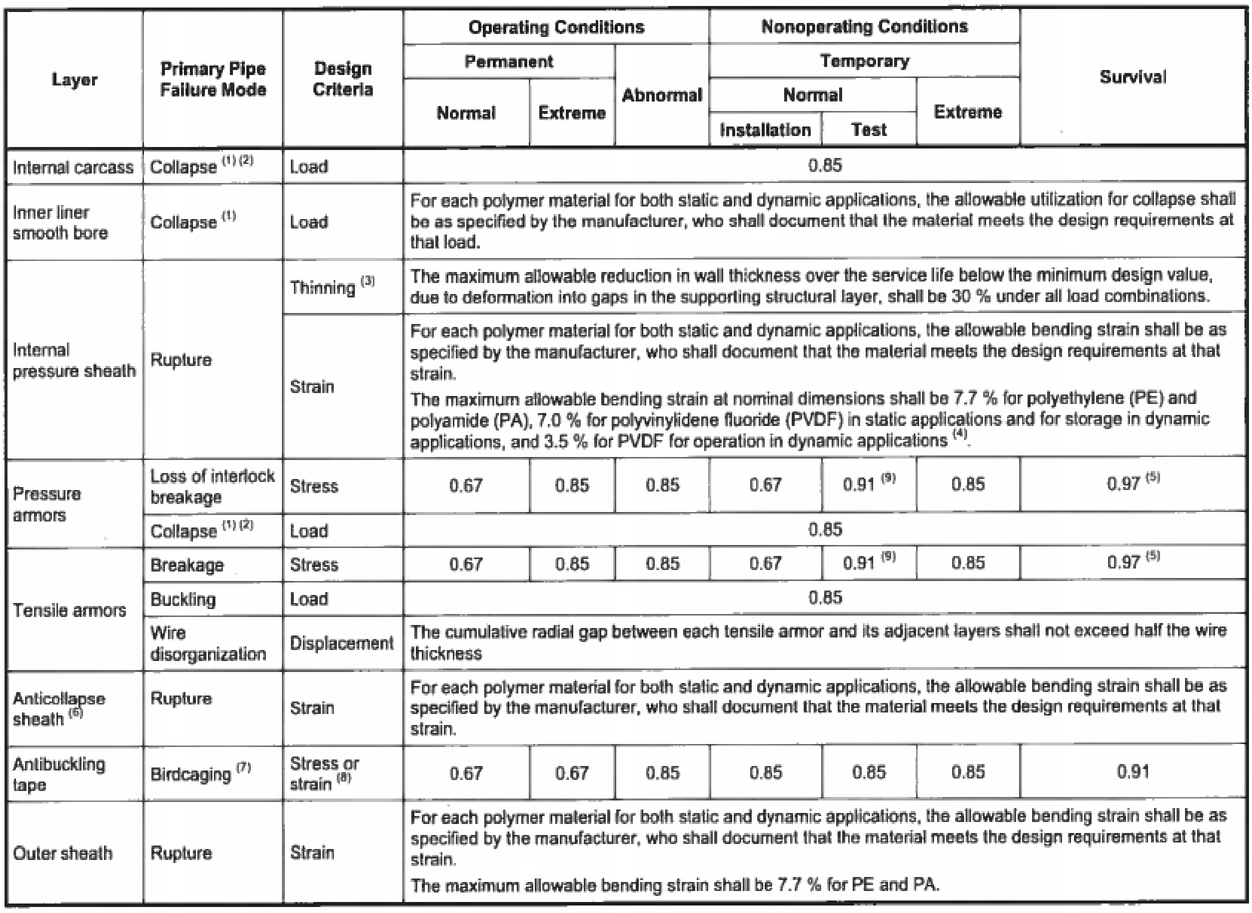
Basic design considerations for the flexible pipe engineer are introduced in the following chapter. Moreover, governing rules and main failure modes are also presented. Special emphasis is made on high pressure design criteria and failure modes.

2.2 Design criteria

The design criteria for unbonded flexible pipe originate from a diverse range of sources including codes for metallic pipes (for armour wires), small-scale and full-scale tests carried out during the early days of the use of the technology for offshore applications.

Table 2.1 below summarizes the carcass and the armour wire stress utilization criteria being proposed by the API 17J [2].

Table 2.1 - Flexible pipe layer design criteria [2]



Layer	Primary Pipe Failure Mode	Design Criteria	Operating Conditions			Nonoperating Conditions			Survival
			Permanent		Abnormal	Temporary			
			Normal	Extreme		Normal		Extreme	
						Installation	Test		
Internal carcass	Collapse (1) (2)	Load	0.85						
Inner liner smooth bore	Collapse (1)	Load	For each polymer material for both static and dynamic applications, the allowable utilization for collapse shall be as specified by the manufacturer, who shall document that the material meets the design requirements at that load.						
Internal pressure sheath	Rupture	Thinning (3)	The maximum allowable reduction in wall thickness over the service life below the minimum design value, due to deformation into gaps in the supporting structural layer, shall be 30 % under all load combinations.						
		Strain	For each polymer material for both static and dynamic applications, the allowable bending strain shall be as specified by the manufacturer, who shall document that the material meets the design requirements at that strain. The maximum allowable bending strain at nominal dimensions shall be 7.7 % for polyethylene (PE) and polyamide (PA), 7.0 % for polyvinylidene fluoride (PVDF) in static applications and for storage in dynamic applications, and 3.5 % for PVDF for operation in dynamic applications (4).						
Pressure armors	Loss of interlock breakage	Stress	0.67	0.85	0.85	0.67	0.91 (9)	0.85	0.97 (5)
	Collapse (1) (2)	Load	0.85						
Tensile armors	Breakage	Stress	0.67	0.85	0.85	0.67	0.91 (9)	0.85	0.97 (5)
	Buckling	Load	0.85						
	Wire disorganization	Displacement	The cumulative radial gap between each tensile armor and its adjacent layers shall not exceed half the wire thickness						
Anticollapse sheath (6)	Rupture	Strain	For each polymer material for both static and dynamic applications, the allowable bending strain shall be as specified by the manufacturer, who shall document that the material meets the design requirements at that strain.						
Antibuckling tape	Birdcaging (7)	Stress or strain (8)	0.67	0.67	0.85	0.85	0.85	0.85	0.91
Outer sheath	Rupture	Strain	For each polymer material for both static and dynamic applications, the allowable bending strain shall be as specified by the manufacturer, who shall document that the material meets the design requirements at that strain. The maximum allowable bending strain shall be 7.7 % for PE and PA.						

The design process of the flexibles can be summarized in following:

1. Identify the limit state (failure mode) most relevant to a flexible pipe application, e.g., burst, collapse, etc.;
2. For a selected limit state determine the loading conditions to be applied in an analysis tool or prediction model;
3. For a loading regime the uncertainties are identified and characterized statistically. These uncertainties will typically include:
4. Material properties and loading;
5. Geometry;
6. Modelling system effects.
7. Estimate the allowable utilization for a target probability of failure.

For a given limit state existing standards use either a load and resistance factor design (LRFD) or working stress design (WSD) format. The latter incorporates uncertainties in loads, analysis methods and material strength into a single safety factor. In the context of flexible pipes the WSD approach is a more practical approach particularly in view of the number of limit states that are likely to be considered.

2.2.1 Standards

The industry standards for unbonded flexible pipe design were developed in their current form through two Joint Industry Projects (JIP), managed by MCS Kenny, from 1994 to 1998. A Specification for flexible pipes was released as API Spec 17J [2], 1st Edition in late 1996 and a major revision to the Recommended Practice for flexible pipes was released as API RP 17B, 2nd Edition, in June 1998 [3]. API Spec 17J and RP 17B replaced many company Specifications that were used up to that time.

In addition to company Specifications a wide range of JIPs and cross-industry initiatives related to flexible pipe technology are on-going or have been recently completed, such as the Flexible Pipe Ancillary Equipment JIP (MCS), Real Life JIP (MCS), Corrosion Fatigue JIP (Marintek) and so on [29].

2.2.2 Failure modes

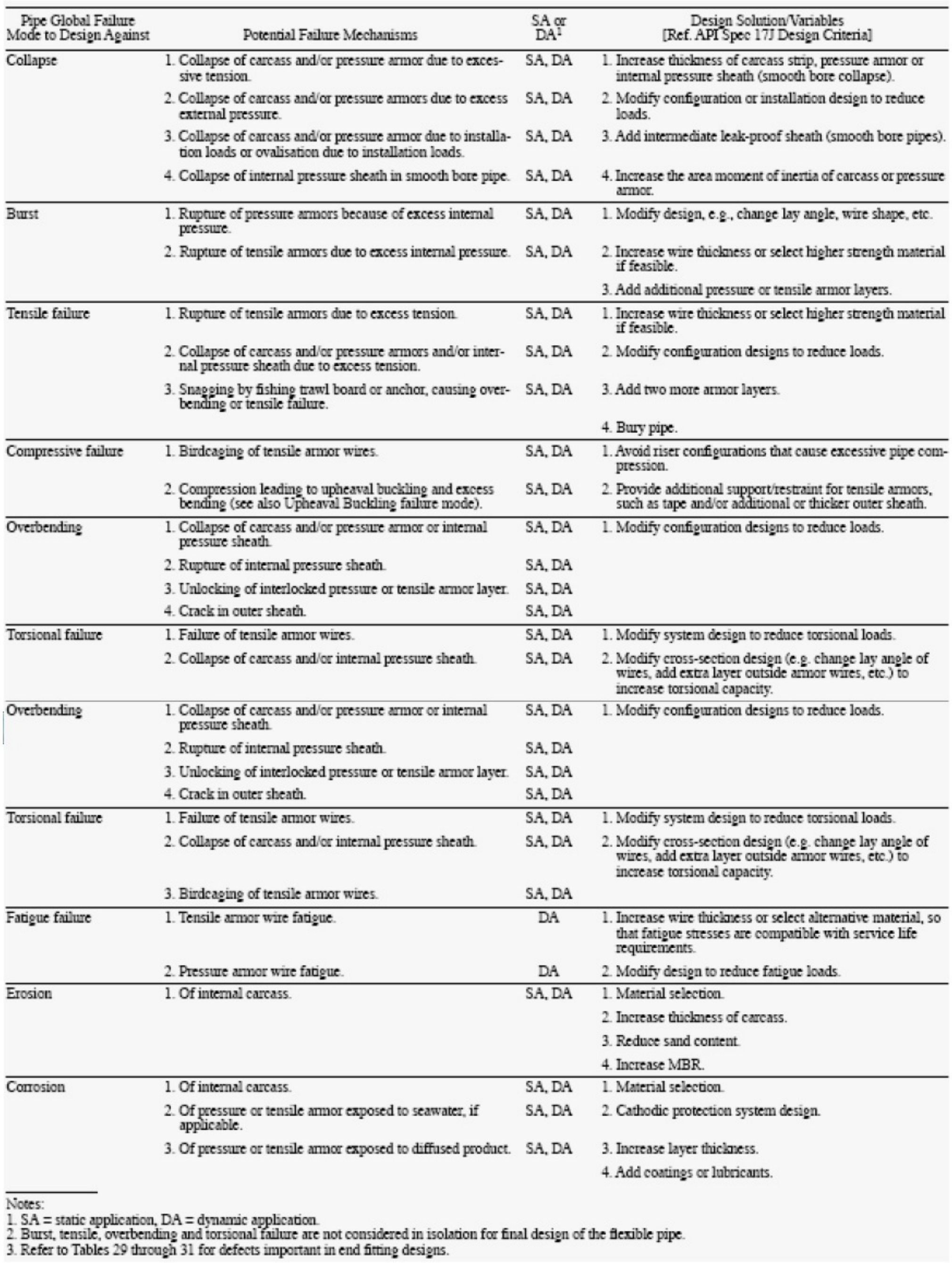
Following is a list of the most prominent influencers on the performance of the flexible pipe and its failure:

1. Temperature, principally that of the bore fluids, but also the external temperature;
2. Pressure of the bore fluids, and the hydrostatic pressure, externally and within the annulus;
3. Extreme/survival loading, influenced by e.g. vessel motions, metocean conditions, marine growth, soil conditions;
4. Variable loading: fatigue;

5. Product fluid composition and the partial pressures of harmful constituents;
6. Corrosion;
7. Erosion of the inside wall by impacting sand;
8. Pipe blockage or flow restriction by hydrates or wax;
9. Accidental damage, such as impact damage or chafing [30].

The number of potential failure modes for a multilayer structure such as a flexible pipe is high. However, the number of different failure modes experienced in operation is more limited. API 17B RP Table 2.2 lists and describes all of the most probable failure modes for a flexible pipe [27].

Table 2.2 - Failure modes for unbonded flexible pipes [27]

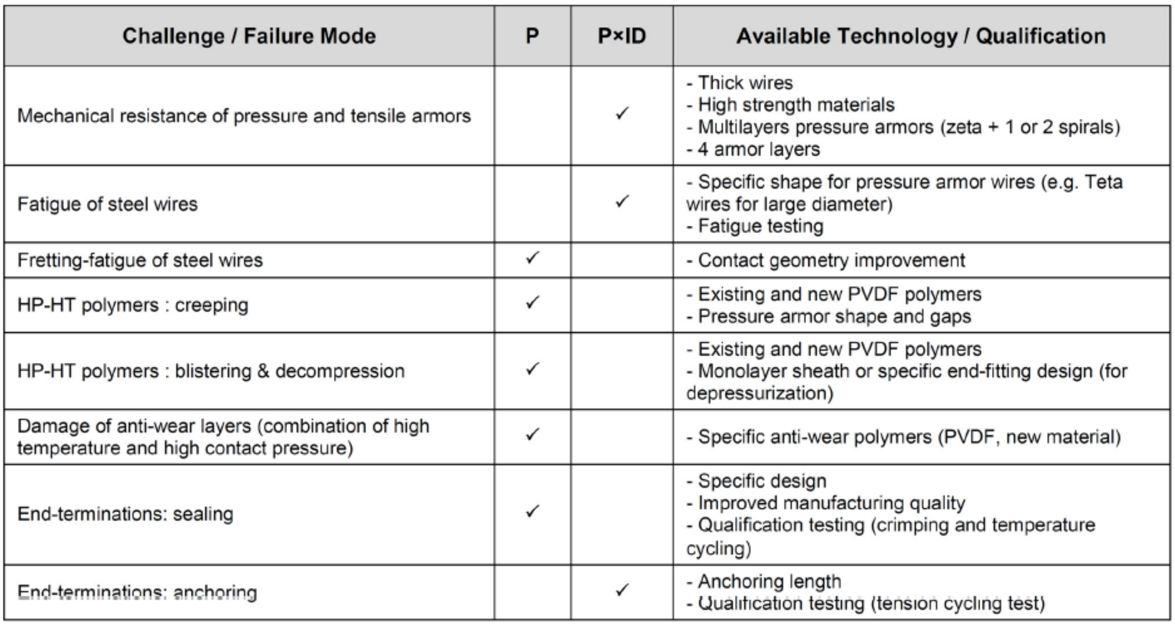


Pipe Global Failure Mode to Design Against	Potential Failure Mechanisms	SA or DA[1]	Design Solution/Variables [Ref. API Spec 17J Design Criteria]
Collapse	1. Collapse of carcass and/or pressure armor due to excessive tension.	SA, DA	1. Increase thickness of carcass strip, pressure armor or internal pressure sheath (smooth bore collapse).
	2. Collapse of carcass and/or pressure armors due to excess external pressure.	SA, DA	2. Modify configuration or installation design to reduce loads.
	3. Collapse of carcass and/or pressure armor due to installation loads or ovalisation due to installation loads.	SA, DA	3. Add intermediate leak-proof sheath (smooth bore pipes).
	4. Collapse of internal pressure sheath in smooth bore pipe.	SA, DA	4. Increase the area moment of inertia of carcass or pressure armor.
Burst	1. Rupture of pressure armors because of excess internal pressure.	SA, DA	1. Modify design, e.g., change lay angle, wire shape, etc.
	2. Rupture of tensile armors due to excess internal pressure.	SA, DA	2. Increase wire thickness or select higher strength material if feasible.
			3. Add additional pressure or tensile armor layers.
Tensile failure	1. Rupture of tensile armors due to excess tension.	SA, DA	1. Increase wire thickness or select higher strength material if feasible.
	2. Collapse of carcass and/or pressure armors and/or internal pressure sheath due to excess tension.	SA, DA	2. Modify configuration designs to reduce loads.
	3. Snagging by fishing trawl board or anchor, causing overbending or tensile failure.	SA, DA	3. Add two more armor layers.
			4. Bury pipe.
Compressive failure	1. Birdcaging of tensile armor wires.	SA, DA	1. Avoid riser configurations that cause excessive pipe compression.
	2. Compression leading to upheaval buckling and excess bending (see also Upheaval Buckling failure mode).	SA, DA	2. Provide additional support/restraint for tensile armors, such as tape and/or additional or thicker outer sheath.
Overbending	1. Collapse of carcass and/or pressure armor or internal pressure sheath.	SA, DA	1. Modify configuration designs to reduce loads.
	2. Rupture of internal pressure sheath.	SA, DA	
	3. Unlocking of interlocked pressure or tensile armor layer.	SA, DA	
	4. Crack in outer sheath.	SA, DA	
Torsional failure	1. Failure of tensile armor wires.	SA, DA	1. Modify system design to reduce torsional loads.
	2. Collapse of carcass and/or internal pressure sheath.	SA, DA	2. Modify cross-section design (e.g. change lay angle of wires, add extra layer outside armor wires, etc.) to increase torsional capacity.
Overbending	1. Collapse of carcass and/or pressure armor or internal pressure sheath.	SA, DA	1. Modify configuration designs to reduce loads.
	2. Rupture of internal pressure sheath.	SA, DA	
	3. Unlocking of interlocked pressure or tensile armor layer.	SA, DA	
	4. Crack in outer sheath.	SA, DA	
Torsional failure	1. Failure of tensile armor wires.	SA, DA	1. Modify system design to reduce torsional loads.
	2. Collapse of carcass and/or internal pressure sheath.	SA, DA	2. Modify cross-section design (e.g. change lay angle of wires, add extra layer outside armor wires, etc.) to increase torsional capacity.
	3. Birdcaging of tensile armor wires.	SA, DA	
Fatigue failure	1. Tensile armor wire fatigue.	DA	1. Increase wire thickness or select alternative material, so that fatigue stresses are compatible with service life requirements.
	2. Pressure armor wire fatigue.	DA	2. Modify design to reduce fatigue loads.
Erosion	1. Of internal carcass.	SA, DA	1. Material selection.
			2. Increase thickness of carcass.
			3. Reduce sand content.
			4. Increase MBR.
Corrosion	1. Of internal carcass.	SA, DA	1. Material selection.
	2. Of pressure or tensile armor exposed to seawater, if applicable.	SA, DA	2. Cathodic protection system design.
	3. Of pressure or tensile armor exposed to diffused product.	SA, DA	3. Increase layer thickness.
			4. Add coatings or lubricants.

Notes:
1. SA = static application, DA = dynamic application.
2. Burst, tensile, overbending and torsional failure are not considered in isolation for final design of the flexible pipe.
3. Refer to Tables 29 through 31 for defects important in end fitting designs.

High Pressure failure modes are related to absolute Pressure P (psi) or Pressure times Internal Diameter (in psi.inch). Available technologies or required qualification tests are given for each failure mode, see Table 2.3 [10].

Table 2.3 - Failure modes for high pressure flexible pipes [10]



Challenge / Failure Mode	P	P×ID	Available Technology / Qualification
Mechanical resistance of pressure and tensile armors		✓	- Thick wires - High strength materials - Multilayers pressure armors (zeta + 1 or 2 spirals) - 4 armor layers
Fatigue of steel wires		✓	- Specific shape for pressure armor wires (e.g. Teta wires for large diameter) - Fatigue testing
Fretting-fatigue of steel wires	✓		- Contact geometry improvement
HP-HT polymers : creeping	✓		- Existing and new PVDF polymers - Pressure armor shape and gaps
HP-HT polymers : blistering & decompression	✓		- Existing and new PVDF polymers - Monolayer sheath or specific end-fitting design (for depressurization)
Damage of anti-wear layers (combination of high temperature and high contact pressure)	✓		- Specific anti-wear polymers (PVDF, new material)
End-terminations: sealing	✓		- Specific design - Improved manufacturing quality - Qualification testing (crimping and temperature cycling)
End-terminations: anchoring		✓	- Anchoring length - Qualification testing (tension cycling test)

2.3 Design of deep water riser systems

For dynamic risers systems used between a subsea facility and a floating platform subject to wind, wave and currents action, flexible pipeline is often the most attractive solution due to its ability to be installed in a compliant configuration enabling to accommodate large displacements. The combination of a compliant configuration and an unbonded construction results in a good fatigue performance of the flexible pipes.

Traditionally, for very harsh environments such as the North-Sea characterized by very high wave height compared to the water depth, a compliant riser system is used (e.g. flexible pipe in Lazy-S or Pliant Wave configuration). In such case, most of the time a bend-stiffener enables to control the riser curvature at the junction point with the floating platform; this is the most dynamic part which is critical with respect to fatigue.

In that case, the curvature variations are large and the top tensions are limited; the layer driving the fatigue life is often the pressure-vault.

With the recent development of deep and ultra-deep water offshore oil fields (beyond 1500 m water depth) in environments milder than the North-Sea such as West of Africa or Brazil, free-hanging configurations can be used. This is indeed a simpler and cheaper riser system requiring the least number of ancillary equipment (no arch, no buoyancy, etc.).

However, such applications induce very high tensile loads in the riser, especially very high top tensions, up to several hundred tons. Reinforced vault layers cannot be avoided to resist the high hydrostatic pressure, even if the use of high strength duplex stainless steels for internal carcass helps to avoid too much weight increase. Then the tensile armour layers tend to become a major driving parameter in the design and in the fatigue life evaluation.

With ultra-deep water, the difference of loads applying on the part of the riser laid on the seabed, subjected to very high hydrostatic pressure, and on the part of the riser connected to the floating platform, subjected to very high tensions, justify the optimization of two different flexible pipe constructions, see Figure 2.1.

One section is made of a pipe construction optimized to resist high tension and fatigue; the construction of the other section being optimized to resist high external hydrostatic pressures. The use of high strength wires is of benefit for both sections: the bottom one has to resist high compressive loads; the top one high tensile loading. Different lay angles of the tensile armours can be done between both sections.

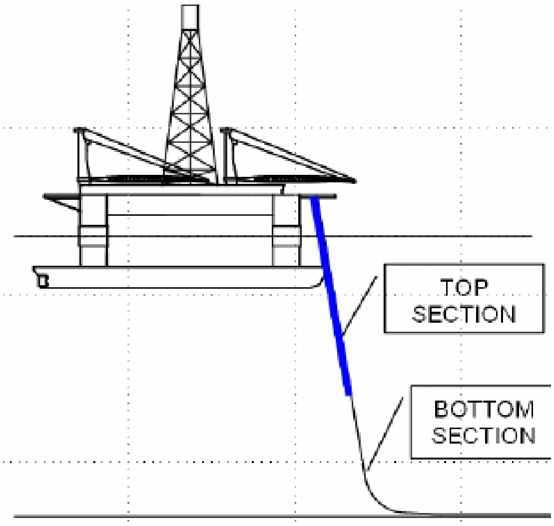


Figure 2.1 - Deep water riser made in two sections and in free-hanging configuration [11]

The tensile armours being helically wound onto the pipe core, one cannot increase the armour thickness above a few millimeters.

Therefore when the tensile loads are too elevated, the design goes from two to four armour layers (an even number of layers is used for torque balance).

The most obvious benefit of high strength steels is for the top section: high strength steel wires enable to push the limits of pipe constructions with 2 armour layers (usually +/-35deg with respect to the pipe axis) before having to go for a 4 armour layers.

Going for 4 armours wires has a huge impact on pipe weight and complexity, and also on installation loads and therefore on cost.

It also has an impact on the size on ancillary equipment for example the bend-stiffener which is a critical component of a riser system.

Extending the utilization envelope of 2 armour layers is of great importance for offshore field developments enabling larger diameter pipes or higher pressures or deeper water (or a combination of the three!) [11].

2.4 Burst pressure

Burst pressure parameter has been chosen as the main indicator of the flexible pipe internal pressure resistance. Therefore, the pressure capacities are presented via bursting failure of the flexibles. Pressure vault and tensile armours ensure the Lion's share of pressure capacities of the pipe, hence the burst pressure is primarily governed by selection of these two components.

API 17 J definition:

The Burst pressure is the pressure at which loss of fluid containment in the pipe occurs due to pipe or end fitting failure.

The bursting pressure is calculated without any other loading than internal pressure i.e. pipe free in elongation and free in rotation. This is a minimum guaranteed value.

2.5 Behavior of flexible pipes under axisymmetrical loads

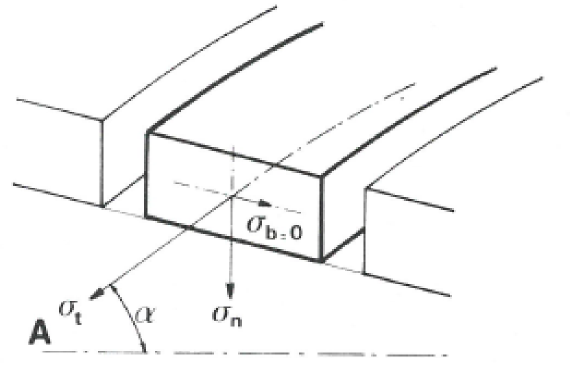
The most usual type of axisymmetrical load (in-service loading) is composed of:

1. an axial force F;
2. an axial moment M;
3. an internal pressure P_i;
4. an external pressure P_e.

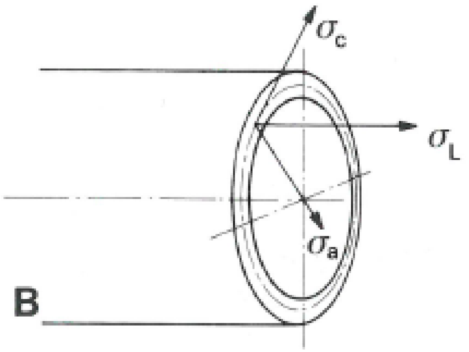
Under such a load, a flexible pipe will undergo the following deformations:

1. a change in length ΔL;
2. a change in radius Δa_i, which might be slightly different for each layer;
3. an axial rotation $\Delta\theta$.

Under load, the tendors of each layer will be submitted to two stresses, Figure 2.2:



For the tendors



For the sheaths

Figure 2.2 - Definitions of stresses [19]

1. an axial stress (tangential) $\sigma_{T(t)}$;
2. a radial stress σ_n.

There is no stress in the third direction (transverse stresses σ_B = 0) as there is some lateral gap between tendors; but this degree of freedom permits a change of the laying angle $\propto$.

The sheaths are submitted to stresses in the three principal directions:

1. an axial stress (longitudinal) σ_L;
2. a radial stress σ_a;
3. a circumferential stress σ_c.

2.6 Flexible pipe pressure capacity

A simplified approach consists in making the following assumptions:

The geometrical deformations are small;

The participation of the plastic sheaths to the resistance of the pipe is negligible;

The plastic sheaths transmit pressure;

The layers remain in contact.

It consists in solving the system of equations obtained in writing:

The three equations of equilibrium between stresses and:

axial forces

$$\sum_{i=1}^{N} n_i \sigma_i A_i \cos\alpha_i = F_0 + \pi P_{\text{int}}^2 a_{\text{int}}^2 - \pi P_{ext}^2 a_{ext}^2 = F; \qquad \text{Eq. 2.1}$$

radial forces

$$\sum_{i=1}^{N} \frac{n_i \sigma_i A_i \sin\alpha_i \tan\alpha_i}{2\pi a_i} = P_{\text{int}} a_{\text{int}} - P_{ext} a_{ext} = \Delta(Pa); \qquad \text{Eq. 2.2}$$

moment

$$\sum_{i=1}^{N} n_i \sigma_i A_i \sin\alpha_i a_i = M; \qquad \text{Eq. 2.3}$$

Where N- number of resistant layers;

n_i- number of tendors of the layer i;

A_i- area of section of the tendor;

$\propto_i$- laying angle, measured from the axis of the pipe, see Figure 2.3;

a_{int} and a_{ext}- radii on which the internal and external pressures P_{int} and P_{ext} apply;

F- axial force including the applied tension F_0 and the end effect due to pressure.

Figure 2.3 - Armouring angle

2.7 Approximate formulas

For quick evaluations, such a system may still be simplified by considering the pressure layers as wound at 90-deg angle and by neglecting the difference of radii between the various layers. Simple formulas are then obtained for tangential stresses in both layers:

$$\sigma_t = \left(\Delta P + \frac{F_0}{\pi a^2}\right)\frac{a}{2e_t \cos\alpha}; \qquad \text{Eq. 2.4}$$

$$\sigma_p = \frac{a\Delta P}{e_p}\left(1 - \frac{\tan^2\alpha}{2}\right) - \frac{F}{\pi a e_p}\frac{\tan^2\alpha}{2}; \qquad \text{Eq. 2.5}$$

Where a – radius;

F_0- applied force;

$\Delta P = P_{int} - P_{ext}$;

$\propto$ - laying angle of the tension layer;

σ – stress in the tendors.

From the approximate equations one can notice, that applied tensions increase stresses in tension tendors subsequently decreasing stresses in pressure tendors.

These equations can be simplified for the pipes with no axial forces and tensions under 0 psi of external pressure:

For stresses in tension layer (tensile armour wires):

$$\sigma_t = \frac{P_{\text{int}} D}{4 e_t \cos \alpha}; \qquad \text{Eq. 2.6}$$

D- diameter of the tension layer.

For stresses in pressure layer (vault) [19]:

$$\sigma_p = \frac{P_{\text{int}} D}{2 e_p} \left(1 - \frac{\tan^2 \alpha}{2} \right); \qquad \text{Eq. 2.7}$$

D- diameter of the pressure layer.

Let’s now appraise the effect of the laying angle (e.g. 30°/45°/55° structures) on the pressure capacities of the flexible pipe. Both $\frac{PD}{4e_t}$ and $\frac{PD}{2e_p}$ members are assumed to be equal to 1 to visualize the results, see Table 2.4.

Table 2.4 - Armouring angle sensitivity on the pressure capacities

α, °	30	45	55
cos(α)	0,87	0,71	0,57
$\tan^2(\alpha)$	0,33	1,00	2,00
σ_t, MPa	1,15	1,41	1,74
σ_p, MPa	0,84	0,50	0,00

From the calculations one can conclude the following:

With increase of the laying angle higher stresses are seen in the tensile tendors (30° - σ_t = 1.15 MPa, 55° - σ_t = 1.74 MPa);

Opposite phenomenon occurs in pressure tendors (30° - σ_p = 0.84 MPa, 55° - σ_p = 0 MPa).

By increasing the laying angle we decrease the loading of the pressure vault, therefore the burst appears on armour wires;

Higher tensions are achievable with lower armouring angles [19].

2.8 Software and prediction models

The following main Technip software packages are used for the pipe design, see Table 2.5.

Table 2.5 - Software and its functions

SOFTWARE NAME	FUNCTION
Structure	Build the structure and perform main computations
EFLEX®	Flexible pipe stress calculation
MOLDI™	Diffusion rate through plastic sheaths calculations
Collapse	Calculation of the collapse pressure of the line
HYPO	Polymer ageing calculation
Therm	Thermal calculations
SLPM	Service life fatigue calculation of the pressure vault
Life	Service life fatigue calculation of the tensile armours

2.9 Overview of the chapter

These are the main subjects discussed in the Chapter 2:

1. Overview of different riser configurations is given;
2. Main design criteria and failure modes of the flexibles are shown;
3. High pressure design criteria are stipulated (basics for the design);
4. Set of simplified equations for pressure vault and armour wires pressure capacities is presented and discussed;
5. Flexible pipe engineering software is briefly introduced.

CHAPTER 3 PRESSURE CAPACITIES

3.1 Introduction

Structural capacities described in this chapter refer to specific applications of the flexible pipe, such as: sweet/sour services, dynamic (flexible risers & top side jumpers)/static pipes (flexible flowlines & subsea jumpers). All the products' applications are therefore accounted for and consequently displayed in terms of internal/design pressures and diameters of the flexibles.

An application is said to be "sweet" service when there is no H_2S (0%).

When there is H_2S, the application is considered "sour" and consequently wires for sour service must be used.

The ultimate goal of the research is to maximize the internal pressure capacities for all components of the flexible pipe (riser, jumper) by selecting optimum material grade/size combination (maximizing strength and dimensions). Henceforth the burst pressure capacity is selected as the overriding design criterion and also as the main failure mode.

In current study of high hoop-stress resistant pipes with two flat steel spiral layers, the potential fretting fatigue phenomenon between the two spiral implies additional restriction in the spiral size selection, limiting the pressure capacities. The effect of the fretting fatigue will be discussed in the following sections.

Main objectives for the study are presented below:

To find maximum burst pressure capabilities for flexible pipes with current technologies available (dynamic/static, sweet/sour, for 25°/35°/45° armouring angle structures (to access the maximum tension/internal pressure balance of the tensile armours performance));

Armouring angle definition (for a single armour wire) is presented below (left-handed helix wire), Figure 3.1.

Figure 3.1 - Laying angle

To build a global picture of the maximum design pressures achievable with the defined scope of structures (prepare Max Design Pressure vs. ID (Internal Diameter) charts);

To make comparisons for dynamic/static structures, for different armouring angles;

To find the consecutive water depths allowable for the structures:

Calculated from collapse pressure for the static applications;

Calculated from tensions (governed by the maximum UF (usage factor in armour layer)) for the dynamic applications.

The following assumptions were set prior to the design research and the consequent simulations:

1. Consider Le Trait manufacturing unit (LTMU) feasibilities (applies limitations onto wire sizes). LTMU is the most advanced Technip's manufacturing facility;
2. Rough bore structure is considered for both static and dynamic application (to account for the severity of the transported fluid and the external pressure). Smooth bore structures are used for stabilized crude oil and for water injection lines [31];
3. [Z-spiral + 2 flat spirals + 2 pairs of armour layers] vault is considered as an optimum choice in order to maximize the pressure capacities;
4. No limitations for the use of materials (to define the best capabilities);
5. Carcass dimensions are at its minimum manufacturing feasibilities (to decrease the weight of the steel and the diameter-associated hoop stresses);
6. The following conventional Z sizes (thicknesses, mm) are considered in the research: Z8, Z10, Z12;
7. Static structures have no anti-wear tapes and are not subjected to fretting fatigue criteria for the spiral layer;
8. Specific rule for the flat spiral dimensioning (ref. section on the flat spiral);
9. Armouring angles considered in this study are 25°, 35°, 45°.

3.2 Description of the software

Technip's in-house "Structure" software was used to build-up the high pressure resistance pipes.

This software is also used to manage and perform main computations related to flexible pipe.

During a lifetime the flexible pipe is subjected to the following axisymetrical loadings, Figure 3.2.

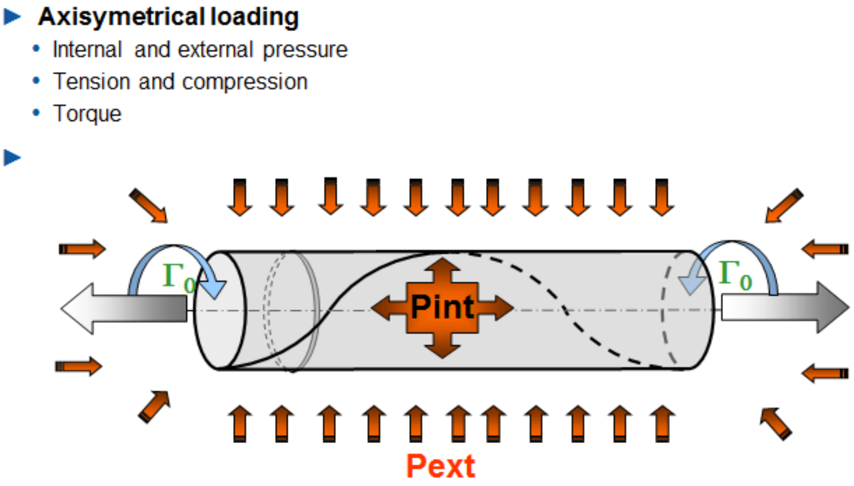


Figure 3.2 - Flexible pipe mechanical behavior [34]

However, in the current study all pipes were subjected to internal/external pressures and torsion only.

Burst pressure calculation

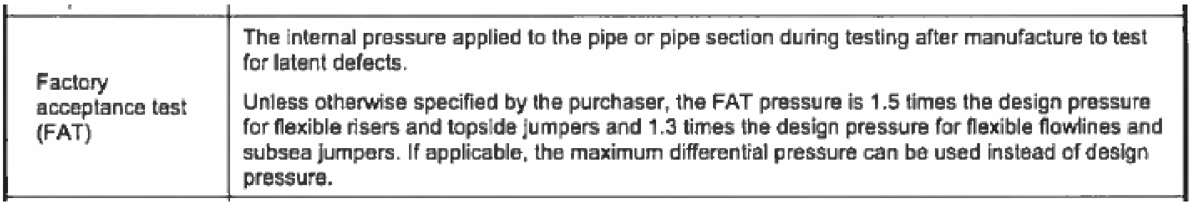
As stated before, burst pressure was selected as the prime indicator of a pipe pressure resistance. EFLEX is a built-un module of the "Structure" software used to calculate the stress in all layers due to the internal pressure.

The outputs of EFLEX software have been compared to numerous full-scale tests have been proven to be reliable. In 1987, Technip had already validated this model with flexible pipe up to API 15,000 psi rating. Since then, each full-scale test has been used to verify the reliability of the predictions.

However, all the design work does not obviate the need for rigorous testing. All new and/or challenging designs are extensively tested through full scale static and dynamic testing. These instrumented tests allow the measurement of the stresses in the external tensile armour layers, along with the deformations, rotations and diameter variation of the flexible pipe for a large number of load cases and boundary conditions. API RP17 B provides guidelines on qualification procedures and tests that are required to qualify flexible pipe components.

3.3 Factory hydrostatic test pressure

Factory hydrostatic test pressure corresponds to the minimum internal pressure that the pipe must sustain during the Factory Acceptance Test (FAT), see Figure 3.3.



Factory acceptance test (FAT)	The internal pressure applied to the pipe or pipe section during testing after manufacture to test for latent defects. Unless otherwise specified by the purchaser, the FAT pressure is 1.5 times the design pressure for flexible risers and topside jumpers and 1.3 times the design pressure for flexible flowlines and subsea jumpers. If applicable, the maximum differential pressure can be used instead of design pressure.

Figure 3.3 – Factory acceptance test [2]

Role of the FAT in the flexible pipe production is undisputed, as it brings a relaxation from the residual stresses induced in the process of manufacturing (during spiralling), Figure 3.4 .

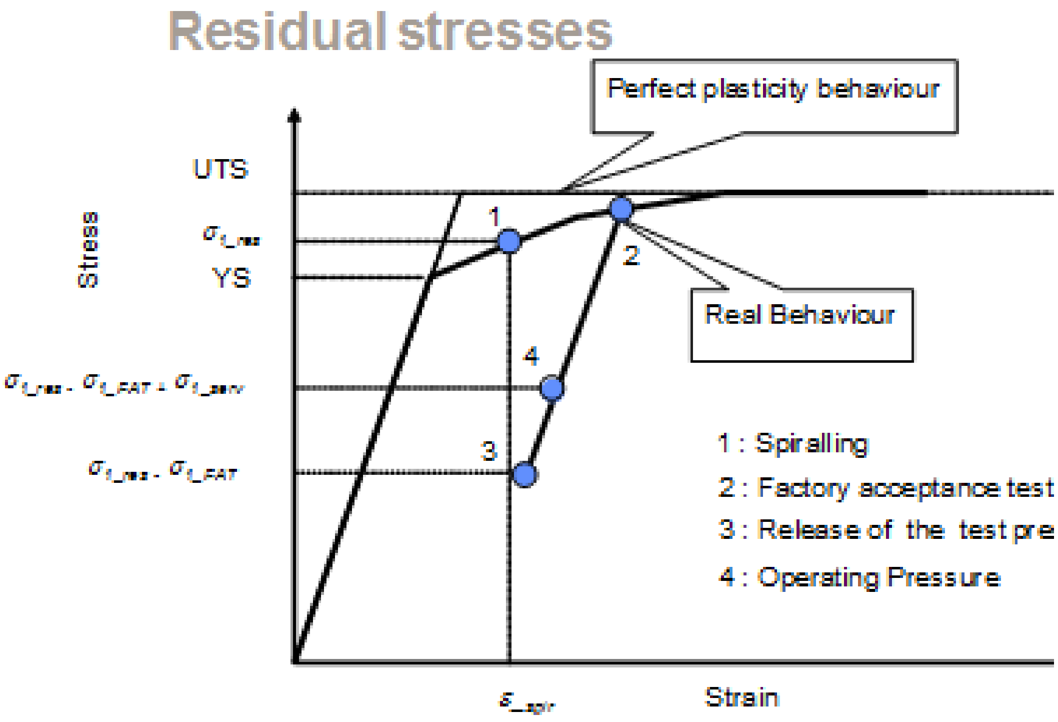


Figure 3.4 - Stresses envelope [32]

The stress distribution in the flexible pipe accounts for the stresses due to spiralling, for the influence of the pressure test and for the stresses generated by static loading of in service conditions (tension and pressure).

Now it is known that residual stresses (due to spiralling) added to stresses due to internal and external pressure have a large influence in dynamic application. Previous study has pointed out that those stresses are in the axis of the wire and are equal to:

$$\sigma_{1-stat} = \sigma_{1-res} - \sigma_{1-FAT} + \sigma_{1-serv}; \qquad \text{Eq. 3.1}$$

Where σ_{1-stat} – the stress resulting from spiraling of the Zeta;

σ_{1-FAT} – the stress relaxation due to the pressure test;

σ_{1-serv} – the stress due to in-service static loading.

This equation is illustrated in Figure 3.4, showing a σ = f (ε) curve with the different steps (manufacturing, FAT, release of the test pressure, in-service conditions).

For each material, a real traction curves of the steel has been entered in Service Life Prediction Model (SLPM) software, defined by an analytical formulation.

Stresses due to application of the internal pressure during FAT or during service are calculated with the mean contact pressures (internal and external) evaluated by EFLEX software with the FAT and in-service loading. The FAT loading must be the nominal FAT pressure.

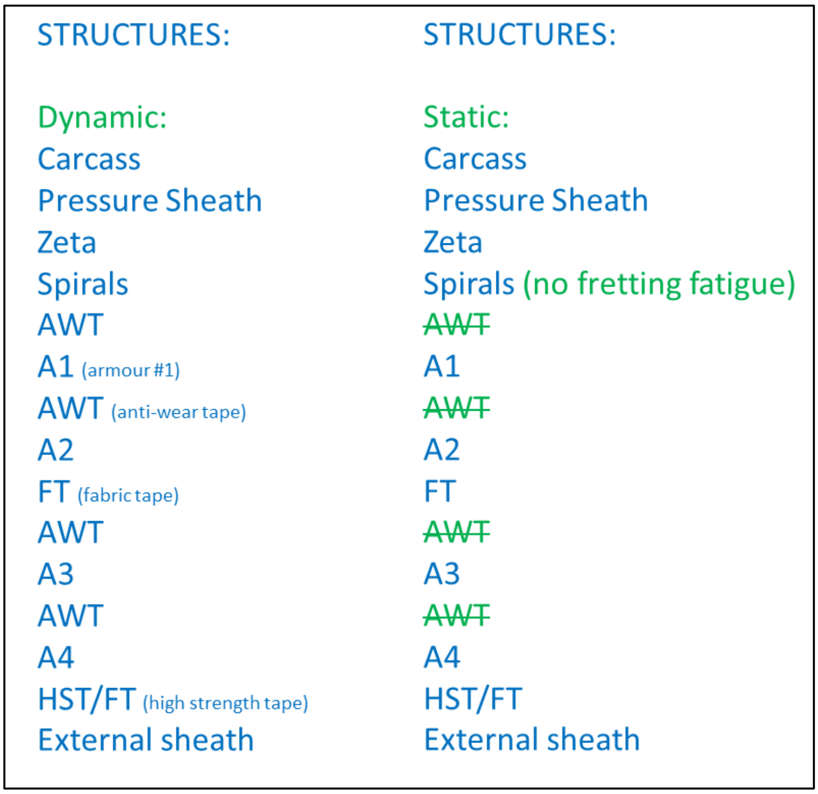
The in-service loading must be the operating internal pressure and the minimum tension for the given wave class.

It clearly appears from Eq.3, that, the higher the FAT pressure is, the lower the static stresses in the axis of the wire are. Increasing the peak pressure during the factory acceptance test has therefore a favorable impact on the global service life.

3.4 Study of limiting structural capacities

Description of the flexible pipe layer selection is presented in the following section. Flexibles are differentiated by the dynamism of the structures, please see Table 3.1.

Table 3.1 - Components of the HPHT flexible pipes from the study



STRUCTURES:	STRUCTURES:
Dynamic:	Static:
Carcass	Carcass
Pressure Sheath	Pressure Sheath
Zeta	Zeta
Spirals	Spirals (no fretting fatigue)
AWT	~~AWT~~
A1 (armour #1)	A1
AWT (anti-wear tape)	~~AWT~~
A2	A2
FT (fabric tape)	FT
AWT	~~AWT~~
A3	A3
AWT	~~AWT~~
A4	A4
HST/FT (high strength tape)	HST/FT
External sheath	External sheath

As seen from the Table 3.1, both options encompass the internal carcass layer (to deal with severe fluids and the external pressure). Zeta layer is considered in this study. Zeta layer is reinforced with two layers of flat spirals (to increase the internal pressure capacities). To prevent wear of metallic layers, specific anti-wear tapes are selected for the dynamic case. High strength tapes applied above the last pair of armour wires are utilized to cope with the reverse end-cap effect phenomenon. Two pairs of armours are being used, see Figure 3.5.

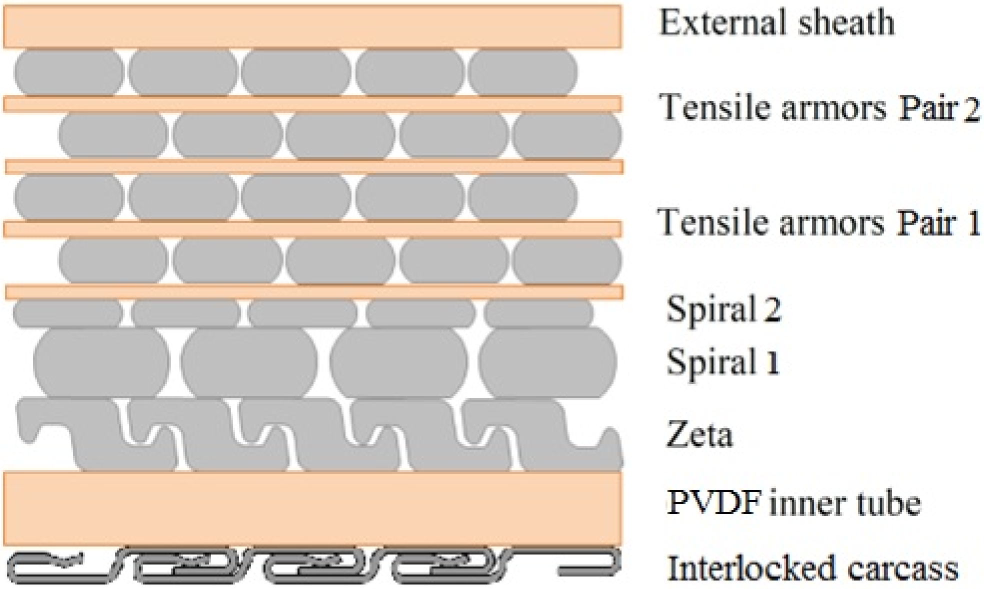


Figure 3.5 - HPHT dynamic structure

Pressure and tensile armour layers structural capacities is defined as the material SMYS or 0.9 x UTS. To conclude with, following Figure 3.6 represents a summary of the pressure limiting capacities of the flexible pipe.

- ▶ **Pressure capacity combination of :**
  - **Pressure vaults**
    - Global thickness
    - Steel strength
  - **Armour layers strength**
    - Wires thickness
    - Steel strength
    - Armouring angle
    - Number of layers
- ▶ **Pressure limitation**
  - **Wires thicknesses**
  - **Polymer creeping**
    - Pressure sheath
    - Anti wear tape
  - **Fatigue**
  - **Manufacturing**
  - …

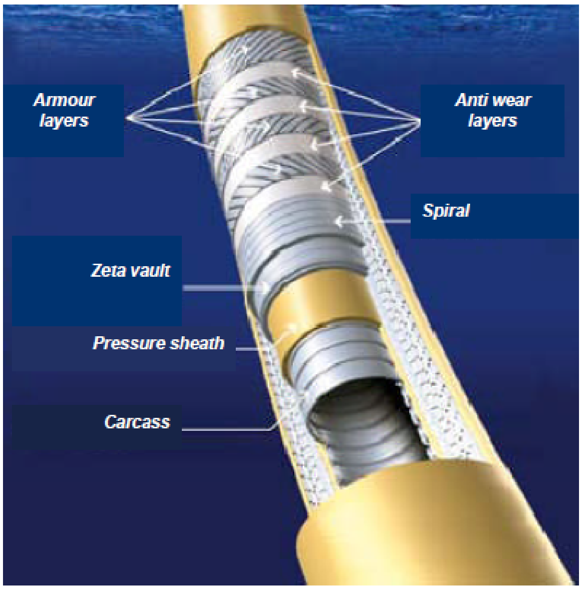


Figure 3.6 - Flexible pipe structure capacity

As seen from the Figure 3.6, pressure capacities of the flexible pipe are mostly driven by the strength of pressure vault and armour wires. To achieve high pressure capacities large amounts of steel are often used, therefore pipes are faced with higher fatigue loadings. The effects of fatigue loadings on pipe pressure capacities will be discussed in CHAPTER 4 .

3.4.1 Carcass

Critical parameters for the carcass selection are: NaCl content [g/L] in the transported fluid and the pH of the fluid. Set of special application envelopes based on the fluid temperature and the partial pressure of H_2S is used for the selection of the appropriate steel grade.

The following grade was selected for both sweet and sour services: Duplex 2205 (used specifically for the carcass applications). Duplex stainless steels are characterized by an austenitic-ferritic microstructure which confers high mechanical properties and good resistance to localized corrosion and stress corrosion cracking. 2205 duplex stainless steel exhibits a good weldability.

No collapse pressure calculations are considered at this stage.

Duplex 2205 carcass has successfully been used for following Technip HPHT projects studied in this paper.

Caesar Tonga/ 4.75” Production Riser/Spar platform hosted/sweet service/1500 m of water depth;

Cascade Chinook/7” Production Jumper/From the floating production storage offloading unit (FPSO) to a free standing hybrid riser (FSHR)/sour service/732 m in length;

Skarv/10” Production riser/FPSO hosted/sour service/370 m of water depth.

3.4.2 Pressure sheath

For most High Pressure applications, operating conditions are associated with High Temperature. Therefore PVDF polymer is often chosen for the pressure sheath material as it can be used up to 150 °C.

Technip has three PVDF polymers that can be used as internal pressure sheath: Gammaflex, Coflon and Coflon XD [10].

Coflon XD has been chosen as the pressure sheath material in current study. This material is currently qualified for the temperature window from -30 to 150 °C and the conventional design pressure domain.

Thickness calculations are performed with the ‘Thickness’ software for the Rilsan P40TL polyamide material (as it yields bigger associated thickness, conservative approach in order to account for the polymer creeping). The following input was considered for the “Thickness” calculation with the Rilsan P40TL: Max. Design Pressure = 15 000 psi, $FAT = 15000 \cdot 1.5 \cdot 1.05 = 23625$ psi, Temperature = 100 °C. However, the temperature limitation for the Coflon XD is 150 °C. Thickness of the pressure sheath is also changing depending on the thickness of the Zeta layer, due to the creeping phenomenon. The software automatically accounts for the polymer creeping.

3.4.3 Zeta layer

With very high $P \times ID$, two pressure armours (zeta or teta + flat steel spiral) have to be used. Uniquely for this study two flat spirals will be used in order to achieve maximum design and operating pressures. With this construction, the potential for fretting fatigue between the two flat spiral layers has to be considered. For more information about fretting fatigue please refer to the section about the use of flat spiral.

Z-layer size applicability is function of the layer internal diameter.

3.4.4 Flat spiral

Pipe pressure capacities in the chosen double spiral construction can potentially be limited by a fretting fatigue phenomenon. It occurs when layers of a flexible experience small repetitive displacements under high contact pressures (dynamic applications). Combined with cyclic fatigue loading, fretting-fatigue can introduce cracking phenomena which reduce the endurance of assemblies. To reduce either the contact pressure or the displacements an engineer shall adjust the pipe structure by changing the wires geometry (reduce the size of second spiral), see Figure 3.7.

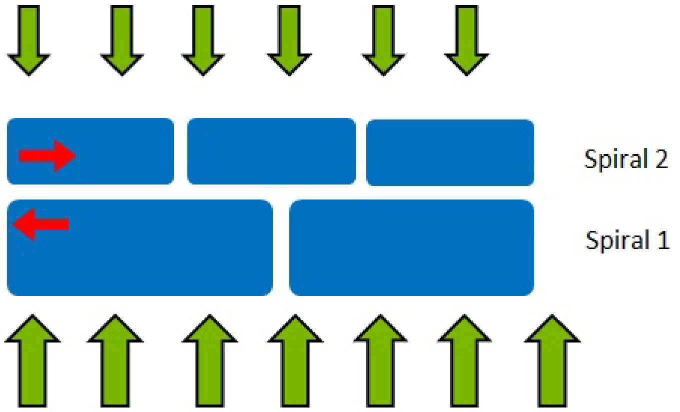


Figure 3.7- Spirals' relative displacements

3.4.5 Anti-wear tape

Anti-wear plastic layers are used for dynamic applications. Their purpose is to suppress wear between metallic layers submitted to relative movements, i.e.:

1. Between the pressure vault and the inner tensile armour layer;
2. Between each armour layer.

For some configurations, it could be economically interesting to mix two anti-wear tape materials within a same flexible pipe (for instance, when the contact pressures vary along the flexible pipe).

PVDF Gammaflex is selected as an anti-wear tape material for all structures (based on the previous experience with high temperature/contact pressures pipes).

3.4.6 Tensile armour layer

This layer is present almost in every flexible pipe. It shall be made up of two sets of wires laid in opposite directions. 25°, 35°, 45° conventional laying angles are considered in this study (25° to 55° are the manufacturing feasibilities of the manufacturing unit).

Two pairs of armour wires unsure higher tensile and pressure capacities.

3.4.7 Fabric tape

Fabric tapes are tapes assemblies, acting only as a manufacturing aid. Fabric tapes purpose can be to hold layers together between manufacturing steps, to prevent scratches during armouring or to prevent lubricant outflow, however they do not have an 'in service' function once manufacture is completed.

When tapes are designed to resist in service to loading such as Reverse End Cap, or Lateral Buckling, they have to be called High Strength Tapes, even if they have the same decomposition as fabric tapes.

3.4.8 External plastic sheath

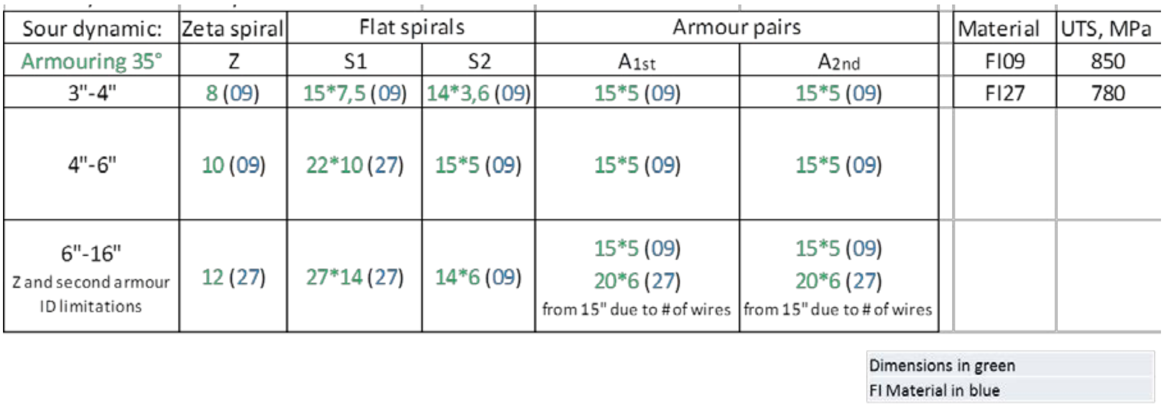
All flexible pipes have an external plastic sheath. Its main purposes are to keep the armours in position, protect them against environmental corrosion and protect the insulation layers against water ingress.

Polyolefin based polymer (Tpe) TPFLEX is used as the external sheath material for all structures. The thickness is assumed uniform for all pipe ID (equal to 15 mm).

3.5 Limiting elements selection

As the result of the material selection study the following structures were created e.g.: for 35° tensile laying angle (see Table 3.2, Table 3.3 and Table 3.4). Material strength ranges from 780 to 850 MPa for sour service and from 780 to 1400 for sweet service due to severity of applications. For 4" ID pipe two possible design options for Z layer are available (8 and 10 mm thick wires). Width and thickness of the second flat spiral is reduced to mitigate risk of fretting fatigue phenomenon for all dynamic applications, e.g.: 5" ID sweet structure combines 10 mm thick and 5 mm thick spirals, see Table 3.9. Due to the manufacturing process of a second pair of armours the pipe inner diameter is limited to 16". For sour dynamic structure the wider armours shall be used for all pipes starting from 15" ID to respect the maximum possible number of wire of armouring machine. Interestingly, burst appears on pressure vault for all 35° sour dynamic structures, therefore smaller tensile wires with higher UTS values perform better in respect to pressure capacities. However, for 35° sour static structures starting from 15" ID burst occurs on the tensile armours, hence the bigger wires are more pressure resistant even with smaller UTS. This phenomenon is noticed for the pipes with high ID and is explained by the end-cap effect forces (please refer to next sections).

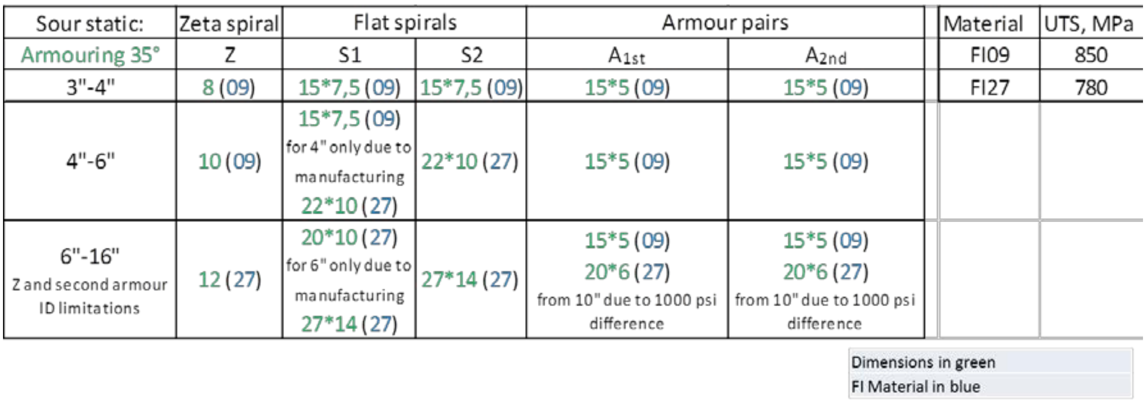
Table 3.2 - Sour dynamic structure



Sour dynamic:	Zeta spiral	Flat spirals		Armour pairs		Material	UTS, MPa
Armouring 35°	Z	S1	S2	A1st	A2nd	FI09	850
3"-4"	8 (09)	15*7,5 (09)	14*3,6 (09)	15*5 (09)	15*5 (09)	FI27	780
4"-6"	10 (09)	22*10 (27)	15*5 (09)	15*5 (09)	15*5 (09)		
6"-16" Z and second armour ID limitations	12 (27)	27*14 (27)	14*6 (09)	15*5 (09) 20*6 (27) from 15" due to # of wires	15*5 (09) 20*6 (27) from 15" due to # of wires		

Dimensions in green
FI Material in blue

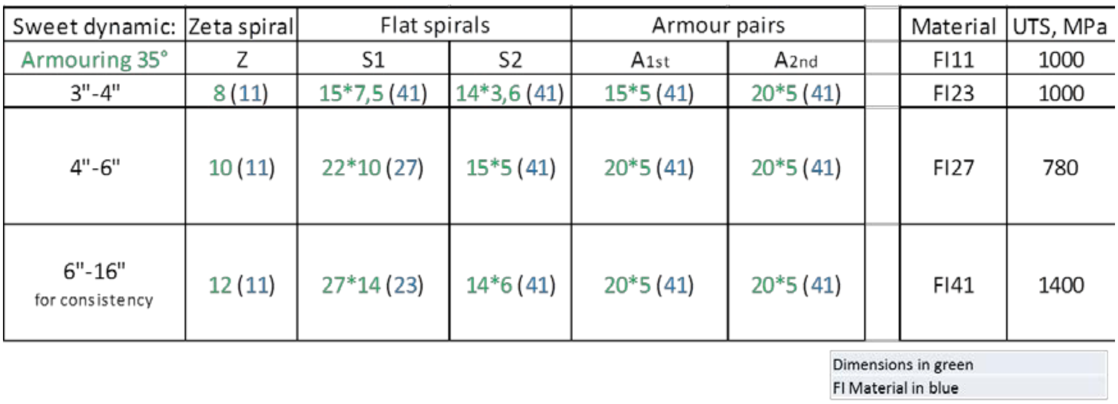
Table 3.3 - Sour static structure



Sour static:	Zeta spiral	Flat spirals		Armour pairs		Material	UTS, MPa
Armouring 35°	Z	S1	S2	A1st	A2nd	FI09	850
3"-4"	8 (09)	15*7,5 (09)	15*7,5 (09)	15*5 (09)	15*5 (09)	FI27	780
4"-6"	10 (09)	15*7,5 (09) for 4" only due to manufacturing 22*10 (27)	22*10 (27)	15*5 (09)	15*5 (09)		
6"-16" Z and second armour ID limitations	12 (27)	20*10 (27) for 6" only due to manufacturing 27*14 (27)	27*14 (27)	15*5 (09) 20*6 (27) from 10" due to 1000 psi difference	15*5 (09) 20*6 (27) from 10" due to 1000 psi difference		

Dimensions in green
FI Material in blue

Table 3.4 - Sweet dynamic structure



Sweet dynamic:	Zeta spiral	Flat spirals		Armour pairs		Material	UTS, MPa
Armouring 35°	Z	S1	S2	A1st	A2nd	FI11	1000
3"-4"	8 (11)	15*7,5 (41)	14*3,6 (41)	15*5 (41)	20*5 (41)	FI23	1000
4"-6"	10 (11)	22*10 (27)	15*5 (41)	20*5 (41)	20*5 (41)	FI27	780
6"-16" for consistency	12 (11)	27*14 (23)	14*6 (41)	20*5 (41)	20*5 (41)	FI41	1400

Dimensions in green
FI Material in blue

Examples of 35° structures are also presented below, see Table 3.5 and Table 3.6.

Table 3.5 - 35° Sour service pipe

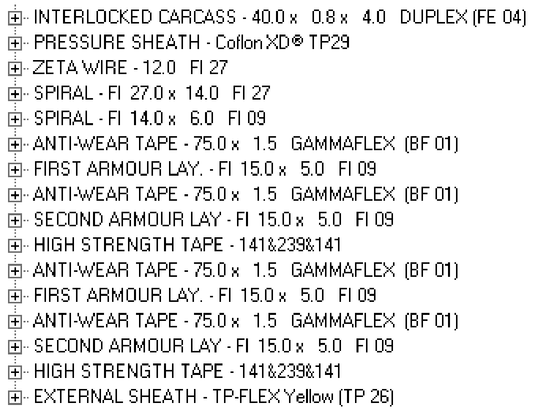
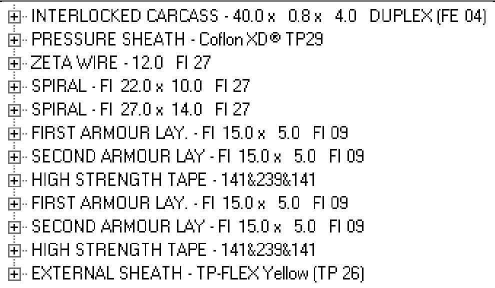
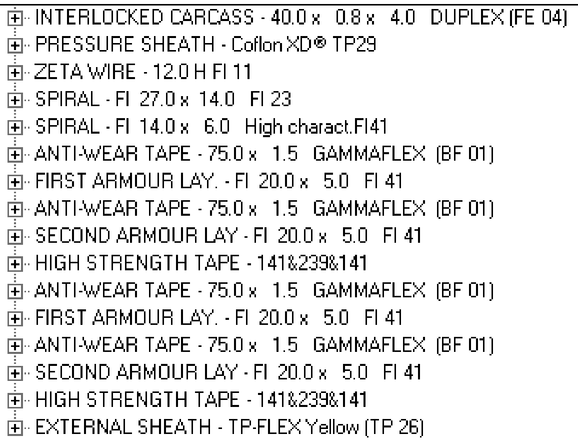
Sour dynamic 6" ID (Z12)	Sour static 6" (Z12)
 INTERLOCKED CARCASS - 40.0 x 0.8 x 4.0 DUPLEX (FE 04) PRESSURE SHEATH - Coflon XD® TP29 ZETA WIRE - 12.0 FI 27 SPIRAL - FI 27.0 x 14.0 FI 27 SPIRAL - FI 14.0 x 6.0 FI 09 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) FIRST ARMOUR LAY. - FI 15.0 x 5.0 FI 09 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) SECOND ARMOUR LAY - FI 15.0 x 5.0 FI 09 HIGH STRENGTH TAPE - 141&239&141 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) FIRST ARMOUR LAY. - FI 15.0 x 5.0 FI 09 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) SECOND ARMOUR LAY - FI 15.0 x 5.0 FI 09 HIGH STRENGTH TAPE - 141&239&141 EXTERNAL SHEATH - TP-FLEX Yellow (TP 26)	 INTERLOCKED CARCASS - 40.0 x 0.8 x 4.0 DUPLEX (FE 04) PRESSURE SHEATH - Coflon XD® TP29 ZETA WIRE - 12.0 FI 27 SPIRAL - FI 22.0 x 10.0 FI 27 SPIRAL - FI 27.0 x 14.0 FI 27 FIRST ARMOUR LAY. - FI 15.0 x 5.0 FI 09 SECOND ARMOUR LAY - FI 15.0 x 5.0 FI 09 HIGH STRENGTH TAPE - 141&239&141 FIRST ARMOUR LAY. - FI 15.0 x 5.0 FI 09 SECOND ARMOUR LAY - FI 15.0 x 5.0 FI 09 HIGH STRENGTH TAPE - 141&239&141 EXTERNAL SHEATH - TP-FLEX Yellow (TP 26)

Table 3.6 - 35° Sweet service pipe

Sweet dynamic 6" ID (Z12)	Sweet static 6" ID (Z12)
 INTERLOCKED CARCASS - 40.0 x 0.8 x 4.0 DUPLEX (FE 04) PRESSURE SHEATH - Coflon XD® TP29 ZETA WIRE - 12.0 H FI 11 SPIRAL - FI 27.0 x 14.0 FI 23 SPIRAL - FI 14.0 x 6.0 High charact.FI41 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) FIRST ARMOUR LAY. - FI 20.0 x 5.0 FI 41 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) SECOND ARMOUR LAY - FI 20.0 x 5.0 FI 41 HIGH STRENGTH TAPE - 141&239&141 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) FIRST ARMOUR LAY. - FI 20.0 x 5.0 FI 41 ANTI-WEAR TAPE - 75.0 x 1.5 GAMMAFLEX (BF 01) SECOND ARMOUR LAY - FI 20.0 x 5.0 FI 41 HIGH STRENGTH TAPE - 141&239&141 EXTERNAL SHEATH - TP-FLEX Yellow (TP 26)	 INTERLOCKED CARCASS - 40.0 x 0.8 x 4.0 DUPLEX (FE 04) PRESSURE SHEATH - Coflon XD® TP29 ZETA WIRE - 12.0 H FI 11 SPIRAL - FI 15.0 x 7.5 High charact.FI41 SPIRAL - FI 15.0 x 7.5 High charact.FI41 FIRST ARMOUR LAY. - FI 20.0 x 5.0 FI 41 SECOND ARMOUR LAY - FI 20.0 x 5.0 FI 41 HIGH STRENGTH TAPE - 141&239&141 FIRST ARMOUR LAY. - FI 20.0 x 5.0 FI 41 SECOND ARMOUR LAY - FI 20.0 x 5.0 FI 41 HIGH STRENGTH TAPE - 141&239&141 EXTERNAL SHEATH - TP-FLEX Yellow (TP 26)

3.6 Maximum Design Pressures Results

Current Technip projects and the maximum design pressures obtained for 35° structures are presented on the charts, see Figure 3.8, Figure 3.9 and Figure 3.10.

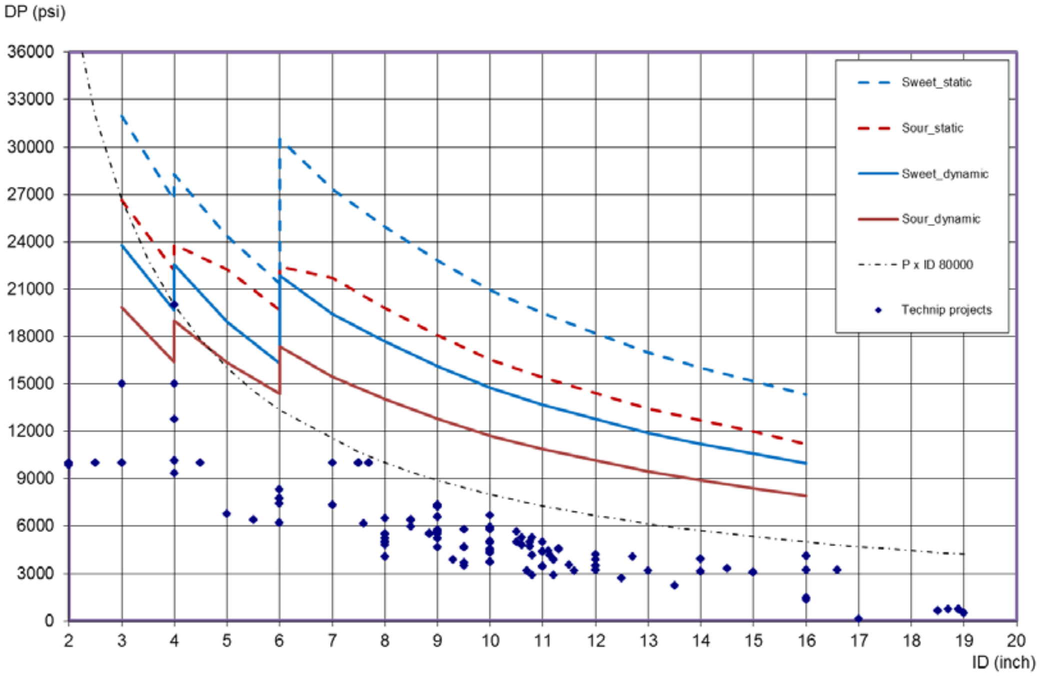


Figure 3.8- Maximum design pressure versus ID all services

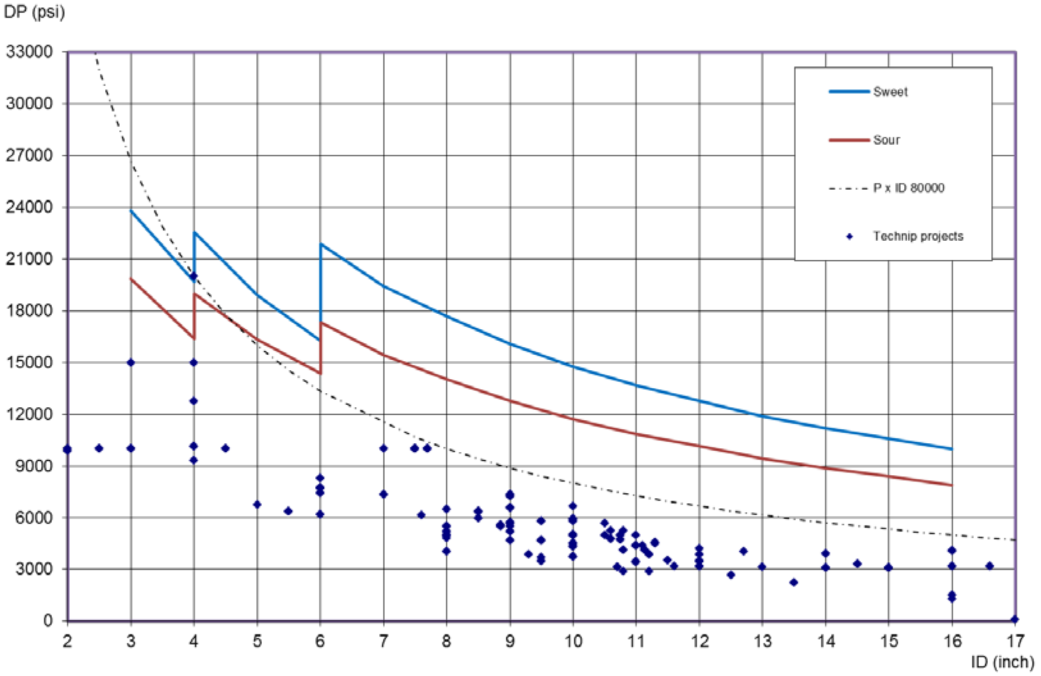


Figure 3.9 - Maximum design pressure versus ID dynamic services

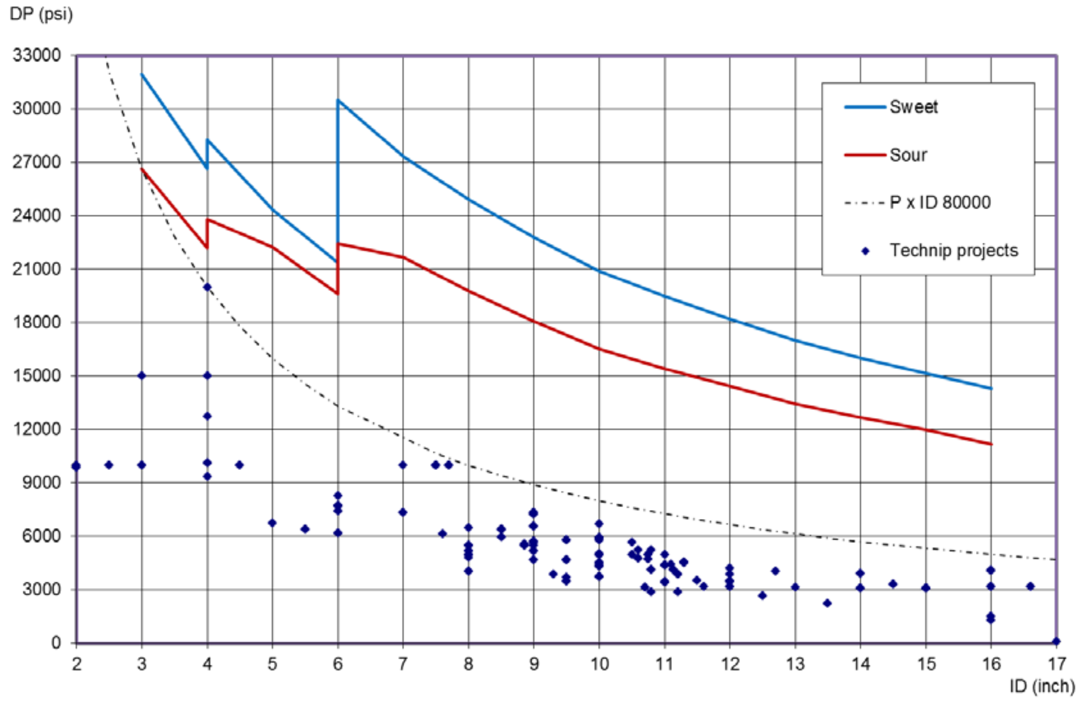


Figure 3.10 - Maximum design pressure versus ID static services

One can notice that 80 000 psi.inch domain was exceeded for all static services. However, it is beyond feasibility limits for dynamic services with 8 mm thick Zeta spiral. Armouring angle greater than 35 degree would then be needed in this case for capability exceeding 80 000 psi.inch.

For all charts the pressure capacities decrease, as pipe diameter increases, creating higher associated hoop stresses. Pressure drops seen for 4” and 6” pipes are explained with difference in Z and spiral thicknesses.

3.6.1 Water depths calculation

It is to be noted that the assessment performed in this section relates to flexible pipe designed with the premises previously presented in order to maximize Design Pressure. Design adjustment with limited impact on pressure capability could significantly increase the allowable water depth for installation. Multi-section section riser would also be a way to increase WD compared to the results presented.

Free-hanging riser configuration was chosen for the maximum allowable water depths (WD) calculations (dynamic applications). It is by far the simplest and most cost effective of all available dynamic configurations. It does not require any intermediate support structures and is consequently the most installation friendly configuration.

There are significant tensions applied onto a structure, which impose water depth limitations for a given internal pressure. These limitations are found via maximum allowable utilization factor (UF) for the armour wires. The armour UF considered for the tensile stress check module is 0.67 (Normal recurrent operation, reference to API 17J [2]).

Tensions assumed to be applied at the top of the riser structure subjected to Maximum Design Pressure. Riser is considered to be filled with oil (constant density = 900 $\frac{kg}{m^3}$).

Associated water depths are obtained with the following formula:

$$WD[m] = \frac{Tension\_[N]}{Weight\_in\_water\_full\_of\_oil\_[N \cdot m] \cdot Catenary\_factor \cdot DAF}; \qquad \text{Eq. 3.1}$$

Catenary factor = 1.1,for simplification purposes;

DAF = *2*. the most conservative.

Catenary factor of 1.1 can be superseded by 1.05 for the deep water applications.

The Dynamic Amplification Factor (DAF) is dependent on the floater type and the prevailing environmental conditions. The following approximations may be used:

Tension Leg Platform (TLP) & SPAR = 1.1;

Semi Submersibles = 1.3 to 1.5;

FPSO or equivalent = 1.3 to 2.0.

The input data for the tension calculations for the sour service (35° structures) are presented in the Table 3.7. Data were obtained from the 'Structure' software.

Table 3.7 - Input data for the tension calculations (sour service)

				TOP riser section					
D, "	D, m	Sour dynamic, psi	Max design pressure Sour dynamic, bar	Pressure internal (bars)	Pressure external (bars)	weight in water empty, kgf/m	volume internal, l/m	weight in water full of oil, kgf/m	weight in water full of oil, N/m
3	0,0762	19841	1368	1368	1	103	5	108	1056
4	0,1016	16403	1131	1131	1	120	9	127	1250
4	0,1016	18985	1309	1309	1	146	9	154	1509
5	0,127	16360	1128	1128	1	164	13	177	1732
6	0,1524	14387	992	992	1	183	19	200	1961
6	0,1524	17346	1196	1196	1	222	19	239	2347
7	0,1778	15446	1065	1065	1	247	26	270	2648
8	0,2032	14039	968	968	1	265	34	295	2899
9	0,2286	12777	881	881	1	290	43	328	3220
10	0,254	11690	806	806	1	314	53	362	3552
11	0,2794	10863	749	749	1	331	64	389	3815

12	0,3048	10152	700	700	1	347	76	415	4073
13	0,3302	9441	651	651	1	372	89	452	4436
14	0,3556	8890	613	613	1	386	103	478	4693
15	0,381	8397	579	579	1	452	118	558	5475
16	0,4064	7904	545	545	1	475	135	596	5845

The results of tension calculations for sour service are presented in the Table 3.8.

Table 3.8 - Output data of the tension calculations (sour service)

D, "	UTS armours (MPa)	Normal operation stress allowed 0,67, MPa	WD, m	Stress observed in Eflex, MPa	Tension, kN
3	850	570	950	565	2208
4	850	570	850	554	2338
4	850	570	700	550	2324
5	850	570	650	554	2476
6	850	570	600	548	2588
6	850	570	450	550	2323
7	850	570	400	549	2331
8	850	570	400	569	2551
9	850	570	350	556	2479
10	850	570	300	547	2345
11	850	570	300	560	2518
12	850	570	300	567	2688
13	850	570	250	549	2440
14	850	570	250	558	2581
15	780	523	350	514	4216
16	780	523	350	521	4501

Results of tension calculations are presented in Figure 3.11 and Figure 3.12. Tensions bring some limitations to vessel and laying equipment capacity used for the installation. The maximum tension calculated is 450 tons for 16” pipe. Current Technip laying equipment capacity is 650 tons.

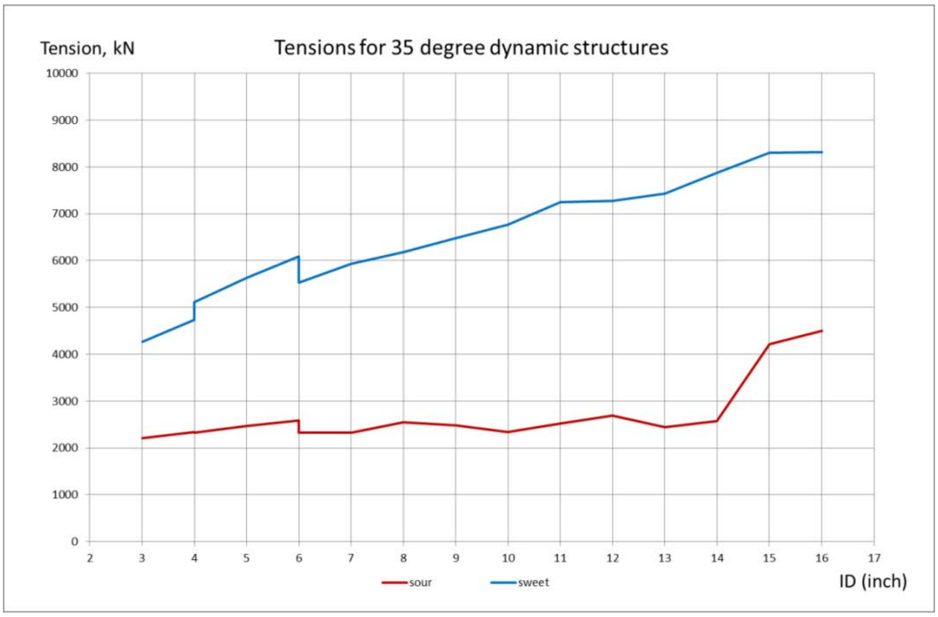
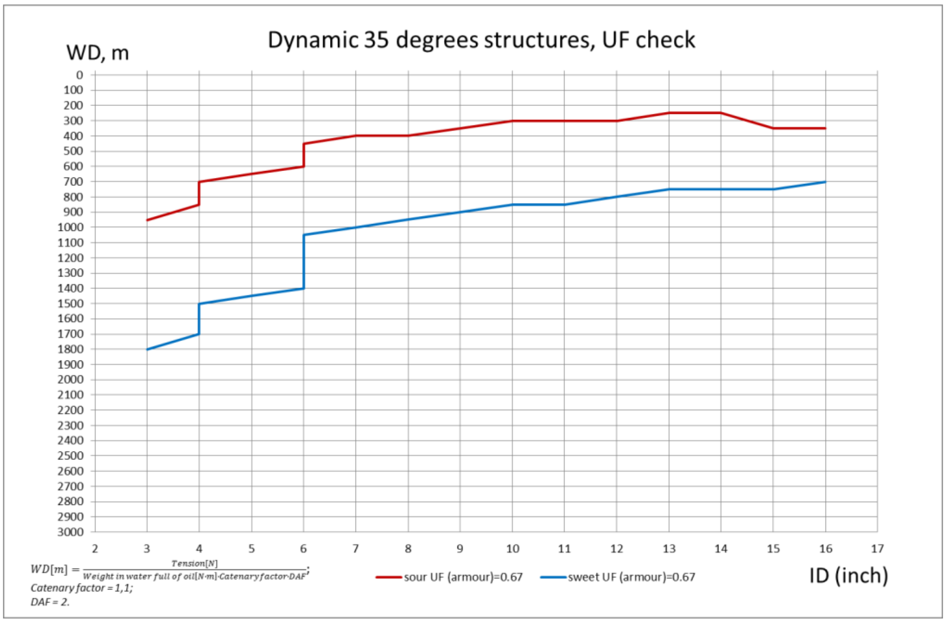


Figure 3.11 - Tensions obtained for the dynamic structures



$WD[m] = \frac{Tension[N]}{Weight\ in\ water\ full\ of\ oil[N \cdot m] \cdot Catenary\ factor \cdot DAF}$;
Catenary factor = 1,1;
DAF = 2.

Figure 3.12 - Water depth limitations obtained for the dynamic structures

Maximum water depths calculation for dynamic applications (35° tensile armour laying angle)

Straight collapse pressures are being consequently obtained for each structure with the “Structure” software. The calculated maximum allowable water depth is indicative as project engineering is performed considering the effect of curvature on the collapse resistance of the flexible pipe. Once the actual curvature at the touchdown point (TDP) is dependent upon the configuration definition, meteocean data and the floating unit motions, straight collapse value has been considered for simplification purposes.

The following formula is being used for the maximum allowable depth calculation:

$$WD = \frac{P_C \cdot MUF}{\rho_{sea\_water} \cdot g} = \frac{Pa}{\frac{kg}{m^3} \cdot \frac{m}{s^2}} = \frac{\frac{kg}{m^2} \cdot \frac{m}{s^2}}{\frac{kg}{m^3} \cdot \frac{m}{s^2}} = m; \qquad \text{Eq. 3.2}$$

Where P_C - collapse pressure calculated with the “Structure” software [Pa];

MUF - maximum allowable utilization factor (taken equal to 0.85 for all cases);

ρ_{sea_water} - density of a sea water (taken equal to 1029) [$\frac{kg}{m^3}$];

g - gravitational constant (taken equal to 9.81) [$\frac{m}{s^2}$];

So, we can simplify the equation above to:

$$WD = \frac{0.85P_C}{1029 \cdot 9.81} = 8.42 \cdot 10^{-5} P_C; \qquad \text{Eq. 3.3}$$

Or (if applying P_C in bars)

$WD = 8.42P_C$.

No major difference of collapse pressures was noticed for the scope of the sweet/sour, dynamic/static structures. Therefore, one set of numbers is assumed, as being relevant for both sweet/sour services. The results are presented in Table 3.9.

Table 3.9 - Water depths associated with the collapse performance of the flexible pipe

ID, "	Collapse, bar	Depths, m
3	323	2720
4	227	1911
5	171	1440
6	135	1137
7	159	1339
8	133	1120
9	145	1221
10	166	1398
11	145	1221
12	128	1078
13	140	1179
14	125	1053

The following figures show the flexible pipe collapse pressures and associated water depths, see Figure 3.13 and Figure 3.14.

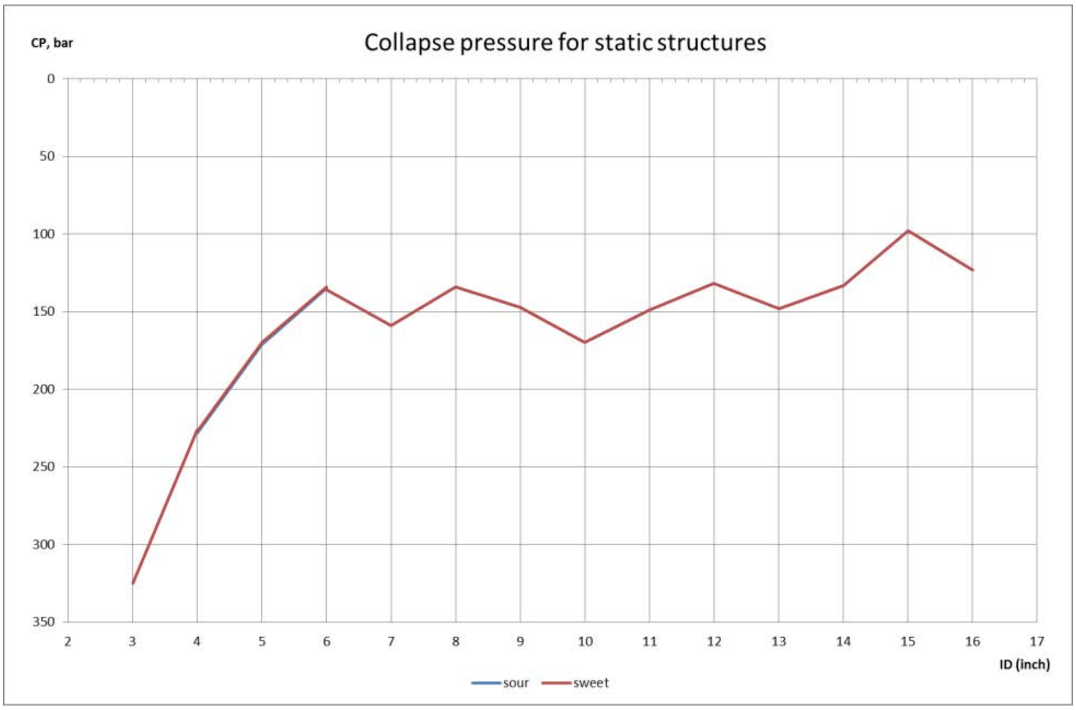


Figure 3.13 - Collapse pressures

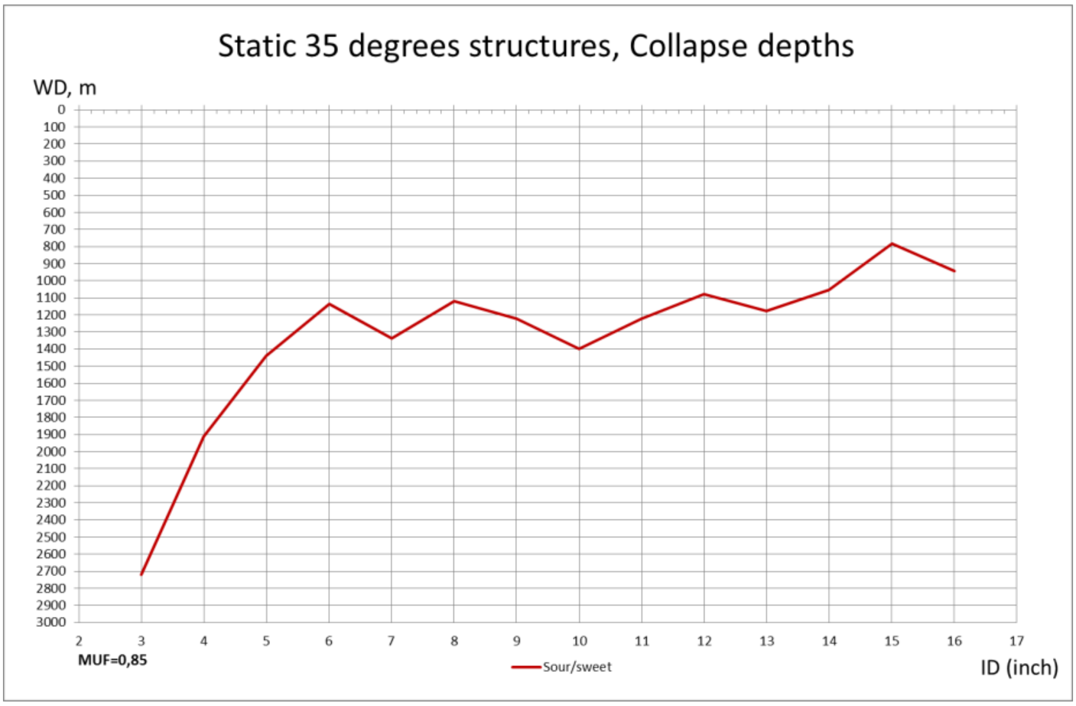


Figure 3.14 - Collapse depths

The summary of HPHT study is presented on the Figure 3.15 and Figure 3.16.

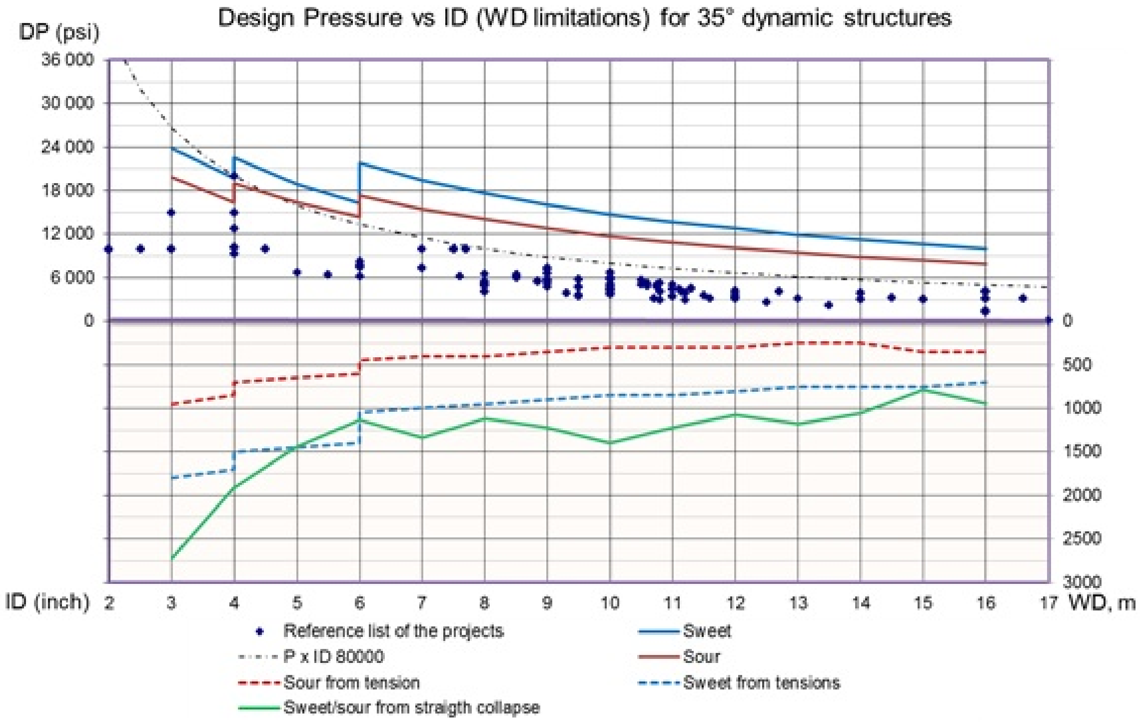


Figure 3.15 - Results of HPHT study for dynamic structures

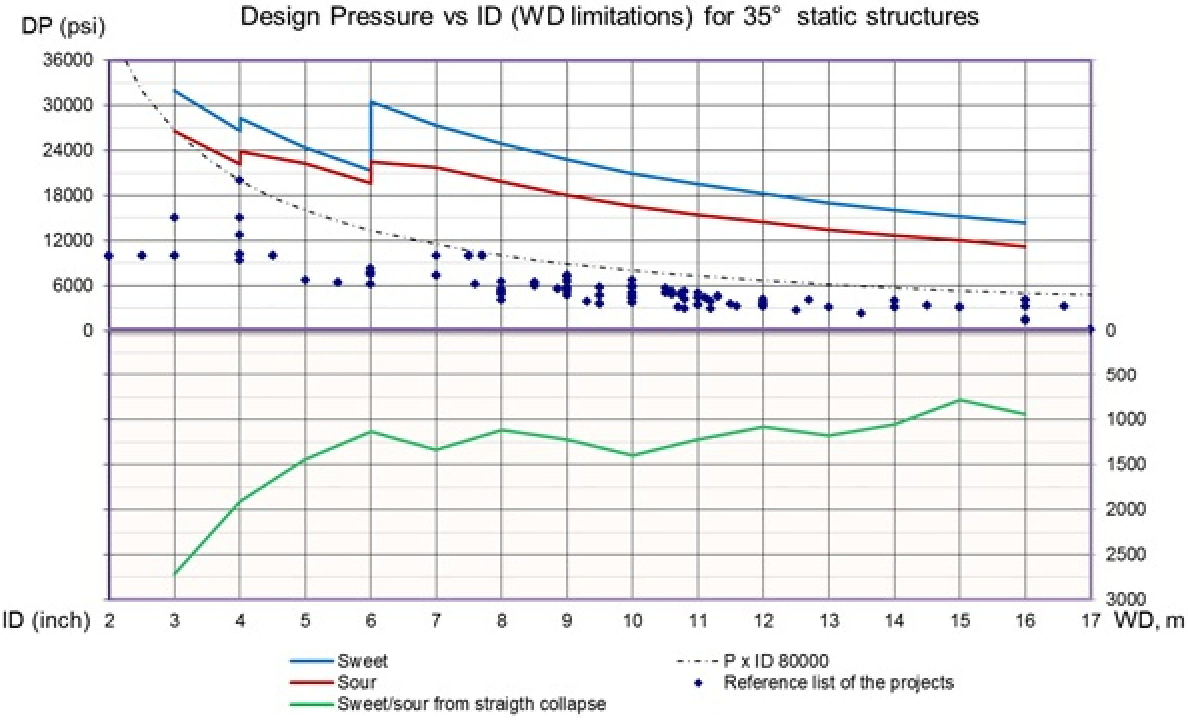


Figure 3.16 - Results of HPHT study for static structures

As seen from the charts, both the pressure capacities and the allowable water depths are decreasing, as the pipe ID increases. One can also notice that tension criteria are limiting over the collapse criteria for the dynamic applications with the design premises used in this study.

3.6.2 Maximum Design Pressure Comparison

Maximum design pressure (burst pressure) is driven by the hoop stresses, see Figure 3.17:

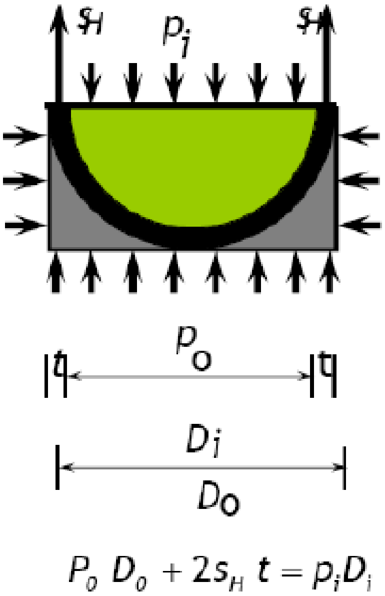


Figure 3.17 - Vertical equilibrium of unit length [22]

$$\sigma_h = \frac{p_i D_i - p_0 D_o}{2t}; \qquad \text{Eq. 3.4}$$

Where p_i – internal pressure;

p_0 – external pressure;

D_i – inner diameter;

D_0 – outer diameter;

t – wall thickness.

However, there are other simplified indicators of the hoop stresses [22]:

$$\sigma_h = \frac{pD}{2t}; \qquad \text{Eq. 3.5}$$

Where p – maximum allowable operating pressure;

D – outer diameter.

$$\sigma_h = \frac{(p_i - p_0)(D_o - t)}{2t}. \qquad \text{Eq. 3.6}$$

For oil and gas risers, a recommendation is to limit the hoop stress to 0.6·SMYS [21].

For the high diameter high pressure pipes studied in this paper significant stresses occur due to endcap effect (due to axial tensions created by internal pressure). End cap effect is the pressure effect in the pipe axial direction. The following equation shows end-cap stresses [22]:

$$\sigma_{ec} = \frac{\frac{\pi}{4}(D_i^2 P_i - D_0^2 P_0)}{A_{ST}}; \qquad \text{Eq. 3.7}$$

where A_{ST} – cross section of the pipe.

As seen from the Eq. 3.8, the End cap effect stresses and hoop stresses are both directly proportional to a pipe diameter. However, the higher End cap effect stresses are seen for the big diameters, due to the second order diameter/stress relationship.

The effect of this phenomena lead to higher stresses in the tensile armours, which has been proved with calculations, as burst appeared on tensile armours; e.g. from 6” for 45° Z12 dynamic structures, see Table 3.10.

Table 3.10 - Burst occurrence map

Service	Application	Laying angle for tensile armour	Zeta	Armour
Sour	dynamic	25°	X	
		35°	X	
		45°		Z12 from 6"
	static	25°	X	
		35°		Z12 16"
		45°		Z12 from 6"

The effect is more severe for the higher laying angles, as tensile armours absorb higher stresses (refer to Eq. 2.6).

The following graphs represent the variability of the maximum design pressures obtained; see Figure 3.18 and Figure 3.19.

As pressure capacities of 45° static/dynamic structures plunge down, 35° structures overperform the formers for big diameters e.g.; from 7” ID for sour static structures, see Figure 3.18.The phenomena were proved for both sweet and sour services

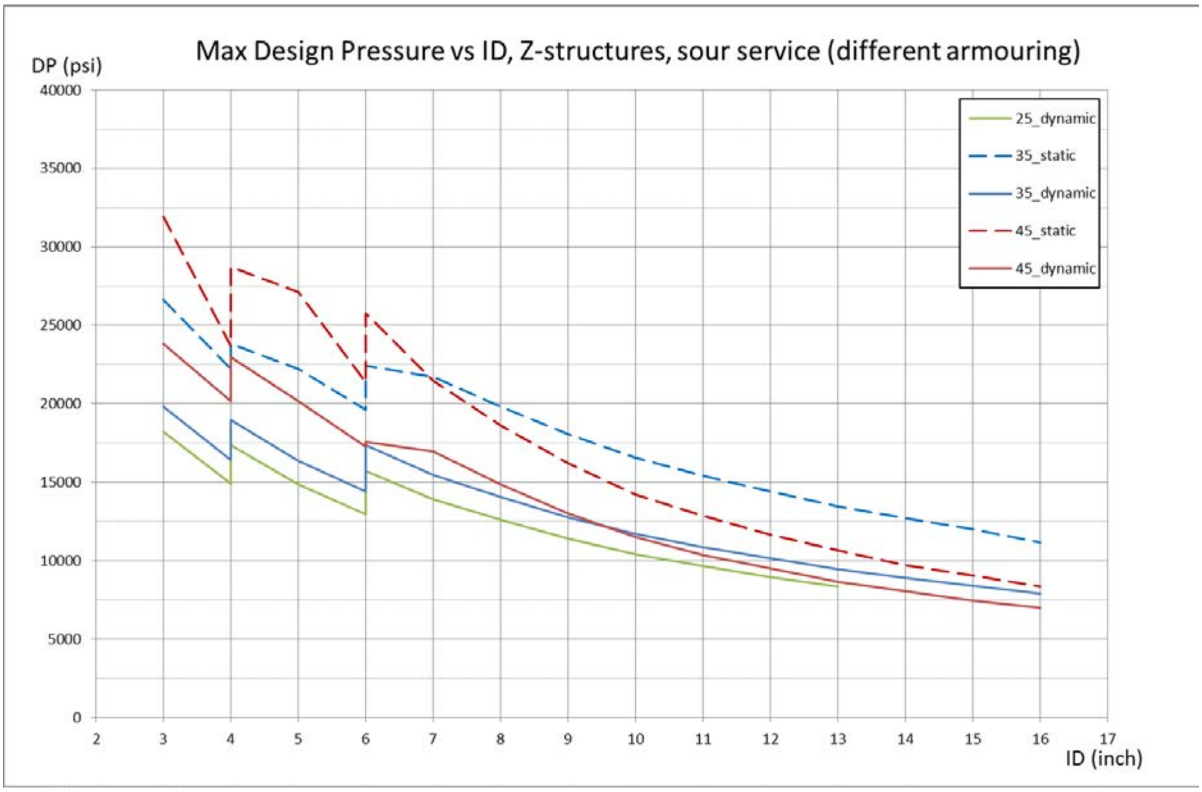


Figure 3.18 - Sour service, different armouring angles

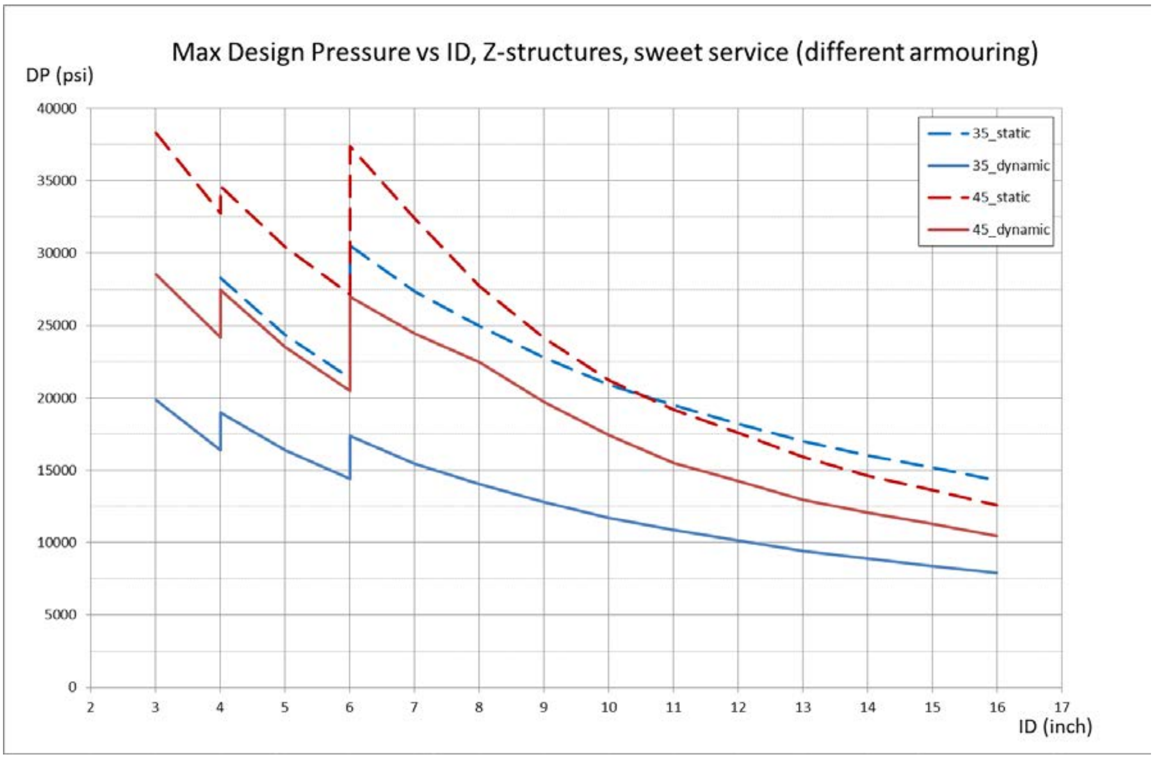


Figure 3.19 - Sweet service, different armouring

Conclusions

The following numerical milestones were identified in the course of this study for 35° flexible structures (present the HPHT capabilities), see Figure 3.16 and Figure 3.17:

1. Static sweet structure enables pressures up to 20500 psi for all pipes with ID less than 11”;
2. Static sour structure enables pressures up to roughly 20000 for all pipes with ID less than 8”;
3. Dynamic sweet structure enables pressures up to 15500 psi for all pipes with ID less than 10”;
4. Dynamic sour structure enables pressures up to 13000 psi for all pipes with ID less than 10”;
5. Dynamic sweet structure from the study with ID less than 8” can be deployed to water depths higher than 1000m (e.g. Brazil, USA applications);
6. Dynamic sour structure from the study with ID less than 8” can be deployed to water depths higher than 400 m (can be used as jumpers to FSHR systems);
7. Static structures from the study with ID less than 6” can be deployed to water depths higher than 1000 m.

Therefore the current design envelopes were increased, enabling to reach very high pressure domain (80 000 psi.inch). Armouring angle adjustment is a potential lever to even increase further the pressure capability summarized here. Design adjustment with limited impact on pressure capability could significantly increase the allowable water depth for installation. Multi-section section riser would also be a way to increase WD compared to the results presented.

3.7 Overview of the chapter

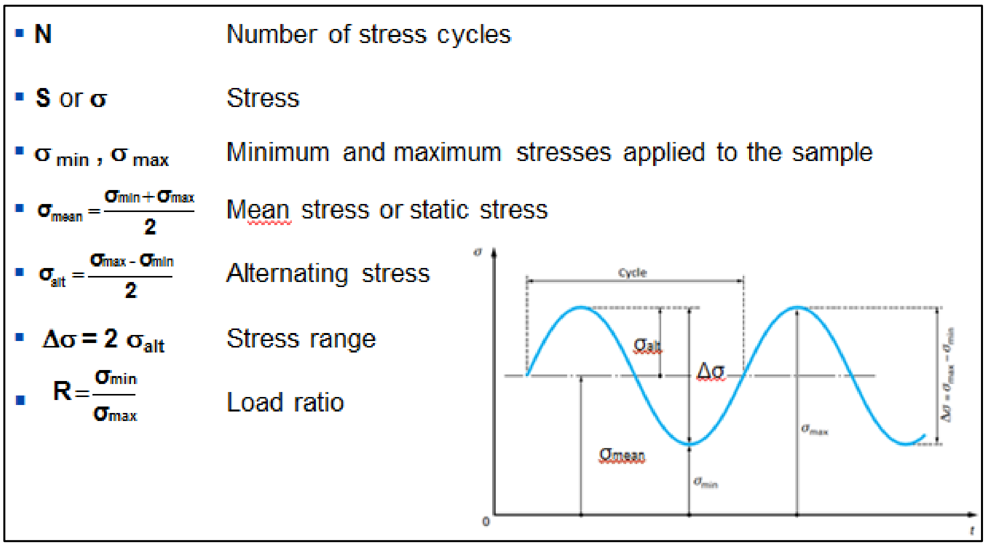
These are the main subjects discussed in the Chapter 3:

1. Objectives for the HPHT study are selected;
2. Burst pressure calculation methodology is described;
3. Role of the manufacturing process in pipe pressure capacities;
4. Introduction of the second spiral layer to a pipe structure;
5. Step-by-step design procedure and the basics of the material selection;
6. Calculation of maximum allowable water depths;
7. Driving mechanisms behind burst phenomena for different applications;
8. Results of the HPHT study.

CHAPTER 4 PRESSURE VAULT FATIGUE METHODOLOGY IMPROVEMENT

4.1 Introduction

Fatigue is a phenomenon leading to the early failure of a structure submitted to modest (compared to static tests) but repeated solicitations. Main fatigue parameters are shown on Figure 4.1.



N	Number of stress cycles
S or σ	Stress
σ_{min} , σ_{max}	Minimum and maximum stresses applied to the sample
$\sigma_{mean} = \frac{\sigma_{min}+\sigma_{max}}{2}$	Mean stress or static stress
$\sigma_{alt} = \frac{\sigma_{max}-\sigma_{min}}{2}$	Alternating stress
$\Delta\sigma = 2\ \sigma_{alt}$	Stress range
$R = \frac{\sigma_{min}}{\sigma_{max}}$	Load ratio

Figure 4.1 - Fatigue basics [24]

In the case of a flexible pipe, risers see lots of cyclic loadings due to vessel motion, waves, currents, so that fatigue has to be considered to calculate the service life of a riser structure [24].

When a flexible pipe is bent, the layers, and the wires in the each layer, slide with respect to each other to adopt the new shape. When the pipe is not pressurized, there is very little resistance to the sliding. In this case sliding of the wires occurs for very small curvature variations. However, when the pipe is pressurized, a contact pressure pushes the layers and the wires against each other. When a moment is applied, a friction force will prevent the wires from moving until the moment is large enough to overcome the friction and the wires start sliding. The higher the operating pressure, the higher the forces in the wires before sliding occurs and thus the higher the stresses, and the larger the fatigue damage induced. In high pressure and ultra-deep water designs, the tensile armours, as well as the pressure armours, are challenged. Life prediction is of uttermost importance for the pressure withstanding layers. The pressure vault lifetime prediction models usually integrate the pressure vault proprietary geometry and all of the experience gained from full scale tests. The Service Life Prediction Model was initiated in the early 90's and has been further validated by each full scale fatigue test performed since [26].

Design is classically evaluating the service life in:

1. Dry conditions (dry annulus): (air) fatigue;
2. Wet conditions (flooded annulus): corrosion fatigue [24].

Fatigue in pressure armours is generally more critical for high pressure pipes and the key drivers are internal pressure and dynamic curvature ranges. Other parameters that affect pressure armour fatigue characteristics include wire shape, material, manufacture process and test pressure history. [1].

One of the advantages of using flexible pipes instead of rigid steel pipes in offshore systems is the compliance of the formers with the movements of floating facilities and, furthermore, the ability to absorb harsh environmental loads. These characteristics derive from its internal structure in which the individual layers are allowed to slide relative to each other. These movements and environmental loads, however, may provoke high tension and curvature variations in the pipe, which may lead to fatigue failure and/or the wear of the metallic layers. Among all metallic layers of a flexible pipe, its tensile armors are especially prone to fatigue failure phenomenon [9].

Petroleum Safety Authorities (PSA), however, have reported fatigue to be the least cause of flexible riser incidents offshore Norway, please see Figure 4.2. The PSA study recall all major incidents occurred from 1995 to 2013. Interestingly, the PSA reveal the corrosion fatigue and corrosion degradation processes as the weak points within modern fatigue design procedures and state the need for improvements in these areas.

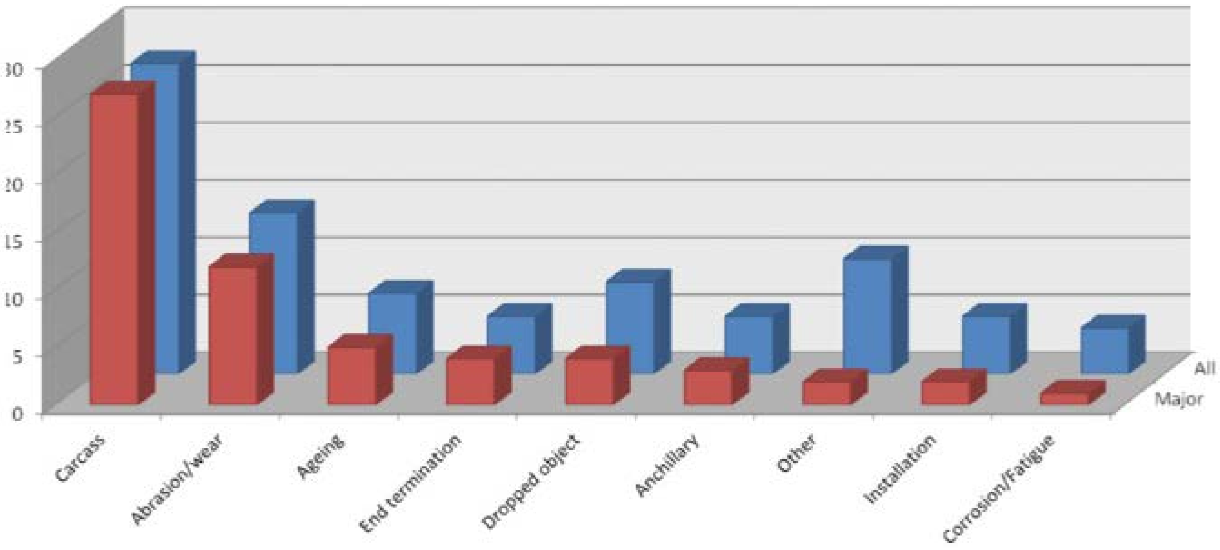


Figure 4.2 - Flexible riser incidents reported to Petroleum Safety Authorities/modified [28]

4.2 Fatigue analysis process

The fatigue analysis process involves several stages and these are, see Figure 4.3:

1. The global fatigue in the flexible riser/jumper;
2. Transposition of load responses obtained from the global analysis for input to the local analysis;
3. The local stress analysis in the steel tensile/pressure armour wires;
4. Fatigue life calculation.

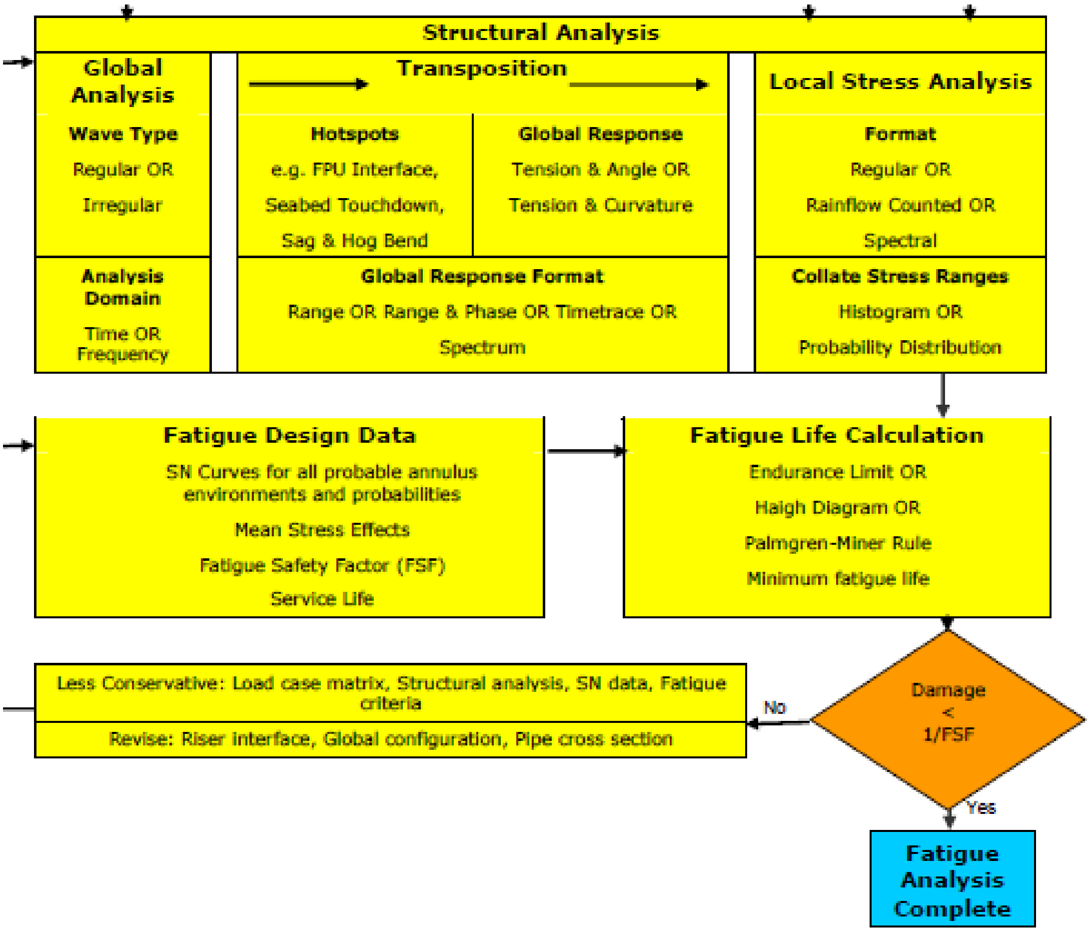


Figure 4.3 - Flowchart of global/local fatigue analysis methodology [1]

The global fatigue analysis is performed as part of the system design analysis in order to provide input to the local fatigue analysis. This can be done in several ways. The classical approach is to use a stochastic approach where the wave scatter is divided into a number of representative blocks, screening for the critical periods with regard to vessel response and discretizing the wave scatter in such a way that the number of occurrences around the critical periods is minimized. The discretization is obtained by a frequency screening of the governing parameters like curvature variations and tension variations. The selected wave bins are then applied to the global analysis model as regular waves with the corresponding period. The simulation is continued for a sufficient number of periods to reach a steady state of the riser system.

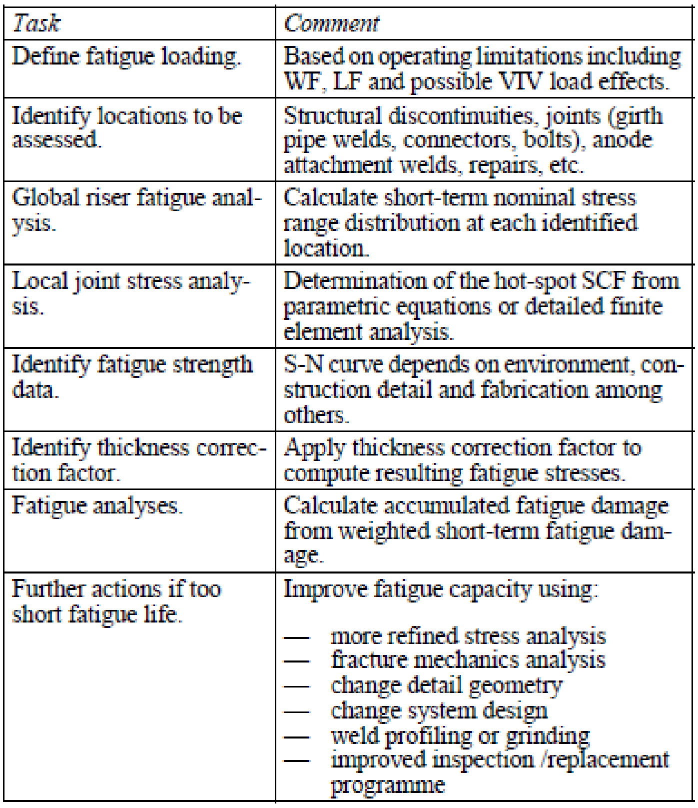
The data are transpositioned into the local model as curvature and tension variations for each element along the length of the riser. In order to be able to account for the mean stress components effect on the stress ranges, the mean curvatures and mean tensions are also considered in a local analysis.

The last step of the fatigue analysis is the calculation of the fatigue life with the consequent design approval [8].

Det Norske Veritas presents the fatigue analysis process in 7 steps (please also see Table 4.1):

1. Defining loading;
2. Identification of fatigue hotspots;
3. Global analysis;
4. Local analysis;
5. Identification of fatigue strength data (correction factors);
6. Fatigue analysis;
7. Adjustments to structure/refinement of the analysis [13].

Table 4.1 - Summary of a typical fatigue assessment procedure [12]



Task	*Comment*
Define fatigue loading.	Based on operating limitations including WF, LF and possible VIV load effects.
Identify locations to be assessed.	Structural discontinuities, joints (girth pipe welds, connectors, bolts), anode attachment welds, repairs, etc.
Global riser fatigue analysis.	Calculate short-term nominal stress range distribution at each identified location.
Local joint stress analysis.	Determination of the hot-spot SCF from parametric equations or detailed finite element analysis.
Identify fatigue strength data.	S-N curve depends on environment, construction detail and fabrication among others.
Identify thickness correction factor.	Apply thickness correction factor to compute resulting fatigue stresses.
Fatigue analyses.	Calculate accumulated fatigue damage from weighted short-term fatigue damage.
Further actions if too short fatigue life.	Improve fatigue capacity using: — more refined stress analysis — fracture mechanics analysis — change detail geometry — change system design — weld profiling or grinding — improved inspection /replacement programme

Technip fatigue analysis procedure is presented below, see Figure 4.4.

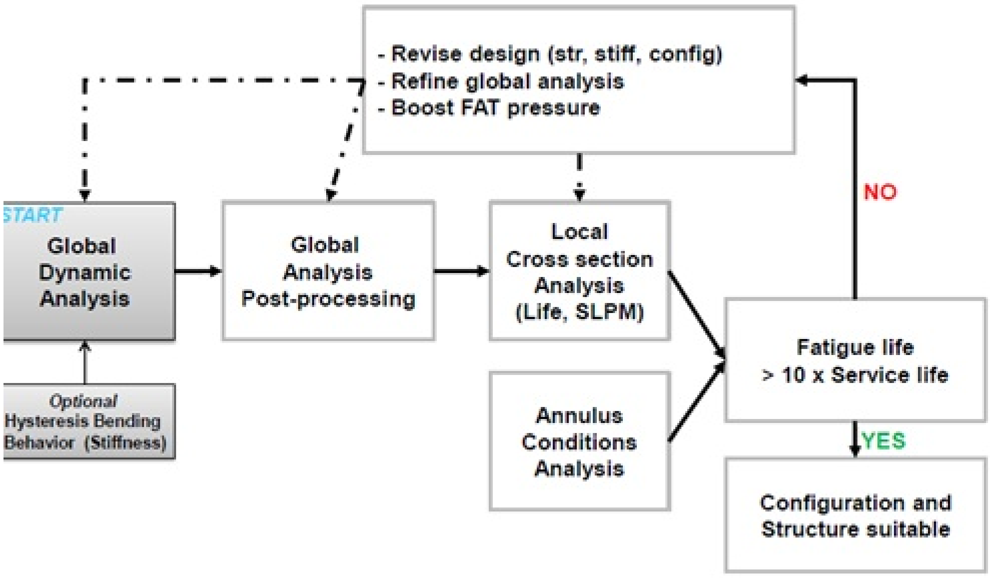


Figure 4.4 - Fatigue methodology overview [32]

The current study elaborates on the local cross section analysis and estimates the damage occurred on pressure. In addition, the main objective of the study is to propose a simplified methodology for an early fatigue severity indicator.

4.3 Fatigue Failure Modes

The fatigue analysis shall be based on the knowledge of the failures modes of each layer together with the pipe behavior. The most important failure modes are:

1. Fatigue of the tensile armors;
2. Fatigue of the pressure armors;
3. Corrosion fatigue;
4. Fatigue of the polymeric layer;
5. Fatigue in the armors at the end fittings [5].

4.3.1 Failure modes in pressure armours

First failure mode is longitudinal (see Figure 4.5 and Figure 4.6), when a crack initiates in the internal groove on the tight or compressed fibre of the pipe, propagates in depth along the groove towards the neutral fibre of the pipe. This failure mode is the most frequent one and therefore is taken as the base for the fatigue damage calculation.

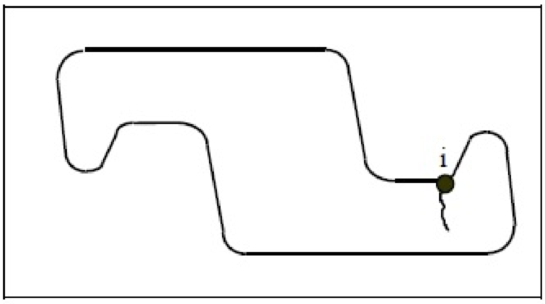


Figure 4.5 - Starting a longitudinal fracture

Figure 4.6 - Longitudinal fracture

Second possible failure is transverse. It initiates on the neutral fiber of the pipe on the edge of the hook and may propagate in the whole section (see Figure 4.7 and Figure 4.8)

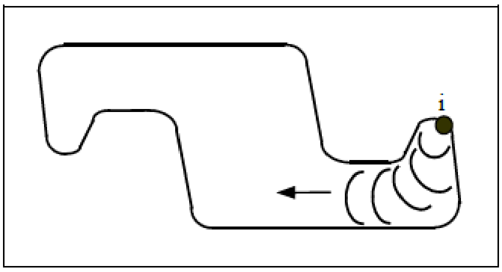
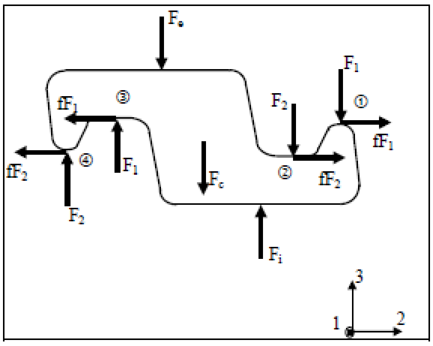


Figure 4.7 - Starting a transverse fracture

Figure 4.8 - Transverse fracture

The stresses on the groove are essentially due to the evolution of the forces between two adjacent Zeta wires. These forces are presented in the following Figure 4.9:



Where F_1& $F_2$are the contact forces;

F_i& $F_e$are the forces due to internal and external contact pressures;

$F_c$is the radial component of the force generated by the hoop stress.

Figure 4.9 - Forces on the Zeta

When the curvature increases, the force distribution varies and leads to only one contact when a certain curvature (called critical curvature) is reached. The critical curvature variation depends on several parameters like diameter, mean radius, type of Zeta, contact pressures.

The effect of critical curvature will be discussed in the next sections.

4.3.2 Failure in tensile armours

The resulting stress distribution in the tensile armour wire cross-section is governed by the global loads due to internal bore pressure, external ambient hydrostatic pressure, riser tension and bending. There are three main stress contributions in a tensile armour wire, ie. due to global riser tension and bending, and from inter-layer friction. The wire tension is governed by the loads from carrying the selfweight of the riser, including its dynamic response, as well as the endcap load due to the pressure difference over the riser cross-section. It may be noted, that the tension and bending induced wire stresses can be considered linear functions of the global riser tension and curvature, respectively.

The stresses from friction exhibit a hysteresis behavior with curvature. For a small curvature the bending stiffness is high due to the stick behavior of the tensile armour wires, but at a specific critical curvature the tensile armour wires slip and the bending stiffness decreases. The stresses from friction are governed by the contact pressures between the individual pipe layers and their coefficient of friction [8].

The selection of materials is of paramount importance for the fatigue design. The correct specification of the internal conveyed fluid, not only at the beginning of the field life but during all its life, together with a careful integrity control, will be the input for a correct evaluation of the pipe annulus environment which the wires will be subject to. The effect of the annulus environment (that could evolve from sweet to sour during pipe's service life) can be schematically seen in the S-N curves shown in Figure 4.10. For a flexible pipe, the effect of the mean stress level shall be also taken into account [16].

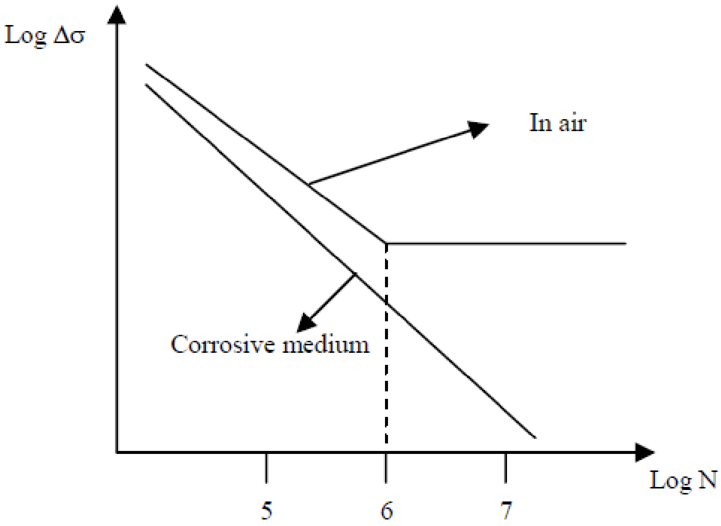


Figure 4.10 - S-N Schematic curves for corrosive and air environment [16]

High strength steel armour wires are required in deepwater risers to reduce weight. Whereas their fatigue performance in a dry environment is similar to lower strength steel grades, their corrosion fatigue properties are commonly reduced in connection with sour service. In case the annulus is partly or entirely water filled the armour wires can be subjected to a corrosive environment that changes significantly along the length of the riser due to the hydrostatic pressure head in the annulus.
Consequently, the fatigue assessment may require that different S-N curves be applied along riser sections reflecting the different partial pressures of the permeated constituents [8].

4.4 Standards

The fatigue design checks are required to verify that the computed fatigue life exceeds the intended service life with a specified factor of safety. The objective is to ensure a high level of reliability in the fatigue design [1].
The following are the main standards governing fatigue behavior of the flexible pipe:

1. API Spec 17J The code specifies core requirements for fatigue design of flexible pipes [2];
2. API RP 17B The code gives guidelines on design, analysis, manufacture, installation and operation of flexible pipes systems [3];
3. DNV-OS-F201 and DNV-RP-F204 The codes contain extensive guidance on global analysis and fatigue criteria that may also be applicable to flexible risers [12].

4.5 Software and prediction models

There are a variety of local models that can be used in the fatigue analysis and generally these are categorized as follow:

1. Analytical or semi-analytical;
2. Finite element models.

The advantage of the analytical or semi-analytical models includes ease of use, simplicity of input and output and efficiency of processing multiple load cases. The advantages of the finite element models include ability to model specialized details, such as localized stiffening effects from end fittings. The disadvantages of the analytical models include assumptions of uniform pipe curvatures. The disadvantages of the finite element models include difficulty of use and potential for modelling errors.

The minimum requirements for the local analysis models are as follow:

1. Verified against full-scale measurements;
2. Capable of modelling tension and curvature ranges;
3. Accounting for hysteresis effects;
4. Calculate stresses at four corners of the wire;
5. Take into account effects of external pressure;
6. Preferably output stresses at 8 points around the circumference, so that directionality effects can be considered [1].

The main objective of this study is to assess the fatigue methodology for the high pressure pipes and to propose a simplified methodology for early fatigue severity indicator. Current section will be limited to a pressure vault behaviour. However, both the pressure vault and the tensile armours were investigated at in the course of the project work.

Technip’s in-house Service Life Prediction Model is designed to estimate the fatigue life of pressure vault layers for any dynamic flexible pipe.

This program calculates the mean and the alternate stresses in the pressure vault when the structure is submitted to several sets of loadings including internal and external pressure loads, tension and in-plate cyclic bending motion. Through the multi axial Sines criterion, those stresses are compared to the allowable stresses for a given material, environment and number of cycles. This comparison is made for each wave class. The total design life and fatigue damage is evaluated with the Milner’s rule:

$$D_i = \frac{n_i}{N_i}; \qquad \text{Eq. 4.1}$$

where D_i- the partial damage for a given wave class i (note i corresponds to the bin number from the scatter diagram);

n_i-the number of cycles experienced by riser for the wave class i;

N_j-the number of cycles to failure for the given mean stress and alternating stress reached during wave class i.

The linear damage of the material is determined using Miner’s rule which accounts for the contribution of each wave/curvature class to the total damage. The cumulated damage is the sum of the partial damages:

$$D = \sum_i D_i = \sum_i \frac{n_i}{N_i} . \qquad \text{Eq. 4.2}$$

The average design life for the flexible risers normally varies between 20 and 30 years. It is required for the flexible risers to demonstrate a safety factor of 10 compared to the design life in normal operating conditions. Therefore, the maximum acceptable damage is 0.1.

SLPM inputs for the local cross section analysis are:

1. Flexible pipe structure file;
2. Haigh Diagram (environment: annulus conditions, probability of failure, see Figure 4.11);
3. Wave Classes Data (from dynamic analysis);
4. Tensions (Min and Max values);
5. Pressures (from Operating Conditions and water depth);
6. FAT pressure .

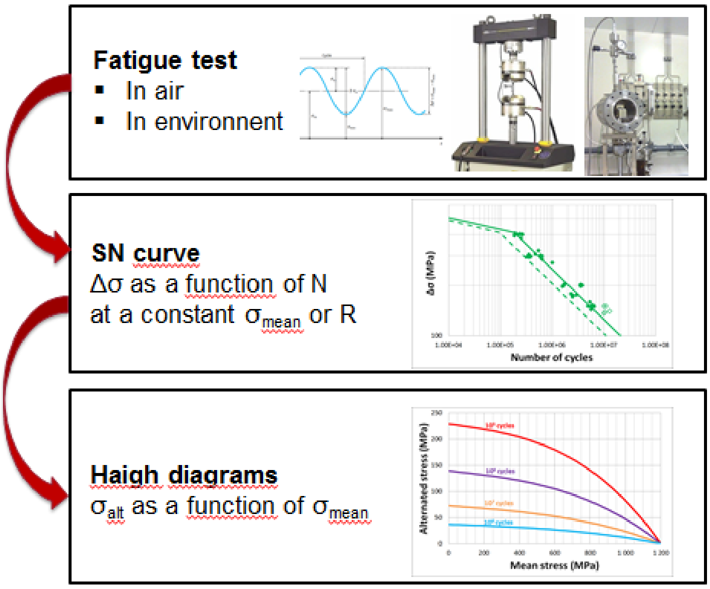


Figure 4.11 - Material data input for the SLPM [32]

The main results of the SLPM calculations are:

1. Cumulated damage in any of the pressure vault layers;
2. Pipe critical curvature;
3. Alternating stress in the layer.

4.6 Study of fatigue limitations for high pressure flexible pipes

4.6.1 Introduction

Technip flexible pipe inventory accounts for almost 12.5 thousand items with a vast majority of applications: oil/gas production, acid transport, gas-lift, water injection and so on. Up until today Technip flexibles have been sold to almost 60 countries around the world. Pipes are bringing both sweet and sour services, fulfilling client's specifications: test samples, riser structures, jumpers and flowlines.

Group of projects investigated in the course of this fatigue study (26 projects) was selected to match the following criteria:

1. Dynamic application;
2. Rough bore structure;
3. Highest possible design pressure (depending on the pipe internal diameter);
4. Recently commenced projects.

Both the pressure and tensile armour layers of the flexible are susceptible to fatigue damage. However, fatigue in the pressure armour can occur in case of large riser curvatures causing excessive ovalisation of the cross section due to bending.

Therefore the aim of this report is to assess the Zeta layer fatigue performance under high curvature variations. Moreover, few cases of Teta vault structures are also illustrated.

Flexible pipe is subjected to various tension and curvature variations under the different geographical environments. Characteristic loadings (sea states, tensions and curvatures) endured by Technip's flexible pipes were analyzed for different regions, such as: 1. West Africa, 2. Norway/the UK, 3. the Gulf of Mexico (GOM).

The definition of the pipe curvature can be seen via its bending radius, see Figure 4.12.

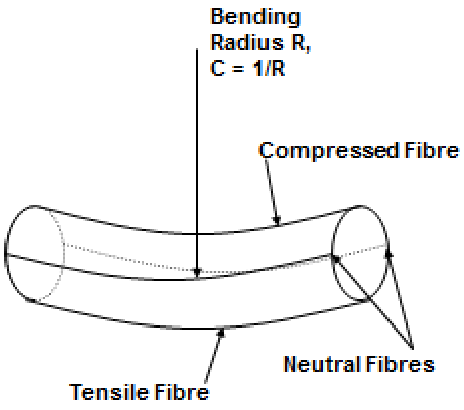


Figure 4.12 - Flexible pipe curvature definition [32]

Global fatigue to local transposition is conducted at locations where the bending (tensions) ranges of the flexible pipe are comparatively large. These locations are known as hotspots in a fatigue analysis. In this study the top riser section was considered as the hotspot. High pressure pipes create very high tensions due to the additional flat spiral and the second pair of tensile armour wires incorporated in structure. Flexible pipe hotspots are also shown below, see Figure 4.13.

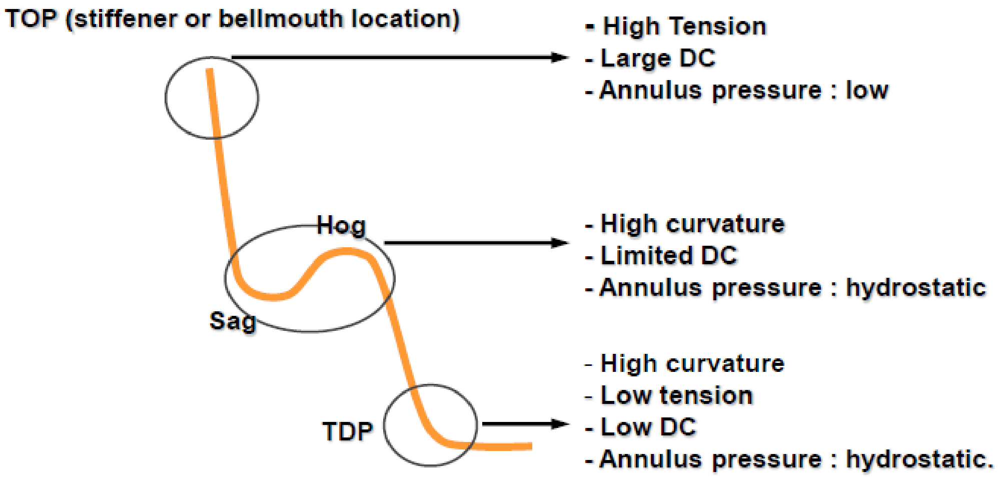


Figure 4.13 - Fatigue hotspots locations

4.6.2 Analysis of the local fatigue data

In the frame of the current study an attempt to simplify the process of local fatigue analysis by introducing indicators/predictors of the service life-in-air will be made.

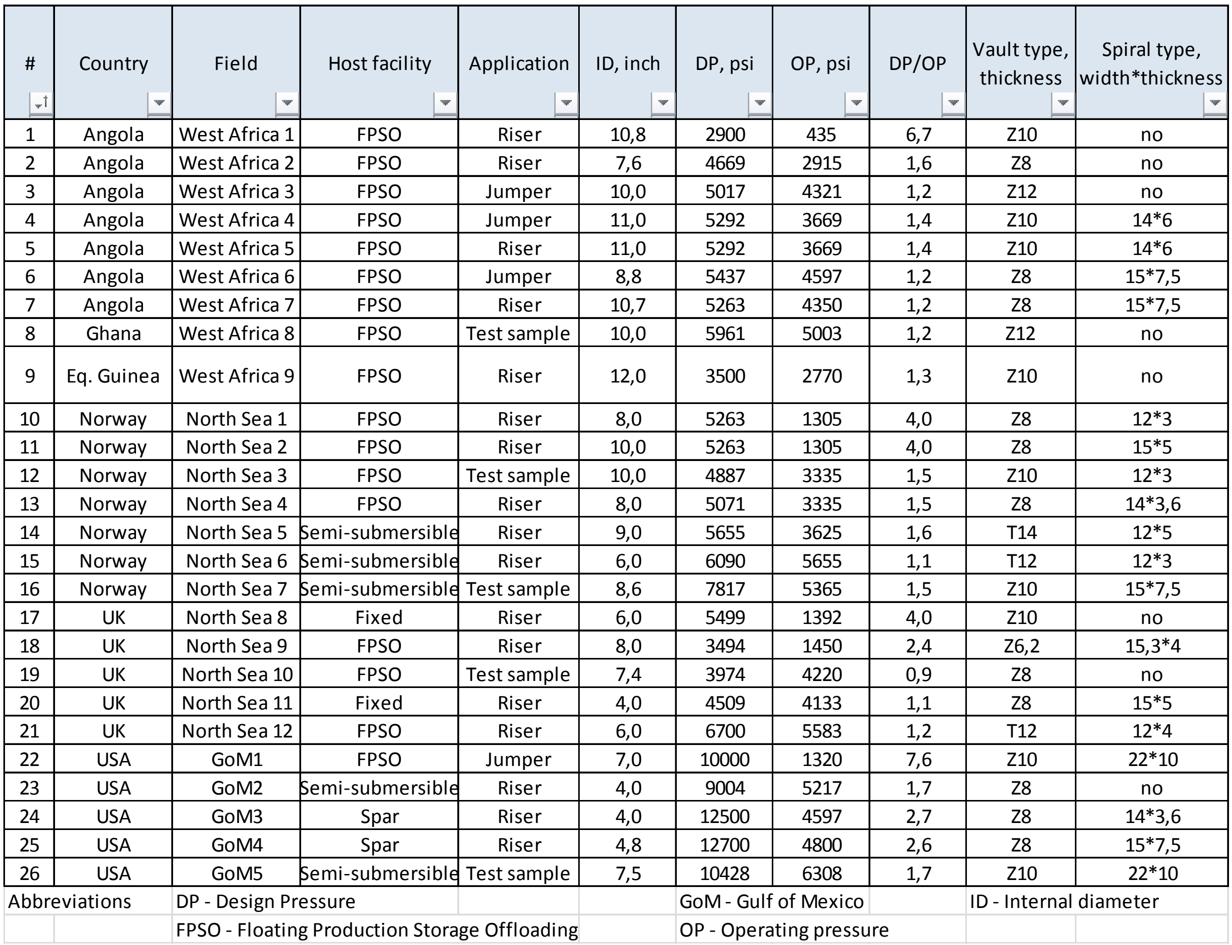
The objective of the project is to have a rapid check of fatigue performance of the pipe without entering rigorous and time-consuming complete global and local fatigue analysis.

Moreover, the project targets to establish the indicator for the maximum operation pressure a given pipe can withstand in the selected fatigue environment.

The impact of the sour fatigue service will also be discussed.

Projects investigated in the course of the study are shown below, see Table 4.2.

Table 4.2 - Technip high pressure projects



#	Country	Field	Host facility	Application	ID, inch	DP, psi	OP, psi	DP/OP	Vault type, thickness	Spiral type, width*thickness
1	Angola	West Africa 1	FPSO	Riser	10,8	2900	435	6,7	Z10	no
2	Angola	West Africa 2	FPSO	Riser	7,6	4669	2915	1,6	Z8	no
3	Angola	West Africa 3	FPSO	Jumper	10,0	5017	4321	1,2	Z12	no
4	Angola	West Africa 4	FPSO	Jumper	11,0	5292	3669	1,4	Z10	14*6
5	Angola	West Africa 5	FPSO	Riser	11,0	5292	3669	1,4	Z10	14*6
6	Angola	West Africa 6	FPSO	Jumper	8,8	5437	4597	1,2	Z8	15*7,5
7	Angola	West Africa 7	FPSO	Riser	10,7	5263	4350	1,2	Z8	15*7,5
8	Ghana	West Africa 8	FPSO	Test sample	10,0	5961	5003	1,2	Z12	no
9	Eq. Guinea	West Africa 9	FPSO	Riser	12,0	3500	2770	1,3	Z10	no
10	Norway	North Sea 1	FPSO	Riser	8,0	5263	1305	4,0	Z8	12*3
11	Norway	North Sea 2	FPSO	Riser	10,0	5263	1305	4,0	Z8	15*5
12	Norway	North Sea 3	FPSO	Test sample	10,0	4887	3335	1,5	Z10	12*3
13	Norway	North Sea 4	FPSO	Riser	8,0	5071	3335	1,5	Z8	14*3,6
14	Norway	North Sea 5	Semi-submersible	Riser	9,0	5655	3625	1,6	T14	12*5
15	Norway	North Sea 6	Semi-submersible	Riser	6,0	6090	5655	1,1	T12	12*3
16	Norway	North Sea 7	Semi-submersible	Test sample	8,6	7817	5365	1,5	Z10	15*7,5
17	UK	North Sea 8	Fixed	Riser	6,0	5499	1392	4,0	Z10	no
18	UK	North Sea 9	FPSO	Riser	8,0	3494	1450	2,4	Z6,2	15,3*4
19	UK	North Sea 10	FPSO	Test sample	7,4	3974	4220	0,9	Z8	no
20	UK	North Sea 11	Fixed	Riser	4,0	4509	4133	1,1	Z8	15*5
21	UK	North Sea 12	FPSO	Riser	6,0	6700	5583	1,2	T12	12*4
22	USA	GoM1	FPSO	Jumper	7,0	10000	1320	7,6	Z10	22*10
23	USA	GoM2	Semi-submersible	Riser	4,0	9004	5217	1,7	Z8	no
24	USA	GoM3	Spar	Riser	4,0	12500	4597	2,7	Z8	14*3,6
25	USA	GoM4	Spar	Riser	4,8	12700	4800	2,6	Z8	15*7,5
26	USA	GoM5	Semi-submersible	Test sample	7,5	10428	6308	1,7	Z10	22*10
Abbreviations		DP - Design Pressure				GoM - Gulf of Mexico			ID - Internal diameter	
		FPSO - Floating Production Storage Offloading				OP - Operating pressure				

Table illustrates a great variety of projects, however elaborates on use of the flat spiral pressure reinforcement. However, few cases of high pressure pipes without the flat spirals are also considered in this study. Role of Teta pressure vault will also be illustrated in this paper.

Fatigue analysis reports of each flexible pipe were studied in order to obtain the data needed for the vault damage calculations, such as:

1. Service life of the flexible pipe in years;
2. Operating pressure of the flexible pipe (as the main parameter for the long-term fatigue effects). This parameter can only be found in the fatigue reports;
3. FAT pressure used during pipe manufacturing;
4. Information about the most severe wave class in the area (results of the global fatigue analysis):

- Number of cycles of wave loading, that flexible will see during its service life;
- Heading angle for the selected wave class.

5. Tensions in the flexible associated with the selected wave class.

As mentioned before, all the data was gathered for the top section of the flexible (under the bend stiffener, below the hang-off point). This area accounts for the largest curvature variations.

The curvature distributions are presented via min/max, 25/75 per cent quantile values. Box and whiskers diagrams were used to illustrate the curvature variations for different applications.

Three different areas can be selected with respect to the dynamism of the application (see Figure 4.14).

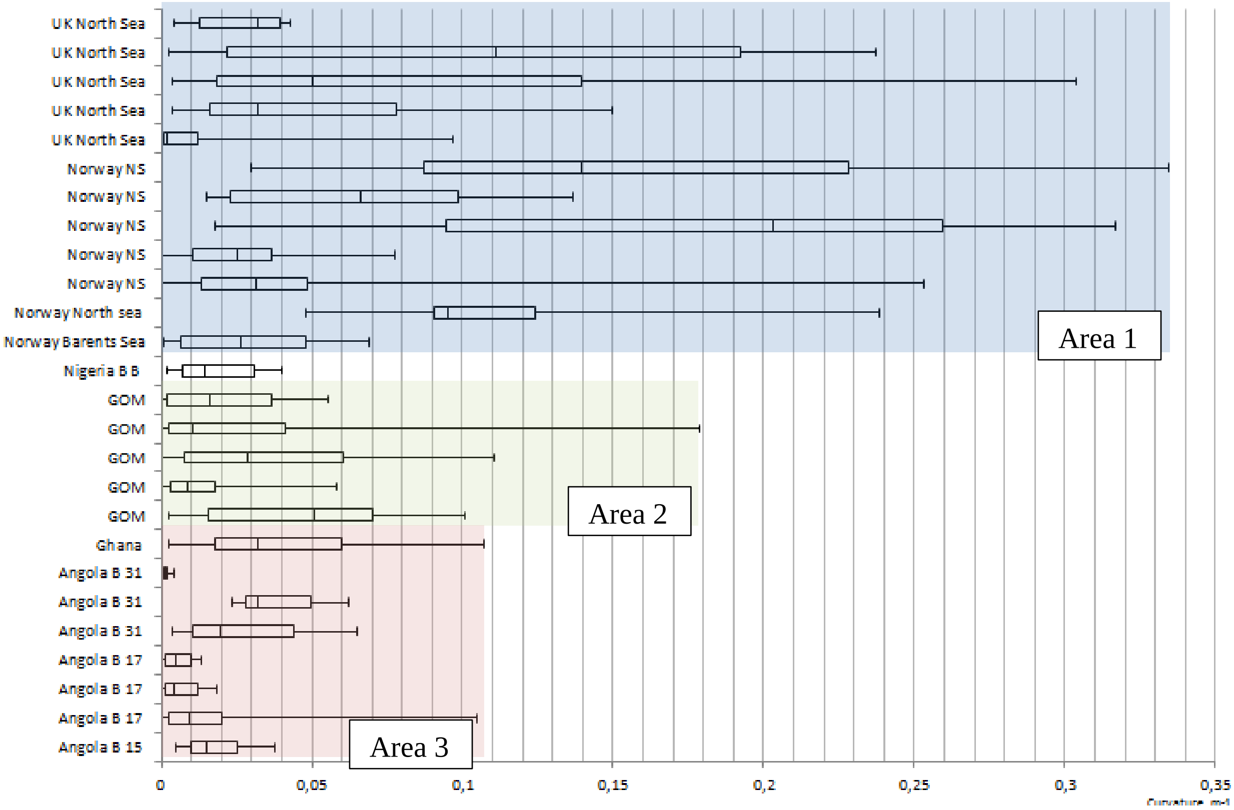


Figure 4.14 - Dynamism of the applications

These areas, expressed in the maximum curvature variations, used for modeling of the pipe behavior are:

1. UK/Norway (most severe environment);
2. GoM;
3. West Africa.

Therefore, a flexible pipe design engineer can use the above data to assign characteristic loadings (mean number of cycles and curvature variations) to any new flexible pipe.

As seen from Figure 4.15, semi-submersible platforms were mostly used in area 1 along with fixed and FPSO solutions. Low dynamism challenges of African projects were resolved with myriad of FPSO platforms.

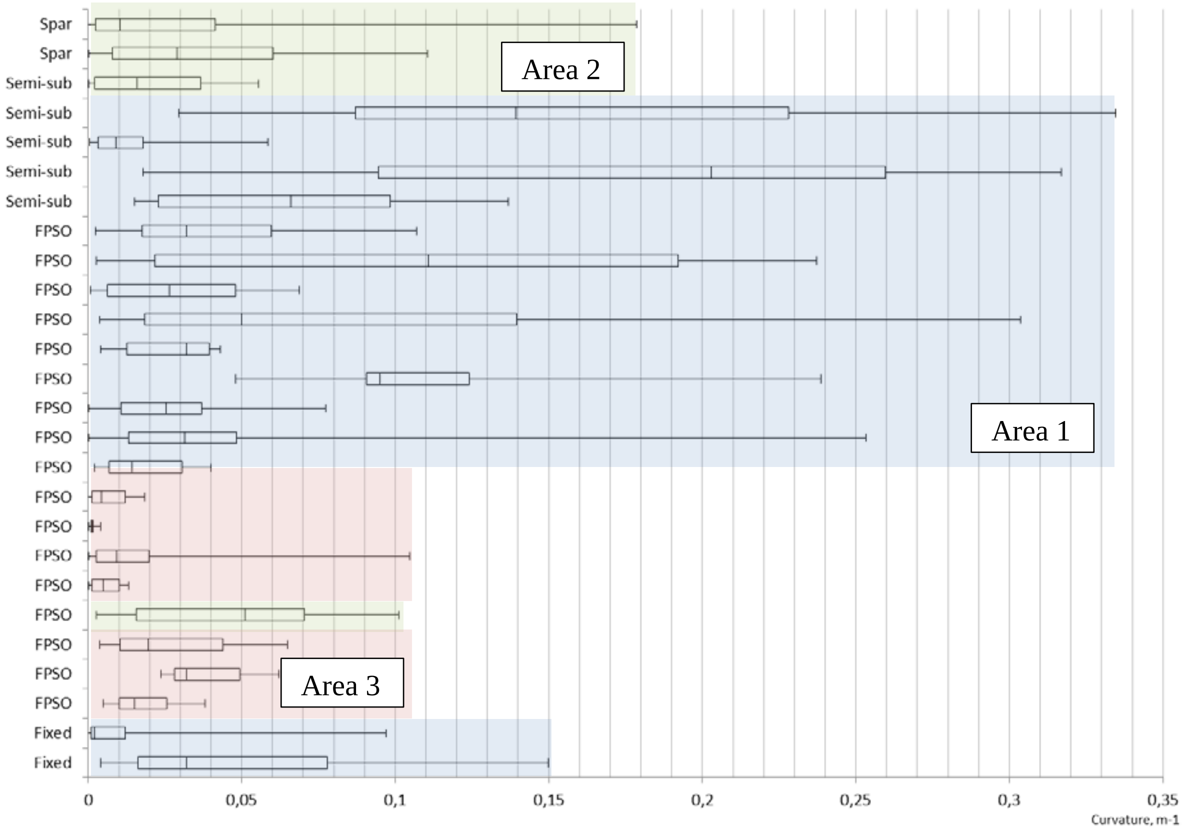


Figure 4.15 - Platforms' variability

Interestingly, high diameter flexibles produced by Technip were used in the limited curvature variation environments, see Figure 4.16.

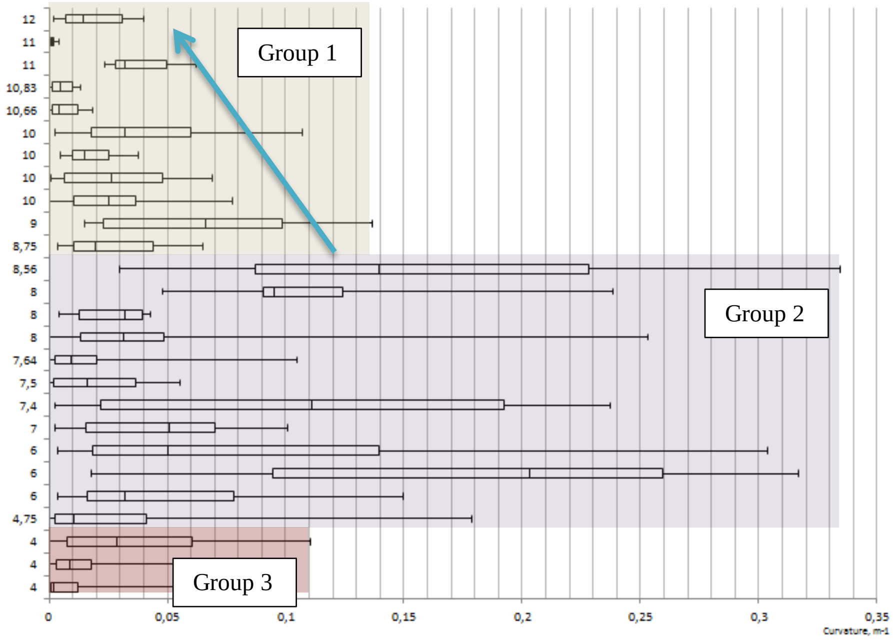


Figure 4.16 - Curvature versus internal diameter of the flexibles

First group of 8.5 to 12 inch pipes has an intermediate curvature range. However, we can see the following trend: the higher the diameter of the application, the lower is the curvature range a flexible pipe has been designed for. This trend can be explained by the increased use of subsea manifolds in West Africa and Norway, when high diameter pipes used in order to ensure the production from the reservoirs (FPSO only). Flexible pipe properties of large diameter pipe (e.g. bending stiffness) tend to reduce the curvature seen by the flexible pipe.

Group two represents all geographical areas and applications. It accounts for the majority of current Technip projects.

Group three shows low diameter American Spar projects.

Design engineer can therefore select an appropriate fatigue criteria based on the pipe inner diameter.

Few cases of Teta vault high pressure flexible pipes were selected (UK/Norway, refer to Table 4.2), Figure 4.17.

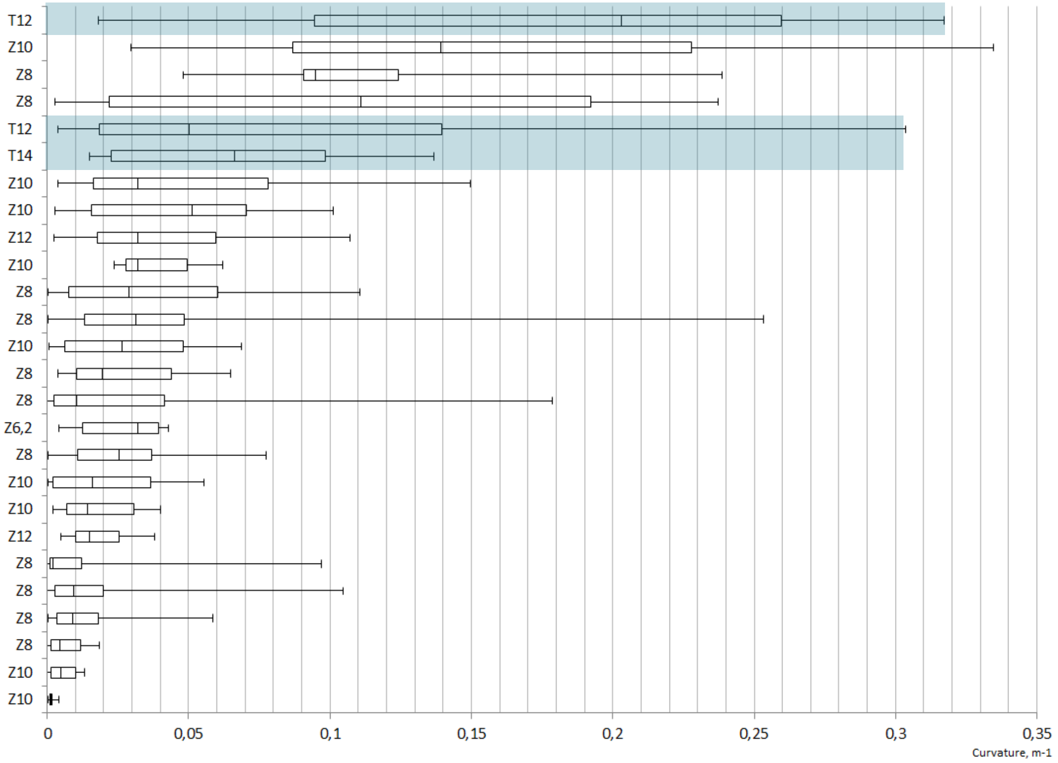


Figure 4.17 - Vault type versus curvature variations (ranged by the mean value)

The nature of the critical curvature (DCC) of pressure vault was previously underlined. Flexibles noticed to be subjected to a higher damage in vault when at one point of contact between Zeta wires (when curvature is higher than critical value). Therefore, the curvatures endured by the flexible pipes (obtained from the SLPM software) were compared to the critical curvatures, see Table 4.3.

Table 4.3 - Critical curvature comparison

#	Country	Field	Host facility	Application	DCC	check	check	check	check
1	Angola	West Africa 1	FPSO	Riser	0,02	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
2	Angola	West Africa 2	FPSO	Riser	0,0619	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
3	Angola	West Africa 3	FPSO	Jumper	0,0498	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
4	Angola	West Africa 4	FPSO	Jumper	0,0778	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
5	Angola	West Africa 5	FPSO	Riser	0,0778	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
6	Angola	West Africa 6	FPSO	Jumper	0,1253	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
7	Angola	West Africa 7	FPSO	Riser	0,153	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
8	Ghana	West Africa 8	FPSO	Test sample	0,0644	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
9	Eq. Guinea	West Africa 9	FPSO	Riser	0,0712	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
10	Norway	North Sea 1	FPSO	Riser	0,0338	Min<DCC	Median<DCC	Q3>DCC	Max>DCC
11	Norway	North Sea 2	FPSO	Riser	0,0432	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
12	Norway	North Sea 3	FPSO	Test sample	0,0581	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
13	Norway	North Sea 4	FPSO	Riser	0,0729	Min>DCC	Median>DCC	Q3>DCC	Max>DCC
14	Norway	North Sea 5	Semi-submersible	Riser		Min>DCC	Median>DCC	Q3>DCC	Max>DCC
15	Norway	North Sea 6	Semi-submersible	Riser		Min>DCC	Median>DCC	Q3>DCC	Max>DCC
16	Norway	North Sea 7	Semi-submersible	Test sample	0,0927	Min<DCC	Median>DCC	Q3>DCC	Max>DCC
17	UK	North Sea 8	Fixed	Riser	0,0154	Min>DCC	Median>DCC	Q3>DCC	Max>DCC
18	UK	North Sea 9	FPSO	Riser	0,0574	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
19	UK	North Sea 10	FPSO	Test sample	0,0827	Min<DCC	Median>DCC	Q3>DCC	Max>DCC
20	UK	North Sea 11	Fixed	Riser	0,0553	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
21	UK	North Sea 12	FPSO	Riser		Min>DCC	Median>DCC	Q3>DCC	Max>DCC
22	USA	GoM1	FPSO	Jumper	0,0248	Min<DCC	Median>DCC	Q3>DCC	Max>DCC
23	USA	GoM2	Semi-submersible	Riser	0,0607	Min<DCC	Median<DCC	Q3<DCC	Max<DCC
24	USA	GoM3	Spar	Riser	0,0649	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
25	USA	GoM4	Spar	Riser	0,0951	Min<DCC	Median<DCC	Q3<DCC	Max>DCC
26	USA	GoM5	Semi-submersible	Test sample	0,1103	Min<DCC	Median<DCC	Q3<DCC	Max<DCC

For high dynamism environment of Norway and the UK (green area of table) minimum values of curvature variations used for the flexible pipe design are noticed to be higher than its critical curvature, shown in red, Table 4.3. Therefore, it is advised to carry a deeper investigation of the critical curvature design premises, when designing a flexible pipe in UK/Norway environments. Most of the pipes are designed for curvatures higher than critical.

In the attempt to predict the fatigue performance of the pressure vault of high pressure pipes, designed in the course of this project, the tensile stresses seen on the pipes in-service were compared to the pipe operating pressures, see Figure 4.18.

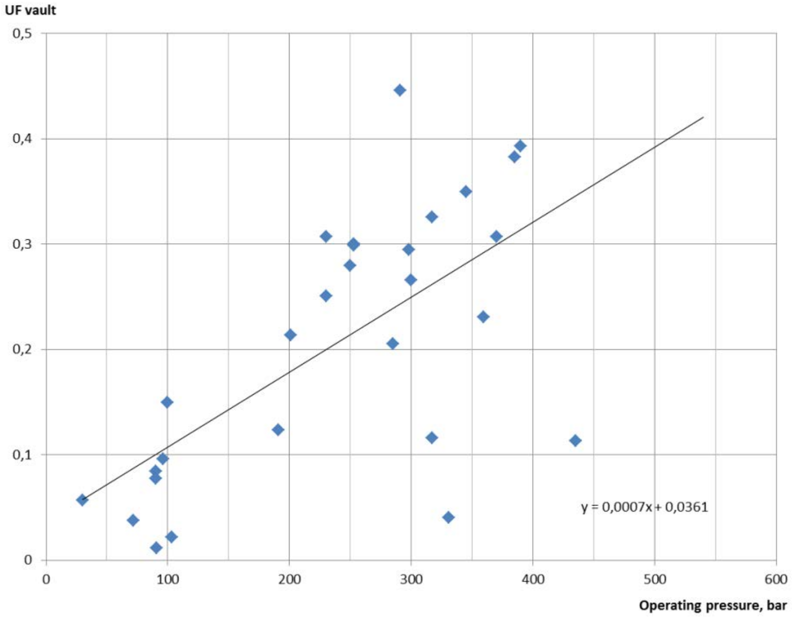


Figure 4.5 - Operating pressures versus UF of vault layer

As seen from Figure 4.18 above, UF on the vault is strongly dependent on the pipe internal pressure; the higher the operating pressures for the flexibles, the more stresses appear on Zeta. Therefore, the idea of a fatigue utilization factor can become reasonable.
Role of the operating pressure in the fatigue behavior prediction will be described in the next section.

4.6.3 Local fatigue analysis results

Local fatigue calculations were performed for the high pressure pipes, built in the course of this study, refer to CHAPTER 3 . Fatigue-in-air results for sour dynamic applications are presented on the Figure 4.19.

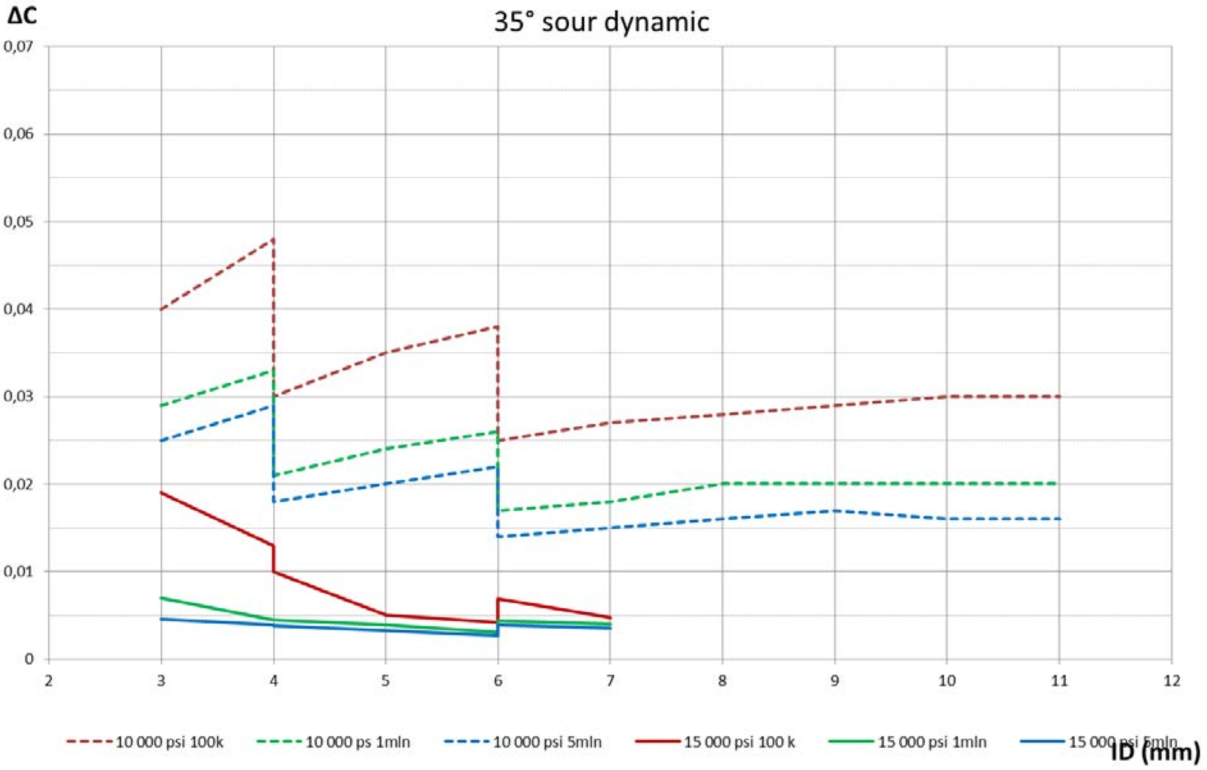


Figure 4.19 - Critical curvature versus pipe diameter

Fatigue acceptance domains were built for operating pressures of 10, 15 kpsi and different numbers of wave cycles (20 years of service life). No effect of axial tension was considered. Curves represent the fatigue damage of 0.1. The more the operating pressure of the pipe and the more cycles a pipe see during its service life, the less curvature variation this pipe will be able to cope with (for the same level of damage).

By assigning characteristic loadings for the selected geography, one can obtain the applicability of the HPHT pipes within a region. Single wave class and statistically averaged critical curvature values can therefore be used.

Based on the 75 per cent quantile data the following mean values were selected to represent the fatigue environment of West Africa 1: 100 000 cycles and 0.027 m^{-1} of curvature variation (fatigue in air).

The applicability domain for HPHT flexibles is presented in the Figure 4.20 (orange horizontal line).

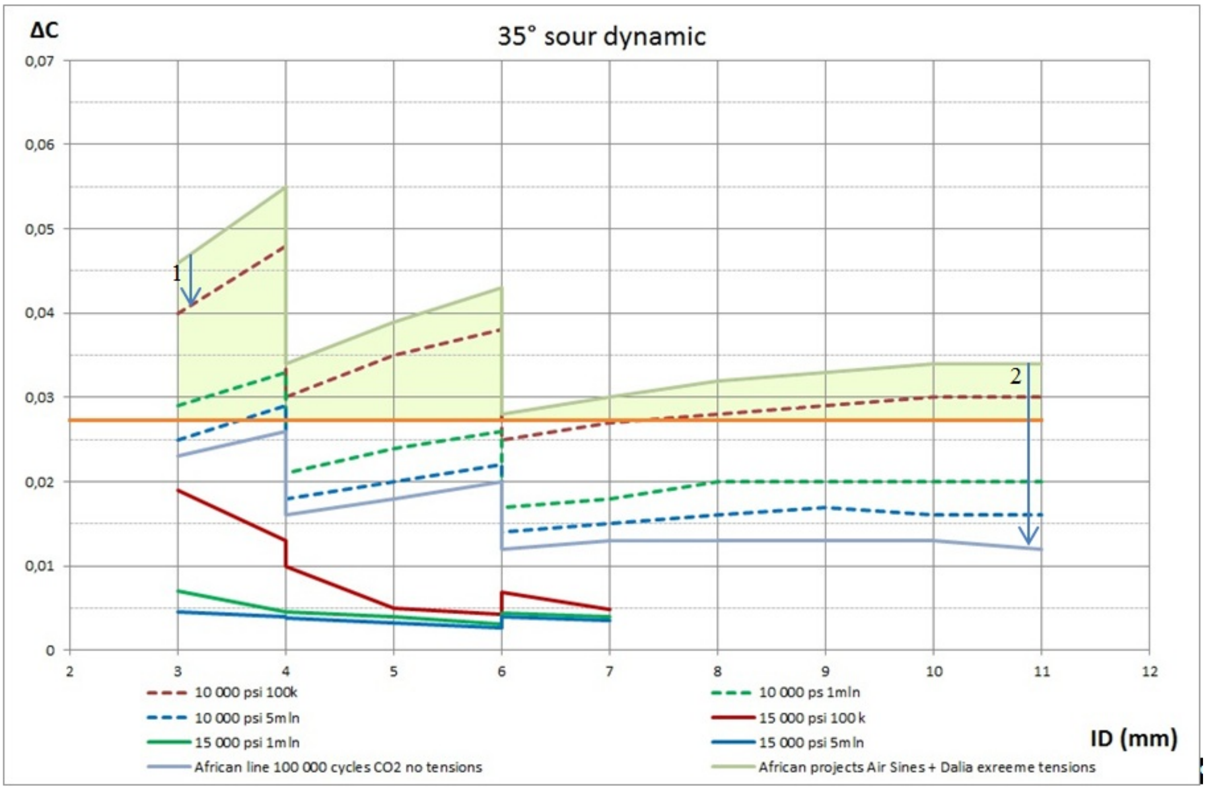


Figure 4.20 - Applicability domain for African projects

Sandy dunes/domains (on the graph above) represents the fatigue margins HPHT pipes have over the "mean" fatigue loadings in West Africa. However, the following are calculated for extreme top section tensions of West Africa project (250 tons). If no tensions applied the applicability domain automatically reduces (step 1; under red dashed line on the picture). Tension counteracts with internal pressure and decreases the wear of the pressure vault. The applicability domain for the HPHT pipes is significantly reduced, if calculated in CO_2 environment (1 bar of partial pressure in the pipe annulus), step 2.

Fatigue-in-air is the main design consideration for the flexible pipe during its service life (if no other requirements stipulated).

This approach of fatigue behavior prediction can be therefore applied to assess the applicability of a chosen flexible pipe solution.

To find another approach, the service life calculations were run for 26 projects selected earlier. The calculated damage is generally far below required 0.1.

The main results of the fatigue damage calculations can be simplify as:

1. The higher the operating pressure, the higher the accumulated damage in the vault layer is;
2. The lower the operating pressure, the higher is the Max DP/OP ratio is.

where Max DP/OP – is the ratio of maximum design pressure to a pipe operating pressure.

Based on these conclusions one can try to establish the prediction of the fatigue performance from the operating pressure.

However, to account for tensions effects onto a the system it is advised to account not only for an operating pressure, but for the contact pressure onto a Zeta layer;P_c- mean contact pressure between the Zeta spiral and first flat spiral.

Based on the data given for flexibles throughout the world, the following indicator (fatigue usage factor) of the fatigue performance can be set:
For the UF on Zeta <0.1,

Then; $$\frac{MaxDP}{OP} = 5.4 . \quad \text{Eq. 4.3}$$

Mean curvature variation is below 0.03 m-1.
Operating pressures are below 200 bars.
For the UF equal or above 0.1,

Then; $$\frac{MaxDP}{OP} = 1.7 . \quad \text{Eq. 4.4}$$

Mean curvature variation is below 0.05 m-1.
Operating pressures are above 200 bars.
The equations are based on the data from 26 projects in different geographical locations (different number of cycles).
Therefore, by knowing the maximum design pressure of the flexibles, a design engineer can pre-estimate the operating pressure range, that a given pipe can hold for a chosen fatigue environment.

4.7 Overview of the chapter

These are the main subjects discussed in the Chapter 4:

1. Importance of a fatigue analysis for the high pressure flexible pipe;
2. Fatigue analysis process;
3. Fatigue qualification testing;
4. Fatigue failure modes (pressure and tensile armours);
5. Governing standards and software;
6. Fatigue data analysis;
7. Two novel fatigue prediction indicators (mean loadings and $Max\ DP/OP$) ratios).

CHAPTER 5 CONCLUSIONS

5.1 Conclusions of the study

In this study the following challenges were addressed, such as:

1. Possibilities to design/manufacture a flexible pipe for design pressures above 20 000psi;
2. Definition of pressure limiting factors;
3. Prediction of operating pressures for flexible pipes by assessing their fatigue behavior;
4. Simplified methodology for early fatigue severity indicator.

However, the following key assumptions have to be respected:

1. Free hanging riser configuration is preferred for the study (simplified solution);
2. Fatigue assessment was performed for the top section of the riser(accounts for most tensions);
3. Fatigue in air was considered (base case failure mode);
4. Current manufacturing feasibilities limit the production of such pipes (above 18 inch);
5. Proposed fatigue methodology is a first estimate of the fatigue performance.

Possible sources of errors in the study are:

1. Changes due to an update of SN curves and Haigh diagrams;
2. Different methodologies used to adapt to specific client requirements;
3. Flexible pipes were not divided by the application (riser, jumper) in order to have a global perspective on their behavior.

The main results of the study are:

1. Research work with the software to design the high pressure pipes;
2. Presentation of the results (pressure capability maps);
3. Estimation of the applicable water depths;
4. Analysis of fatigue behavior of high pressure flexible pipes around the world;
5. Development of the characteristic fatigue loads to represent different geographical areas;
6. Development of fatigue behavior indicators (to predict the operating pressure).

5.2 Recommendation for further work

In order to further advance in the subject the following areas might be elaborated on:

1. Development of single wave class indicators for the regions;
2. Access fatigue performance at the TDP and other fatigue critical areas;
3. Use the critical curvature criteria in the preliminary fatigue assessment;
4. Develop an early fatigue severity indicator for tensile armours;
5. Fatigue full scale testing of a dual spiral layer prototype and a consecutive interpretation of the results.

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