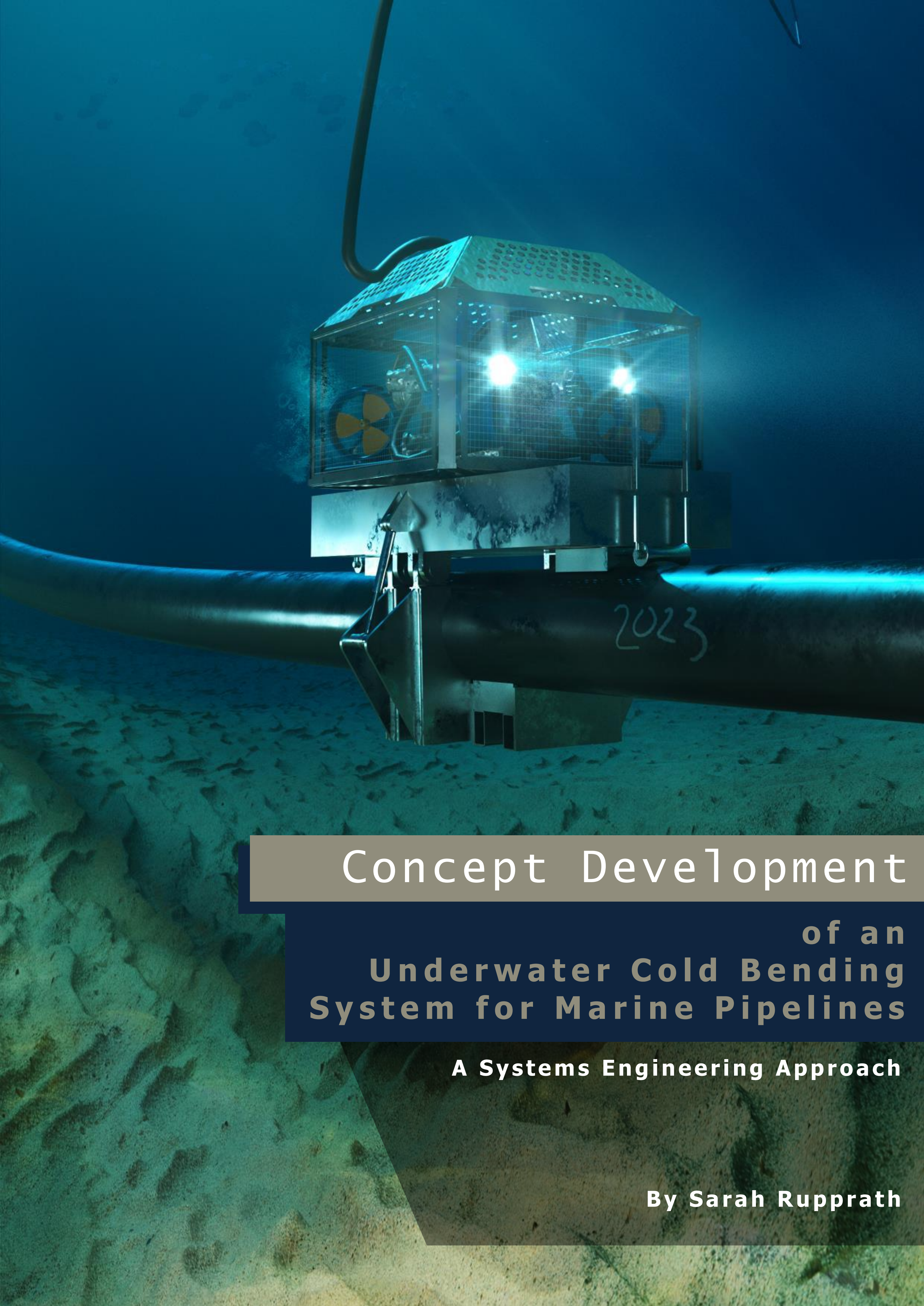

Concept Development

of an Underwater Cold Bending System for Marine Pipelines

A Systems Engineering Approach

By Sarah Rupprath

CONCEPT DEVELOPMENT OF AN UNDERWATER COLD BENDING SYSTEM FOR MARINE PIPELINES

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By

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ABSTRACT

One of the key challenges for offshore pipeline installation are free spans, which are sections not supported by the seabed. Within these sections the stress in the pipeline is increased due to local buckling which is caused by high bending moments at free span shoulders and midspan. Additionally, hydrodynamic loading due to currents and waves, and vortex induced vibrations increase fatigue damage.

There are many solutions to mitigate free spans and their negative effects, like building supports underneath the pipeline, or reducing the span length by burying a section of the pipeline in the seabed at the shoulder. From a deficiency analysis it is concluded that there is a need for a new system concept especially for steep slopes. This new free span mitigation shall be suitable to be performed by Allseas, an offshore construction company, during pipeline installation with the S-lay method, without the need for subcontractors like dredging companies.

Oil and Gas pipelines which are installed onshore do not create free spans as they follow the topography by being bent. This makes route intervention less extensive as it is done offshore.

The standard DNV-ST-F101 which contains requirements, principles and acceptance criteria for submarine pipeline systems, allows for cold field bends for submarine pipelines as long as certain requirements are met. There are many ideas published to bend the submarine pipeline such that it follows the seabed topography like it is the case for onshore pipelines. These approaches have been either patented or presented as case studies but have not been used in projects, so far.

It has been estimated in some of these previous case studies that the bending moments at the shoulders are reduced and stresses in the bend are within an acceptable range. From the presented opportunities and the described need, it is concluded that it is reasonable to develop a feasible new system concept which satisfies the operational and functional requirements defined in this thesis.

From combining different solutions of common subfunctions a number of concepts have been found which have been analysed and narrowed down to one possible most promising design. Using this new tool which is used in combination with an AUXROV it is possible to bend one 12m joint of the size of 32” by $18.5°$. With the given parameters of this design the free span length and height are narrowed down and the pipeline follows the topography when bent. It is verified that the bending moment at the free span shoulder is indeed reduced as presented in the literature.

The concept design, presented in this thesis, is at an early development stage but can serve as basis for detailed engineering, the next step in concept development. With a tool like this a new opportunity as standard solution for free span mitigation at steep slopes is introduced.

AKNOWLEDGEMENTS

Success can be measured according to different criteria. It can be the salary, the amount of responsibility, reaching goals or gain experience. For me it all starts with the following quote:

"To be successful, the first thing to do is fall in love with your work." - Mary Lauretta

To fall in love with my work would not have been possible without the people who have been part of this journey of writing my thesis. This is why I would like to take the chance to extend my deepest gratitude for their invaluable input.

First I would like to express my deepest appreciation to Ir. Thomas Dronkers who found this thesis topic and for enabling me to write this thesis with him as supervisor in cooperation with Allseas. After nine month of working on this topic I still find it very interesting and enjoy discussions about possible optimizations and reasoning why the developed concept looks now the way like it is presented below. Thank you for encouraging me to contact different colleagues within the company to pick their brains on new points of views and expertise in specific fields.

To all those colleagues special thanks for being open minded about new ideas and contributing to a realistic concept design.

I could not have undertaken this journey without my supervising professor Dr. Austin A. Kana who took the time besides his busy schedule to proofread all the chapters and let me not lose focus between all the interesting ideas and subjects I came across during research. Thank you for being so engaged in the process and for your consistent valuable feedback which greatly improved my thesis.

Further, I would like to thank Bernard Elsman for creating the great animation of the final design of the system concept on the pipeline which became the title page of this report.

Last but not least, I would like to express my appreciation for my family for their support and encouragement throughout my academic journey. Your believe in me motivated me to find the work that I love.

With this great experience within Allseas I am looking forward to my first job as engineer, working with the colleagues who I already got to know and those which I will get to know in the future.

Sarah
Delft, November 2023

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LIST OF FIGURES

LIST OF ABBREVIATIONS

Abbreviation	Meaning
40D	40 times the pipe outer diameter
AHC	Active heave compensation
AR	As received
AUXROV	Auxiliary ROV system
CB	Centre of buoyancy
CG	Centre of gravity
FEM	Finite element method
HPU	Hydraulic power unit
HT	Heat treated
IHPU	Isolated hydraulic power unit
LARS	Launch and recovery system
MFE	Mass flow excavator
PE	Polyethylene
PLET	Pipeline end termination
PP	Polypropylene
R&D	Research and development
ROV	Remote operated vehicle
SPC	Special purpose crane
TRL	Technology readiness level
TSM	Teaser management system
VIV	Vortex induced vibrations
WROV	Work class ROV

LIST OF SYMBOLS

Symbol	Meaning
M	Bending moment
S	Effective axial force
p_e	External pressure
p_{min}	Minimum internal pressure
p_c	Characteristic collapse pressure
γ	Safety factor
ρ_w	Water density
OD	Outer pipe diameter
U	Instantaneous (time dependent) flow velocity
y	Pipe lateral displacement
C_D, C_M and C_a	Drag-, inertia- and added mass coefficients
Z	Bending modulus of circular hollow section
S_P	Stress due to internal pressure
S_T	Stress due to temperature (only considered under restrained conditions)
S_X	Axial force applied on the pipe
S_B	Stress due to bending moment (different equations for straight pipes and pipe bends)
M_t	Torsional Moment
M_i	In-plane bending moment
M_o	Out-of-plane bending moment
i	Stress intensification factor
M_R	Enhanced bending moment
D	Hydrodynamic diameter
V	Characteristic current induced flow
$V(z_r)$	Current velocity at distance z_r above the seabed
Z_{0a}	Effective seabed roughness
Z_r	Reference measure height above the seabed
Θ_c	Relative angle between pipeline and current direction
W	Elastic bending resistance
σ_T / SMYS	Maximum tensile stress = 458.58MPa
s_{ext}	Extra percentage due to higher specifications than listed = 20%
I	Inertia of the cross-section
t	Wall thickness
E	Young's modulus
k	Curvature of the bend
R or r	Bending radius = 246m in sagbend
h	Stroke of hydraulic cylinder
α	Angle of the circle segment depending on the leaver arm
F_r	Force of the rack
Tp	Torque pinion
r_p	Radius of the pinion
P	Pressure
Q	Flow rate
A	Area
F	Applied Force
hp	Horse power
L	Length
$t[s]$	Time
c	Arc length

DEFINITIONS

Commonly used terms for parts and angles of the pipeline for the S-lay method of pipeline installation are illustrated in the schematic below.

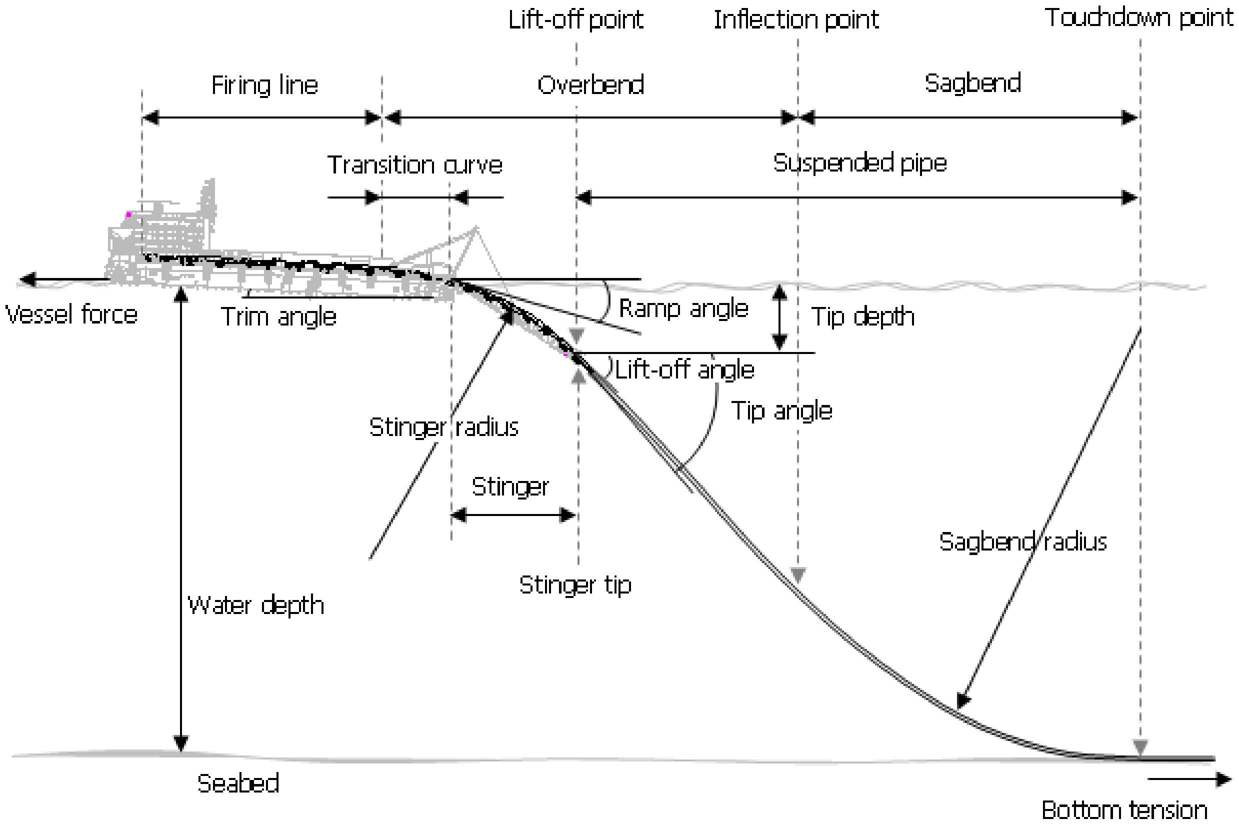


Figure 1: Schematic Side View of Terms Used for Parts and Angles of the Pipeline [1]

1 INTRODUCTION

The main idea of this thesis is the concept development of an underwater cold bending system for marine pipelines. In this report it is described in detail why it is interesting to investigate this topic and which approaches and analyses have been performed to create a feasible design of a tool which is suitable perform this function.

This master thesis is written in cooperation with the company Allseas. The company specializes in heavy lifting, deep-ocean polymetallic nodule collection, river waste collection and in offshore pipeline installation. These projects are executed by a versatile fleet with specialized equipment to increase efficiency.

Within Allseas the S-lay method of pipeline installation (Figure 1.1) is the preferred method for which a bending system of marine pipelines should be compatible. In the S-lay configuration, 12m long pipe segments are welded together onboard a specialized installation vessel to one straight, continuous pipe. Once another section is welded to the continuous pipe the coating covering the weld is completed and the pipeline is guided through tensioners and over the stinger to the seabed. Because bent sections of pipe do not fit through the tensioners, which are needed to hold the end of the pipeline onboard the installation vessel, the bending of the pipe should be done after the last tensioner.

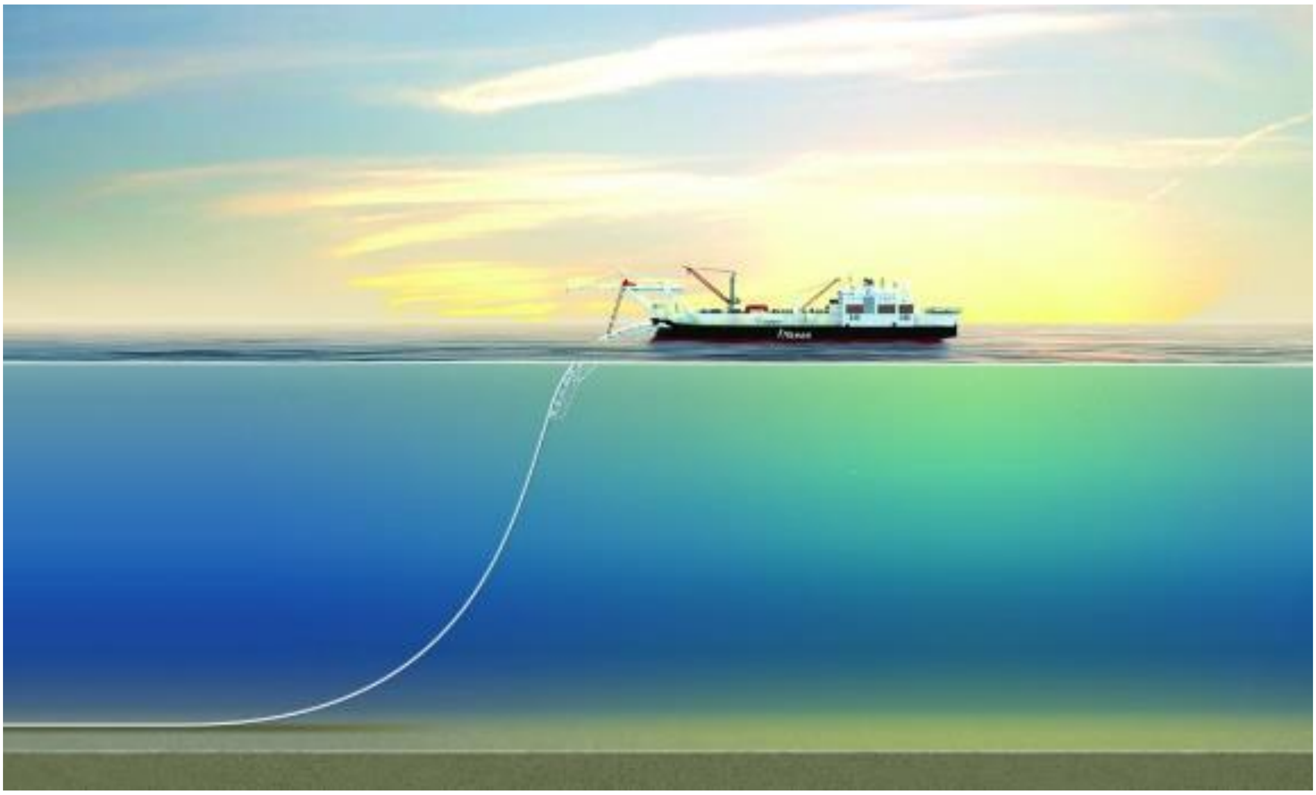


Figure 1.1: S-Lay Method Pipeline Installation [2]

Due to an uneven seafloor, one of the main challenges during offshore pipeline installation is to avoid free spans. Free spans, sections which are not supported by the seabed, occur where the straight pipeline has to bridge a valley on the seafloor or a sharp slope transition as shown in Figure 1.2.

A free span can lead to damage of the pipeline. This failure can be local buckling due to excessive stress in the pipe at the free span shoulders or at midspan, where the moment is increased due to for example self-weight as it can be seen in Figure 1.2. Another possible failure scenario related to free spans can be fatigue due to hydrodynamic loading or vortex induced vibrations.

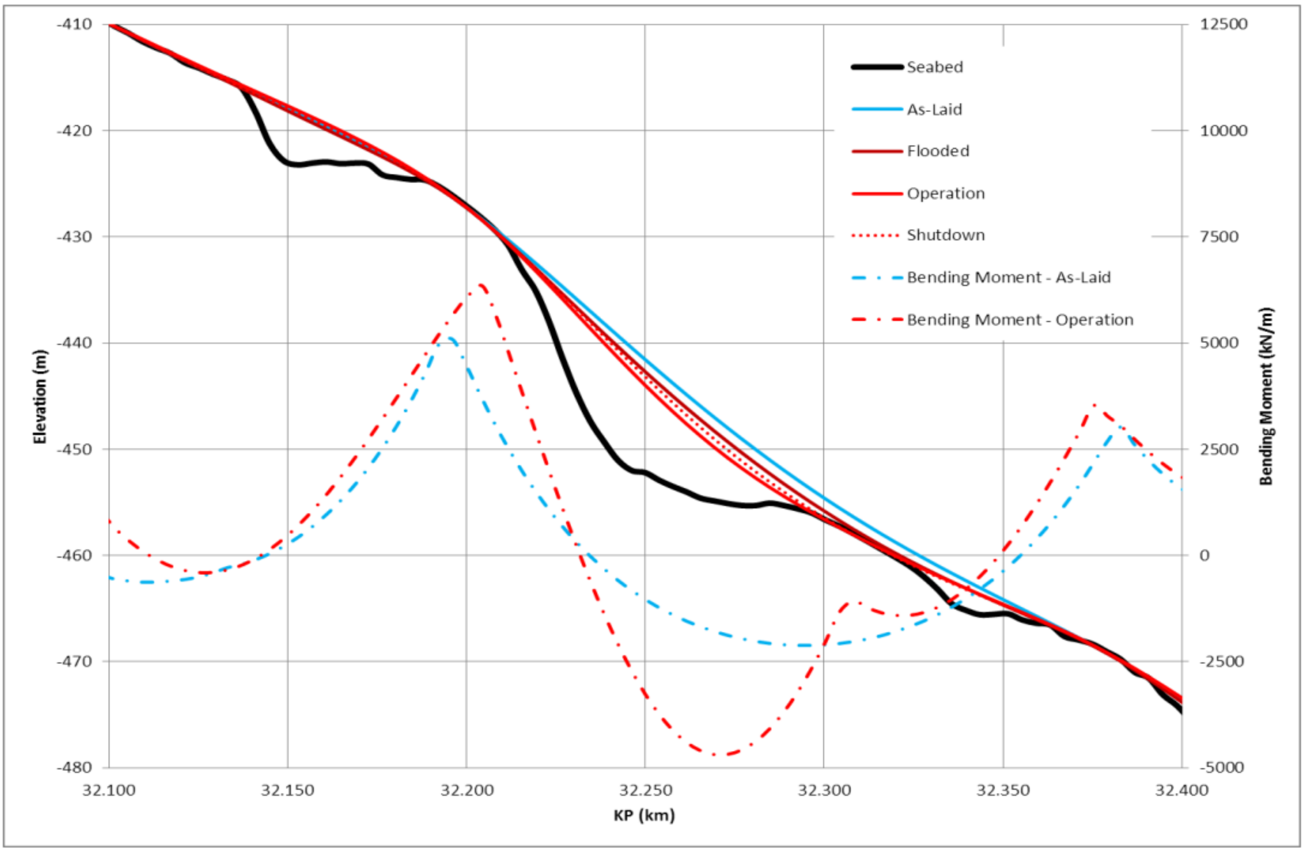


Figure 1.2: Example of Bending Moments in a Pipeline over a Free Span [3]

There are mitigation measures available for this problem. A first measure is to install support structures underneath the pipeline. A second is to use dredging to shape the edge of the slope. This creates a smother morphology and therefore reduces free spans as well. Both methods lead to a modified seabed but create a large environmental impact and high costs.

Onshore pipelines, on the other hand, are installed following the terrain. This means that the pipeline can be bent at the site or prebent parts are welded to the line pipe to ensure that the pipeline does not experience free spans.

The overall objective of this thesis is to investigate the possibility of bending marine pipelines as free span mitigation and to develop a new concept of an underwater cold bending system for marine pipelines so that instead of modifying the seabed the pipeline follows the topography. The bending procedure is restricted to cold bending as field bends are only permitted as colds bends according to the DNVGL-ST-F101 standard. To avoid interaction of the bending system with the motions of the ship the new system shall be developed such that it is placed under water. This system concept shall serve as basis for the next phase Engineering Development which is succeeding the Concept Development phase according to the systems engineering approach. In the systems engineering approach which is followed for this thesis, the Concept Development stage, the first stage in a systems lifecycle, is divided into three steps as it can be seen in Figure 1.3.

	Phase					
	Concept development			*Engineering development*		
Level	Needs analysis	Concept exploration	Concept definition	Advanced development	Engineering design	Integration and evaluation
System	Define system capabilities and effectiveness	Identify, explore, and synthesize concepts	Define selected concept with specifications	Validate concept		Test and evaluate
Subsystem		Define requirements and ensure feasibility	Define functional and physical architecture	Validate subsystems		Integrate and test
Component			Allocate functions to components	Define specifications	Design and test	Integrate and test
Subcomponent		Visualize		Allocate functions to subcomponents	Design	
Part					Make or buy	

Figure 1.3: Status of System Materialisation of Concept Definition Phase [4]

- Needs Analysis

The first level of the concept development is the Needs Analysis which is performed in a literature review. Within this level the question why a new system concept is needed should be answered. When the needs are understood operational requirements are defined. These requirements describe what the new system is designed to do, and the overall objectives of the system. [4]

- Concept Exploration

The next phase of a new system concept development is the Concept Exploration phase. In this phase a variety of system concepts are developed based on the operational requirements which have been defined at the end of the Needs Analysis phase. From these operational requirements a set of high level functions and a set of functional requirements are derived which all alternatives of the new system concept should satisfy. The presented underwater pipe bending methods as well as onshore pipe bending machines are used as starting point for the development of different system concepts. It should be ensured that all suggested system concepts are feasible. [4]

- Concept Definition

The most preferable concept of the ones found in the previous stage is selected based on predefined performance and compatibility criteria with a trade-off analysis. The selected concepts are then specified by defining the functional and physical architecture, and allocating functions to components of the system concept. Further it can be explained how the new system concept operates and how it impacts normal pipeline installation. [4]

To achieve the thesis objective, five research questions have been defined:

a) What is the state of the art in pipeline bending onshore as well as offshore and what is needed to address current limitations? (Needs Analysis)
b) What are the operational requirements of the new system concept? (Needs Analysis)
c) What are alternative new concepts for an underwater cold bending system for marine pipelines and what are their common functions and parameters which are described as functional requirements in the systems engineering approach? (Concept Exploration)
d) Which of these alternative concepts is the most promising one based on predefined criteria and how does the systems architecture look like, for example preliminary sizing of components of the subsea bending system? (Concept Definition)
e) How does the new system concept impact normal pipeline installation? (Concept Definition)

Needs Analysis

2
PROBLEM BACKGROUND

Each pipeline project goes through different phases in which economic as well as technical factors are analysed. The aim is to optimize the transportation of the product, oil or gas, from the reservoir to the desired destination which can be for example processing plants or distribution centres [5]. Free span management starts at the early project phase. Throughout the pre-engineering survey, environmental data is gathered including seabed topography, waves and currents. The detailed "design phase is the core of the free span integrity management" [6]. During construction free span mitigation measures are installed where applicable and after commissioning free spans are monitored, according to a maintenance plan, to ensure the integrity of the pipeline where the seabed is mobile or accidental loads can damage the pipeline [6].

In the Needs Analysis presented in the following four chapters of the report, the focus is laid first on the detailed design phase to describe how the pipeline integrity is assessed. In the following, measures which can be undertaken during the construction phase to mitigate free spans are discussed. In the detailed design phase different analysis and surveys are performed. As the results of the different analysis influence the other analysis the whole pipeline design is an iterative process until the optimal configuration is achieved. The different assessments include [5]

- Route selection and survey,
- Environmental impact and risk assessment,
- Corrosion and material selection,
- Coating selection,
- Hydraulic analysis and line sizing,
- Mechanical pipeline design.

Free spans are assessed during the mechanical pipeline design, after deciding on "diameter, material, wall thickness, potential trenching, and coating for weight and insulation, any global buckling design and release of effective axial force" [7]. Considerations which shall be taken into account to estimate these pipeline parameters in the mechanical pipeline design of the detailed design phase include the following [5]:

- Internal pressure
- External pressure
- Construction loads
- Stability
- Expansion stresses
- External damages
- Free spans

According to DVNGL-ST-F101, a standard for submarine pipeline systems, free span rectification is required for all spans exceeding the specified acceptable length or height for the specific location [8]. What free spans are, why they are a problem, and what can be done to mitigate these, is presented in this report.

2.1 Free Spans Offshore

Free spans are sections of underwater pipelines that are unsupported by the seabed [9]. This happens because "oil and gas pipelines have large diameters and are not very flexible" such that they cannot follow the seabed topography [10].

In the DNGL-RP-F105 the following definitions are given for a free span which are used in this report respectively (Figure 2.1a). "The gap, *e*, is the distance between the bottom of the pipe and the seabed as a function of the pipe position *x*. The free span length, *L*, is defined as the length of a continuous section with positive gap, *e(x) > 0*. Sections of continuous support on either side of a free span, where *e(x) = 0*, are defined as span shoulders, with lengths *Lsh*." [7]

There are two different span scenarios considered in DNV-RP-F105 [7]: The isolated single span as it can be seen in Figure 2.1a and interacting multi-span where the free spans are located close, such that the vibrations of one influences the behaviour of the neighbouring span (Figure 2.b).

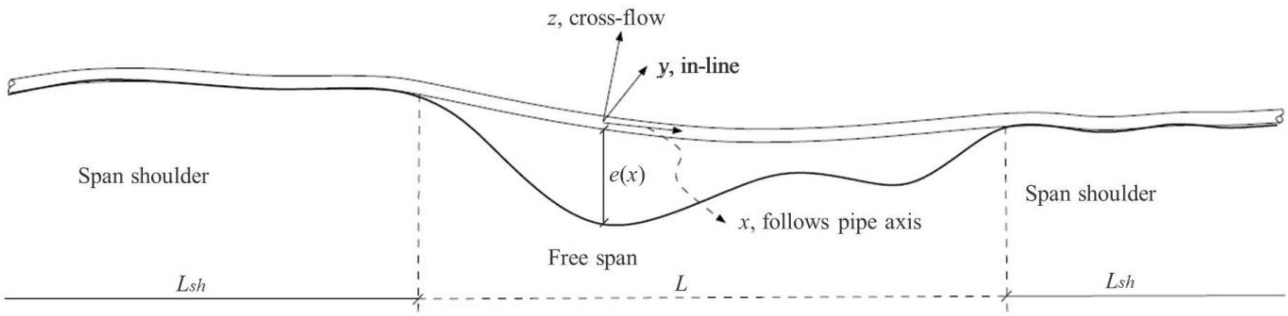


Figure 2.1a: Isolated Single Free Span DNV-RP-F105 [7]

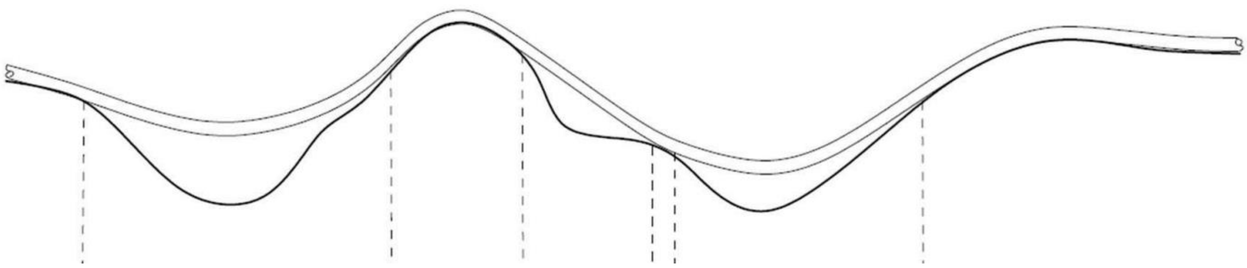


Figure 2.1b: Interacting Multi-Span DNV-RP-F105 [7]

2.1.1 Free Span Causes

According to DNVGL-RP-F105 pipeline free spans can be caused by [7]**:**

- **Seabed unevenness**

Seabed unevenness includes outcrops of rocks or coral reefs and pockmarks which are craters in the seabed. Pockmarks are caused by fluid erupting the sediment. They can become as big as 700m in diameter and 15m deep. [11]

A specific area in which free spans can occur is at the transition zone between the continental and oceanic crust. As it can be seen in Figure 2.2 the seabed experiences an abrupt slope change at the shelf break where the generally very gentle slope of the continental shelf (less than 1°) transitions to a much steeper angle, at the so called continental slope [12]. At some locations the angle of the continental slope can be between 30° and 40° [13]. Continental shelf breaks can be found typically in water depth between 100m and 200m [13]. Figure 2.3 shows a free spanning pipeline at an escarpment similar to a continental shelf break.

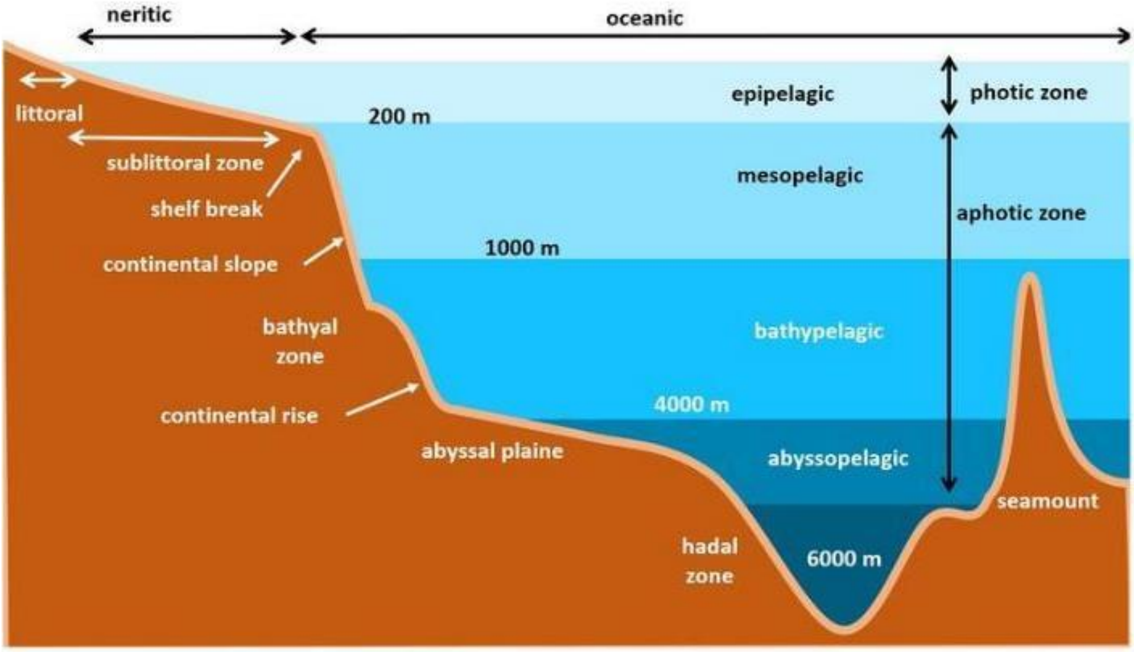


Figure 2.2: Ocean Zones [12]

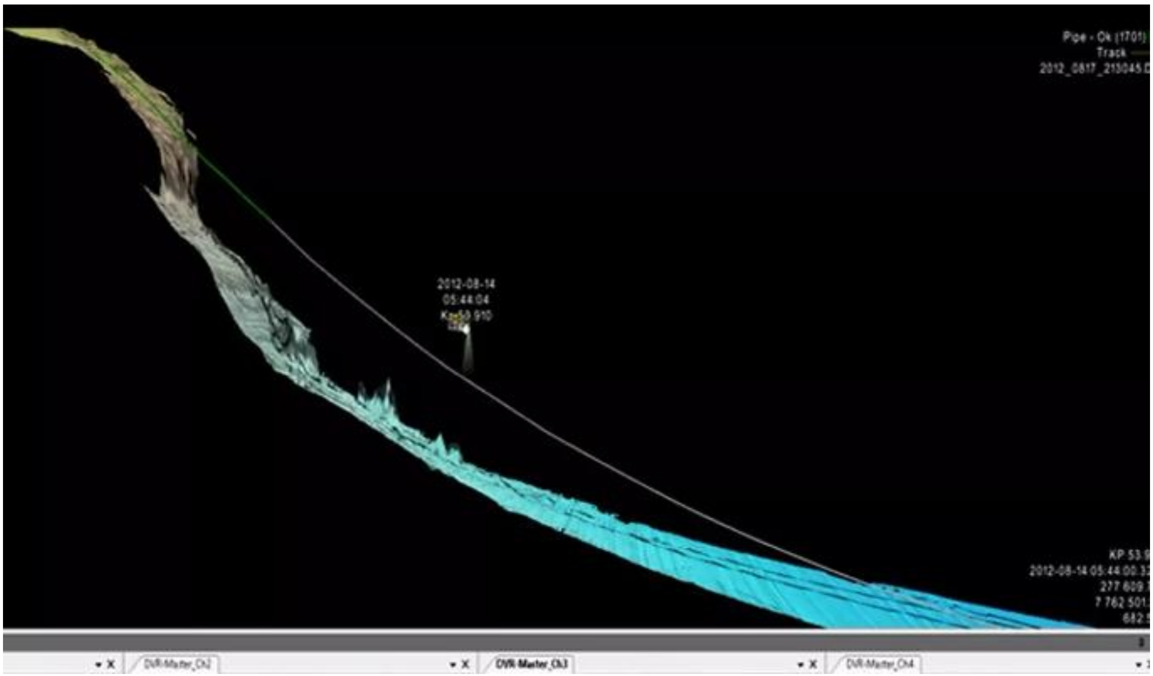


Figure 2.3: Free Span at Enscarpment [14]

- **Change of seabed topology** , e.g. scouring, sand waves

Free spans can also develop over time when the topography changes due to for example sand waves. Sand waves look a bit like water waves and occur due to relatively high current velocity where the seabed consists of sand or gravel. They can cause free spans as they form hills which can be as high as 20m. Sand waves can be stationary but also in motion. When in motion, they can cause a free span over time where the pipeline was initially supported by the seafloor. [11]

Another phenomenon which changes the topography over time and causes free spans is the effect of scour, as shown in Figure 2.4. A flow is created in the soil underneath the pipeline due to waves or currents which develops a gap between the pipe and the seabed. As it grows larger (Tunnel erosion) vortex shedding occurs which increases the erosion (Lee-wake erosion) until the gap is so large that an equilibrium is reached. [9]

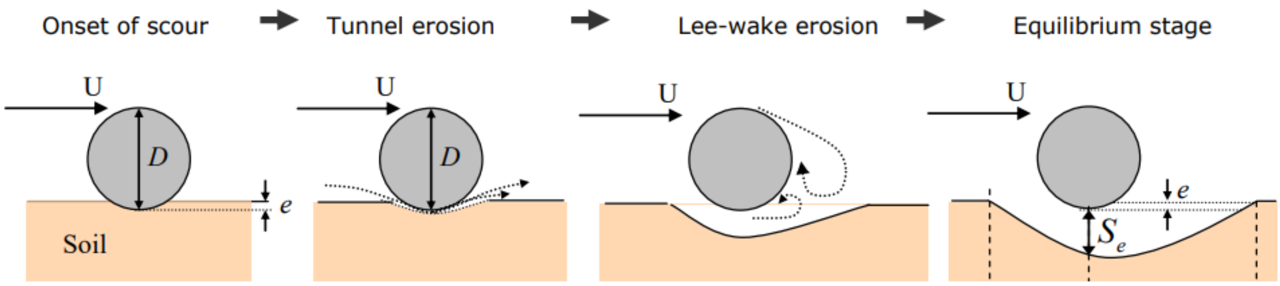


Figure 2.4: Onset of Scour Leading to Formation of Free Spans [9]

- **Artificial supports, crossings and end terminations**

Lastly free spans can be created artificially. One example is the free spans which occur at pipeline crossings (Figure 2.5a). The pipeline at the top is unsupported close to each side of the crossed pipeline. As it can be seen in Figure 2.5b a free span is also created at a pipeline end termination (PLET). On the left the pipeline is unsupported because the connection point, painted white, lays a little higher than the seabed.

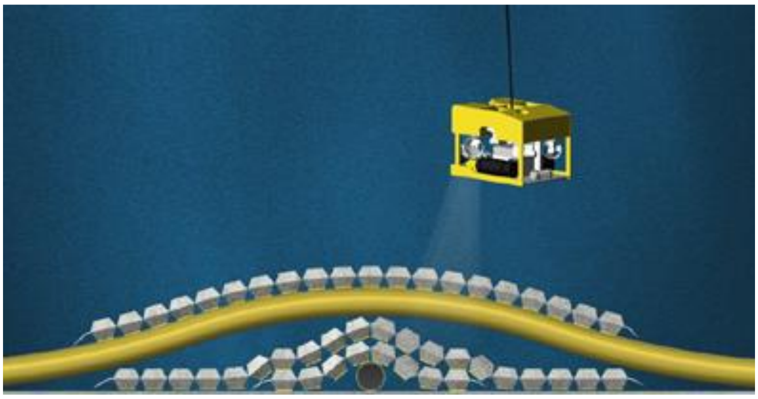


Figure 2.5a: Free Span at Pipeline Crossing [3]

Figure 2.5b: Free Spanning Pipeline at PLET Connection [15]

2.1.2 Failure Scenarios due to Free Spans

According to DNVGL-RP-F105 section 2.2.1 all temporary and permanent free spans shall be assessed to ensure the pipeline integrity with respect to local buckling and fatigue [7].

Local buckling is the failure mode where a short section of pipe experiences gross changes of pipe cross-section which can lead to for example a system collapse, localised wall wrinkling and kinking [8]. Local buckling occurs due to excessive bending at the free span shoulders or mid-span. It is induced by self-weight of the pipeline, internal loads like tension or expansion of the material, or external loads like trawl gear impact or anchor drag [3]. Figure 2.6a and Figure 2.6b show buckled pipe sections. The figure on the left shows a buckled pipe which is set under internal pressure after the buckling occurred. An example of this scenario is when the installed pipeline buckles, then taken into operation and filled with the product under operation pressure. The right side shows a buckle developing further under external pressure. This kind of buckling occurs in the as laid condition when the empty pipeline is exposed to the external pressure of the water column.

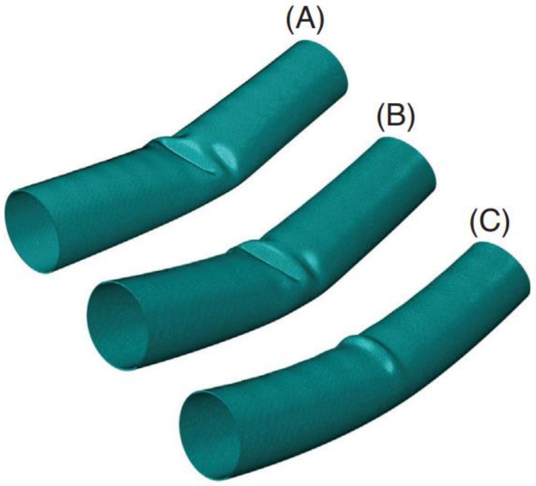


Figure 2.6a: Buckled Pipeline: A) no pressure, B) and C) internal pressure [16]

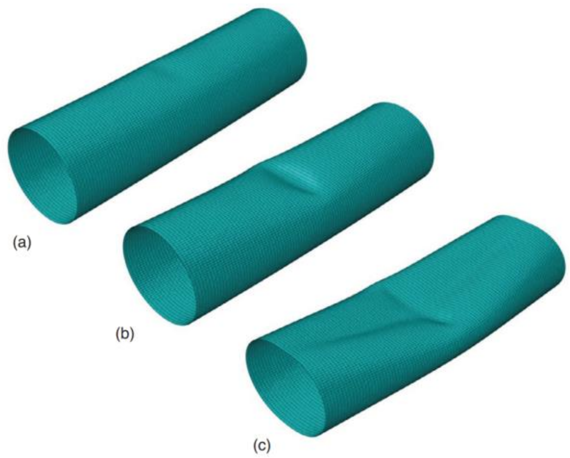


Figure 2.6b: Buckled Pipeline under External Pressure [16]

A local buckling check needs to be satisfied at all cross-sections for different scenarios during its design life which are the following [8]:

- During installation
- As laid, when the pipeline is empty
- Water filled
- System pressure test, when the pipeline is filled with water under a pressure exceeding the operating pressure. Safety factors to determine test pressure can be found in DNVGL-ST-F101 Table 5-8
- Operating, when the pipeline is filled with the product, oil or gas, and is at operating temperature (-33℃ to $+180$℃) and pressure (16 500 psi) [17]

This buckling check is satisfied when the relevant criterion is met. DNVGL-ST-F101 explains different criterions for different load scenarios which are applicable for the aforementioned stages of the design life of the pipeline. One example is a combined loading which considers the bending moment, the effective axial force and external overpressure for the scenario "as laid". Figure 4.18 shows examples of results for this buckling check. This criterion can be expressed according to equation 2.1 (equation 5.28 [8]):

$$\left[\gamma_m \cdot \gamma_{SC,LB} \cdot \frac{|M_{Sd}|}{\alpha_c \cdot M_p(t_2)} + \left(\gamma_m \cdot \gamma_{SC,LB} \cdot \frac{S_{Sd}(p_i)}{\alpha_c \cdot S_p(t_2)}\right)^2\right]^2 + \left[\gamma_m \cdot \gamma_{SC,LB} \cdot \frac{p_e - p_{min}}{p_c(t_2)}\right]^2 \leq 1 \qquad 2.1$$

Fatigue is the second possible failure that a pipeline can experience at a free span. It can be caused by hydrodynamic loading and/or vortex induced vibrations (VIV).

The hydrodynamic loading is the load the pipe experiences due to waves and currents. For slender structures this load can be calculated per unit length with the Morison equation 2.2 which is presented below. [7]

$$\begin{gathered} P(x,t) = \frac{1}{2}\rho_w OD\ C_D(U - \dot{y})|U - \dot{y}| + \frac{\pi}{4}\rho_w OD^2 C_M\ \dot{U} - C_a\frac{\pi}{4}\rho_w OD^2\ddot{y} \\ hydrodynamic\ loading = drag\ force + inertia\ force - added\ mass \end{gathered} \qquad 2.2$$

The velocity U is the instantaneous flow velocity of the surrounding water which includes the waves and currents. The pipe lateral displacement y describes the relative pipe motions which can be caused by for example vessel motions.

Vortex induced vibrations (VIV) are caused by a current flow around the pipeline [18]. The current flow is separated by the pipeline and vortices shed in the wake region which can be seen in Figure 2.7a. The vortices are shed alternating at either side of the pipeline. The vorticity of the vortices on one side is in clockwise direction (in Figure 2.7b named as vortex A and C) whereas the direction of the vortices on the other side is counter clockwise which are shown as vortex B. The vortex shedding causes the pipeline to vibrate at a certain frequency which can be close to one of the natural frequencies of the pipeline. If that is the case, resonance occurs which leads to large amplitudes of the vibration and the pipeline can experience fatigue damage. [19]

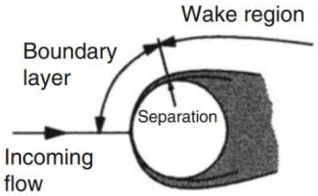


Figure 2.7a: Sketch of Flow Separation [19]

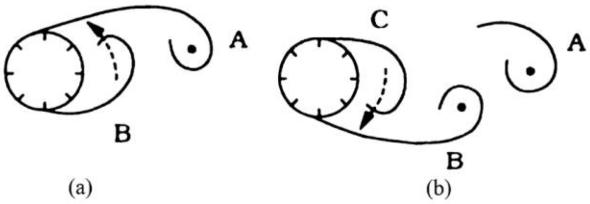


Figure 2.7b: Mechanism of Vortex Shedding [19]

2.2 Problem Background Onshore

Similar to the DNV standard which provides rules and regulations for subsea pipelines, there are standards for onshore pipelines as well. Which standard to follow, for the mechanical pipeline design, depends on the specific project and country. Widely accepted American design standards are the ASME B31.4 which is applicable for onshore pipelines transporting liquids and ASME 31.8 for gas pipelines [16]. A Dutch equivalent is the NEN 3650.

In the Dutch regulation it is defined that the equivalent stress from the load combination has to be smaller than the limiting stress for onshore free spans [20]. The stress from the load combination is calculated during the mechanical design phase when the pipeline diameter and wall thickness are determined. As burst of the steel pipeline is the major mode of failure, main focus lays on the pressure design which is applicable for the scenario of a pipeline during operation. Considered loading during this pressure design are transverse loading, axial compression, internal pressure and longitudinal stress. [16]

In the calculation of the longitudinal stress there is a distinction made for stresses due to bending moments between straight pipe sections and bent pipe sections as it can be seen in the equations below. The equation to calculate the longitudinal stress, according to ASME B31.8 reads [16]:

$$S_L = S_P + S_T + S_X + S_B \tag{2.3}$$

For straight pipe sections the stress due to bending moment can be calculated with the following equation:

$$S_B = \frac{M}{Z} \tag{2.4}$$

with

$$Z = \frac{\pi D_m^2 t}{4} \tag{2.5}$$

Whereas for stress due to bending moment in pipe bends the following equation is applicable:

$$S_B = \frac{M_R}{Z} \tag{2.6}$$

Where the "enhanced" bending Moment is

$$M_R = \sqrt{(0.75 i_i M_i)^2 + (0.75 i_o M_o)^2 + M_t^2} \tag{2.7}$$

It shall be noted that the equations for the stress calculations slightly differ depending on the standard used [16].

In the ASME B31.8 standard there are also information about requirements for field bends [16]. This is the case because onshore pipelines follow the topography in contrast to offshore pipelines as it can be seen in Figure 2.8a and Figure 2.8b below. This way free spans are avoided as the pipeline is continuously supported by soil.

If and where pipe bends are needed is influenced by the route selection. To ensure the pipeline integrity the routing of the pipeline should be such that the minimum allowable curvature which is specified in the structural design is not exceeded [20]. However, in certain areas the route selection is limited and the pipe route follows a difficult terrain where the minimum curvature of the elastic bend cannot always be maintained. These areas include [21]

- Densely populated areas
- Mountainous regions
- Nature protected zones.

Figure 2.8a: Onshore Pipeline Lowered into a Trench [16]

Figure 2.8b: Onshore Pipeline Installed Aboveground [22]

Where the pipeline cannot follow the route through elastic bends, when it is too stiff, cold formed bends or induction bends are used [20]. The idea of plastically bending the pipe developed historically parallel to the pipeline system itself (Figure 2.9). The question on how and if to bend the pipe as investigated in this thesis for submarine pipelines has been answered some time ago for onshore pipelines.

The first pipelines before 1887 were made out of wood or wrought iron. Since then steel pipes began to replace these. Over time the pipeline construction and fabrication methods developed rapidly.

First attempts of bending the pipeline "resulted in circumferential pipe deformation and wrinkles centred at the bending radius" [23]. This is why bends which are produced before 1950s are called wrinkle bends. Different processes which sometimes included heating the pipe section prior bending, resulted in a rage of quality of the wrinkle bend.

In the 1940s smooth bends developed due to the requirements for war emergency pipelines. Machines which could produce smooth bends on site included external bending shoes. In 1950s the first hydraulic bending machines are introduced.

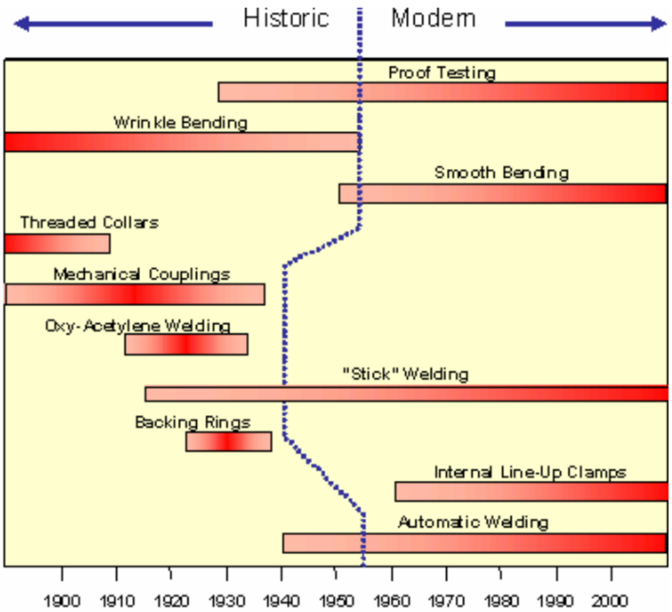


Figure 2.9: Historic Pipeline Construction Practices [23]

The so called miter bends were an alternative to bending the pipe. They were fabricated by cutting pipe sections under an angle and subsequently welding these together. Miter bends and hot field bends are prohibited in many regulations for oil and gas pipelines since late 1940s and early 1950s [23]. The Dutch standard NEN-3650-2 states that miter bends shall not be used for group 1 pipeline systems. Group 1 includes, pipelines transporting substances which are dangerous for humans like highly flammable petrol or natural gas with an operating pressure exceeding 1.6 MPa [24]. Miter bends can only be used for low pressure systems transporting substances which are not dangerous for humans (categorized as group 2) if a stress analysis is performed to prove the integrity of the bend [20]. NEN-EN 14870-1 prohibits hot bends unless they are followed by a full heat treatment [25]. The procedure of the heat treatment is described in Appendix B. As it is described there, performing a heat treatment at the building site is not practical.

The present solutions to construct the onshore pipeline such that it follows the topography are presented in the following section.

3
SOLUTIONS ONSHORE

Bent pipe sections can be fabricated at an off-site facility or they can be manufactured in the field [26]. The detailed phases of the pipeline installation onshore is presented in Appendix A. The overall aim is to ensure the integrity of the pipe whichever option is chosen. Possible failures which can be caused by bending are presented in Figure 3.1.

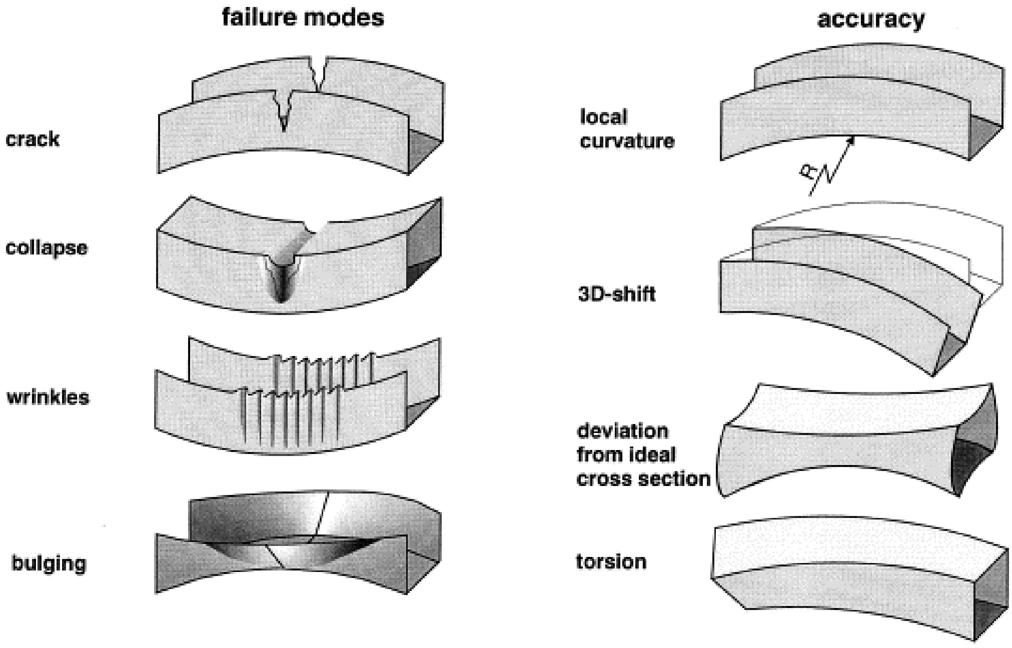


Figure 3.1: Possible Failures Caused by Bending [27]

3.1 Prefabricated Bends

There are different procedures to bend a pipeline which are used by different manufacturers depending on material, wall thickness, tolerances and bending radius [28]. These procedures can be performed as hot or cold bends. The difference between the procedures lays in the substantial step whether the pipeline is heated or remains at room temperature during bending. Cold bends are defined as all bends which are produced without heating the pipe. In the following two different cold bending procedures are presented.

1) Press bending

   Figure 3.2a shows the principle of press bending. A bending tool placed at the top on the pipe is pressed downward against two counter rollers. The counter rollers on the bottom are adjusted such that the desired bending radius is achieved. According to Franz Iten AG this procedure is suitable for thick walled tubes with large bending radius which might make it suitable for submarine pipelines as well. [28]

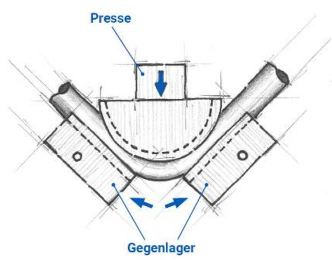


Figure 3.2a: Schematic Press Bending [28]

2) Three-roll bending

A schematic of three-roll bending is presented in Figure 3.2b. Similar to press bending there are two stationary counter rolls and the work roll presses down toward the middle of the stationary rolls. In contrast to press bending the rolls rotate and the process might be repeated until the desired radius is achieved. [28]

This procedure is adapted to submarine pipelines which are installed with the Reel-lay method. Detailed explanation is given in chapter 4.2.

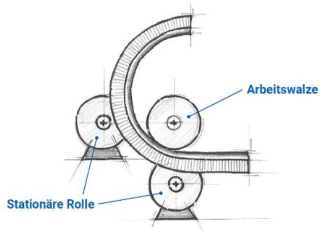


Figure 3.2b: Schematic Three-Roll Bending [28]

Two additional cold bending procedures used in the industry as well as a hot bending process are presented in Appendix B. All prefabricated pipe joints which are fabricated in an off-site facility can be produced under controlled conditions and be thoroughly inspected and tested. As the coating is applied after bending the risk of damaging the coating during bending is eliminated. However, according to Waanders this procedure is costly and time consuming [29].

3.2 Bending on Site

Bending pipe joints in the field before they are welded together is performed by specialized bending machines. Specific bending requirements have to be fulfilled for field bends which can be found for example in the codes ASME B31.4 and ASME B31.8. [5]

"The general rule for field cold pipe-bending is to make bending steps in the same distance as the outer diameter of the pipe and with bending angles between 1.0 and 2.0 degrees" [30]. This results in a bending radius of approximately 40D (40 times the outer diameter of the pipe). The exact angles and steps depend, amongst other factors, on material, thickness and diameter of the pipe [30]. In the following two different bending machines are presented.

3.2.1 Vertical Pipe Bending Machine

One company offering different machines which are each fit to bend a specific range of pipe sizes is called Vietz GmbH. With their machines it is possible to bend pipe joints from 6" diameter, with the smallest machine, to 64" which are bent with the biggest machine.

Before use the machine is equipped with a bending set suitable for the pipe according to its size and coating which conventionally can be polyethylene PE, polypropylene PP or a thin layer epoxy coating [30]. Once on site, the machine is moved on wheels or tracks along the pipe route [30]. There the pipe sections have been laid out according to the design plan as sections are designed specifically for their intended location [26]. The pipe section which has to be bent is guided into the machine by a pipe handling tool which can be for example a crane or a side boom as shown in Figure 3.3c.

An example on the functionality of such a bending machine can be seen in the schematic in Figure 3.3a. This schematic is part of a patent which was filed 2010 by CRC Evans, another company which provides equipment and services for pipeline production [31]. The working principle of the bending machine is assumed to be similar and is therefore described in the following. Once the pipe is guided into the machine it is clamped on the right hand side of Figure 3.3a in a clamp which is not shown in this simplified schematic [31]. Then a mandrel is inserted into the pipe joint (Figure 3.3b). These devices are used to support the pipe wall from the inside during bending to prevent buckling and out-of-roundness. They are placed inside the pipe at the position of the bend then expanded to provide internal pressure [30]. The pipe is bent against the fixed die, D, when the stiff-back, T, is moved upwards by a wedge, W, which is moved to the left due to force, F1 (Figure 3.3a). The force, F1, can be induced by a hydraulic cylinder [31]. The stiff-back and the clamp disengage and with the help of a hydraulic winch installed on top of the machine and the winch wire attached to the pipe joint end the pipe is moved from one bending position to the next [30].

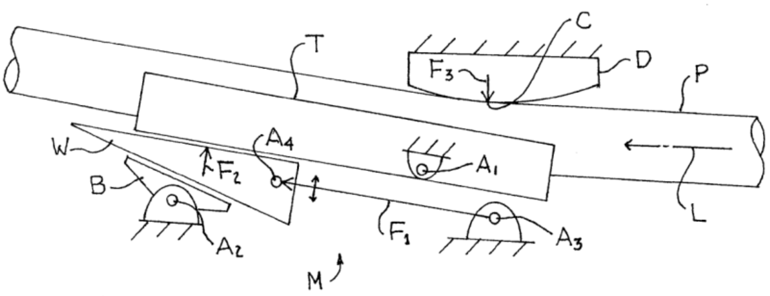


Figure 3.3a: Schematic Onshore Field Bending Machine [31]

Figure 3.3b: Mandrel [30]

Figure 3.3c: Vertical Onshore Pipe Bending Machine [32]

3.2.2 Internal Bending Machine

Maats B.V. developed a pipe bending machine which bends pipe joints in the field from the inside. This machine is especially designed for pipes with external insulation as the external bending machines are damaging the coating. The patent of this design is active and expected to expire in 2034. This machine is claimed to have the advantage that it is faster and more cost effective than fabricating bends in an off-site facility as only strait pipes have to be shipped to site and adjustments to unexpected changes in routing or terrain conditions be performed at the site. For different pipe size ranges, in total from 18″ to 32″ pipes, machines can be rented from Maats. [33]

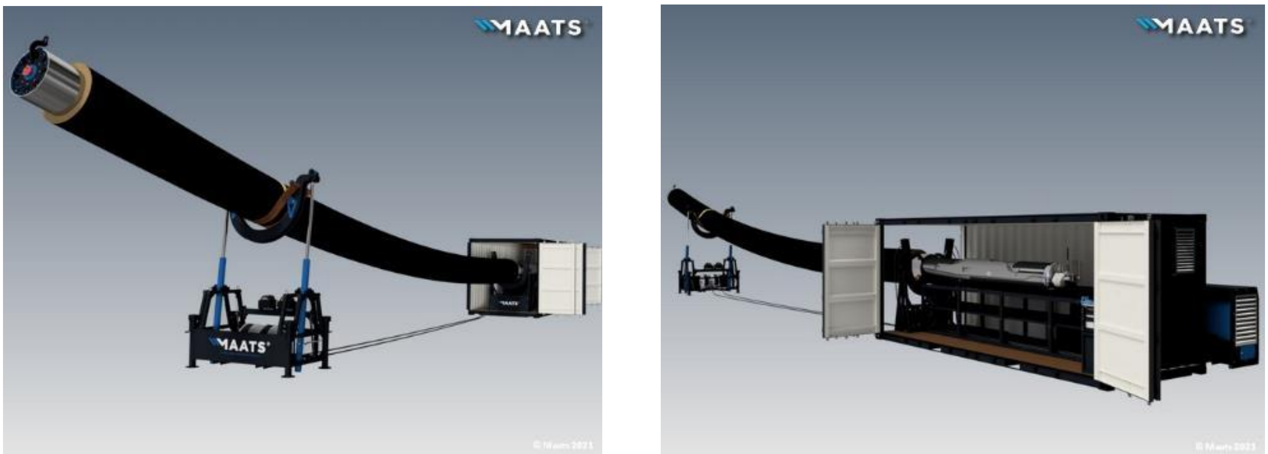


Figure 3.4: Internal Bending Machine Onshore [33]

The bending procedure can be described as follows: Straight 12m long pipe joints are delivered to site. When the machine is set up the steps and angles are feed into the software of the bending machine as this system is "largely automated". Then the pipe joint is picked up by for example a crane and one end is placed on a support in the container of the bending machine the other end is placed on a support which can change the height as it can be seen in Figure 3.4. Once the pipe joint is fixed into position, the bending machine is guided into the inside of the joint. [29]

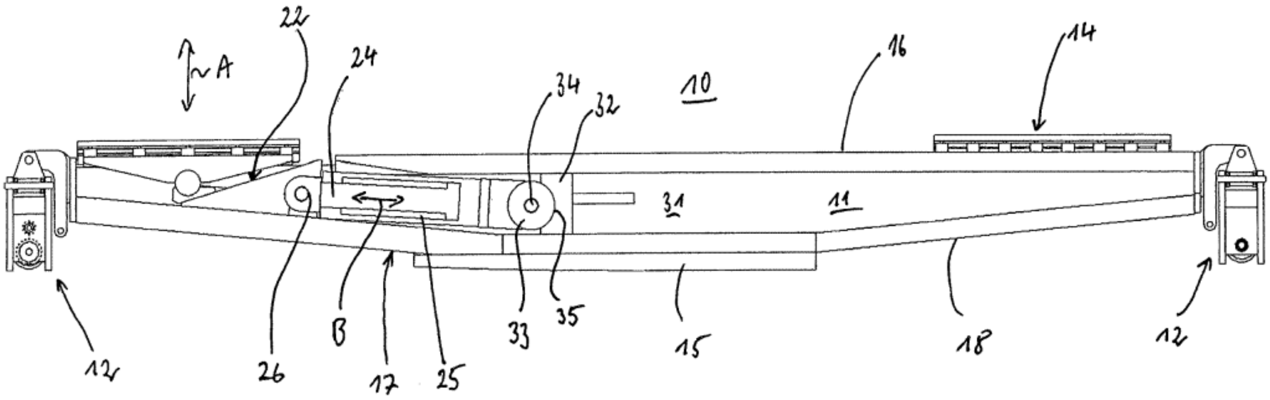


Figure 3.5: Internal Pipe Bending Device - Sectioned View Longitudidal [34]

The bending machine moves inside the pipe on rollers, 12, (Figure 3.5) to be placed in the correct position for the bend. These rollers are then lifted such that they are not in contact with the pipe wall during the bending process. As it can be seen in Figure 3.5 the hydraulic cylinder, B, extends, such that the contact element, A, contacts the inside wall of the pipe to develop the bend. [34]

After the bend reached its required angle, the contact element is moved back into its initial position, the rollers are extended and the machine can be moved to the next bending position until the total desired angle is completed and the bending machine is removed from the pipe joint [34]. Due to the length of the bending machine an unbend end is produced at each end of the pipe (about 2m depending on the machine) [33]. The bend joints are subsequently welded into one continuous length [26].

4

SOLUTIONS OFFSHORE

Offshore, the pipeline does not follow the topography, as it does onshore, but the seabed is modified to create a stable foundation along the selected route. The offshore industry developed a process which is followed to mitigate free spans, unsupported pipe sections which can cause damage of the pipeline. This process, summarized in Figure 4.1 below, consists of the following steps [9]:

Figure 4.1: Free Span Rectification Process [9]

- Free Span Survey and Span Identification

  A survey of the route is performed to gather data of the topography of the seabed. After processing, information about the free span is available for assessment. These include the span length, the size of the gap between pipeline and seafloor, and distance between free spans.

- Engineering Assessment

  In the engineering assessment it is estimated whether the free span causes loads on the pipe which are within the specified limitations. If the limitations are not satisfied a free span mitigation method is selected based on the data of the survey and calculations to ensure pipeline integrity.

- Rectification Planning

  The exact steps on how to perform the free span mitigation are identified in this phase of the free span rectification process. This includes also the implementation of the free span mitigation in the overall project timeline.

- Free Span Rectification

  During this phase the free span mitigation is executed according to the planning of the previous phase.

- Post Rectification Survey

  An as built survey is performed to confirm the final position of the pipeline and that the free span is rectified.

In the Engineering Assessment phase different approaches are assessed to mitigate the free span. Various solutions are available which can be the most preferred option depending on environmental conditions, availability of assets and costs. Solutions which are proven concepts for free span mitigation are presented in the following. These can either be used as a single solution or being combined.

- **Rock/Soil Dump**

Through rock dumping, rock berms can be created as support for the pipeline which limits the free span length. Rock berms can be considered as environmental robust as they are tolerant to scour and suitable for all soil types [36]. However, it has to be considered that they are not stable on steep slopes. In deep water the rocks are installed through a fall pipe from a specialized vessel which is equipment that is typical for dredging companies. The position of the end of the fall pipe can be controlled by thrusters (Figure 4.2). This method is considered as "relatively fast and easy if the transport distance of the rocks is not too far away" [35]. This free span mitigation can be used pre- and post-lay of new pipelines. [35]

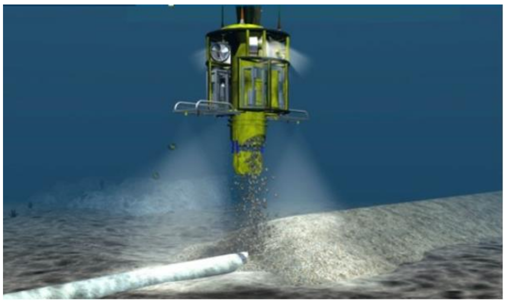


Figure 4.2: Rock Dumping by Flexible Fall Pipe Method [35]

- **Flexible Concrete Mattresses**

Flexible concrete mattresses are not only used as pipeline protection but also as support for pipelines as free span mitigation. This is possible as they can be stacked to piles which support the pipeline. Like rock berms this free span mitigation can be installed on all kind of soil types before or after the pipeline installation [36]. Concrete matrasses can be installed diverless with an quick release frame in very deep water down to 10,000ft which is equal to approximately 3000m as well as in shallow water up to the shoreline. Mattress installation belongs to the activities performed by Allseas. In general, this free span mitigation is "recognized as stable, versatile and easy to apply" [37]. Limitations of this free span mitigation are that they are not stable on steep slopes and they can be stacked about 2m high [36].

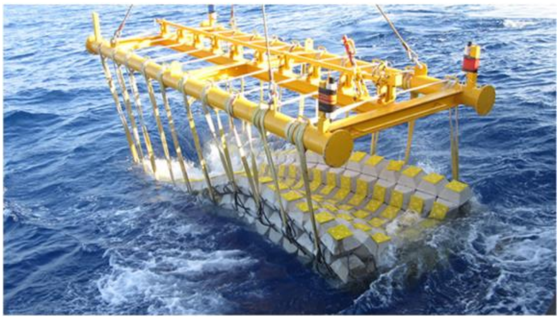


Figure 4.3: Concrete Matress on Installation Frame [37]

- **Grout Bags**

Grout bags are a post-lay free span mitigation which can be performed by Allseas when required. They are bags out of fabric which are filled with grout under water. When the grout hardens the grout bags provide support to the pipeline on different types of soil (Figure 4.4) [38]. As all mitigations which provide support, the bending moments are reduced when using grout bags and the VIV are mitigated as the natural frequencies of the pipeline are changed. The hydrodynamic load on the pipe remains the same. The mitigation of free span by installing grout bags is considered as a standard and cheap solution for water depths up to 1500m but can be affected by scour and wave loadings [9] [38]. Further, this method is not considered to be stable on steep slopes [36].

Figure 4.4: Grout Bag [36]

- **Mechanical Supports**

Mechanical support structures are stable on steep slopes and all soil types. This is an advantage over other mitigations which provide support to the pipeline like grout bags or rock dumping [36]. They are installed by Allseas and have a water depth limitation of 2000m according to Bin. Mechanical support structures require detailed engineering for each free span [36], are subsequently built in an onshore facility and installed offshore after the pipeline is laid. An example is show in Figure 4.5 in which an external clamp can be seen on the right. On the left hand side the installation procedure for this particular structure is sketched. It can be noted that these structures are sensitive to lateral hydrodynamic loads [36].

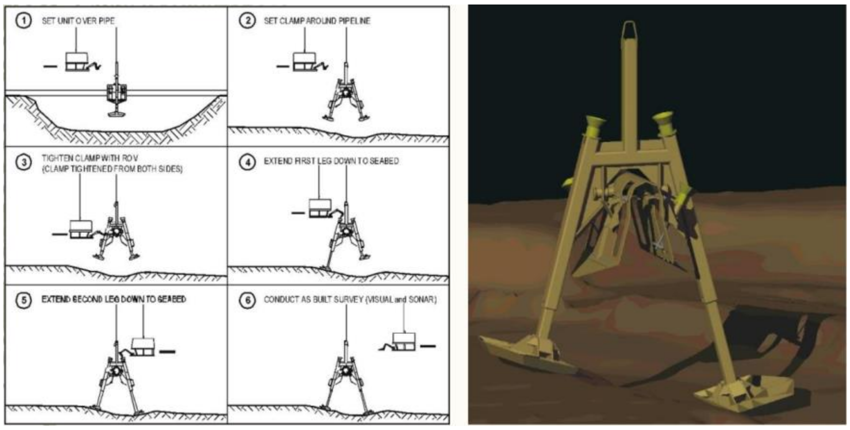


Figure 4.5: Mechanical Support Structure [36]

- **Jetting / Mechanical Trenching**

Jetting and trenching can be performed by Allseas with a trencher called the *Digging Donald* to mitigate free spans. The trencher "creates a V-shaped trench underneath the pipeline using mechanical digging arms which can be seen in Figure 4.6, and multi-pass jetting" after the pipeline is installed by the pipelay vessel [39]. The pipeline sinks into the trench which is created at the free span shoulders. A specific depth of lowering is estimated according to the environmental conditions, including currents, such that the buried sections are not exposed during pipeline's design life. By reducing the span length this way the bending moment, hydrodynamic loads and VIV are reduced. Due to the combined techniques of jetting and mechanical trenching this tool is suitable for all types of sedimentary soils such as sands and clay as well as specific rock types. "The *Digging Donald* can operate in water depths up to 450m" but is not suitable for steep slopes. [39]

Figure 4.6: Trenching and Jetting Tool [39]

- **Ploughing**

Similar to jetting and mechanical trenching a plough can be used to lower the pipeline at the free span shoulders. The working principle of a subsea plough is similar to the technique of an agricultural plough. The plough is deployed onto the seabed, then it is towed by a vessel such that it moves over the seabed on skits to create a trench in which the pipeline settles. This mitigation can be used before or after pipeline installation. If the backfill does not occur naturally due to for example currents close to a soft seabed, then the plough can be used to backfill the trench mechanically. [35]

Figure 4.7: Plough [35]

DeepOcean is a company who operates a plough, the AMP1500, up to a maximum water depth of 1000m [35]. This plough is suitable for a range of soil types including sands, very soft and hard clay. Ploughing is considered as not suitable for steep slopes.

- **Mass Flow Excavator**

Another tool to lower the pipeline into the seabed at the free span shoulders is the mass flow excavator (MFE) which can be seen in Figure 4.8. The MFE fluidises the soil with water jets such that the pipeline sinks into the soil due to its self-weight [9]. It can be used in high water depth and stiff soils such as hardened clay but is not suitable for soft or sandy soil [35]. As the MFE is simply suspended above the location to be excavated from the vessel this free span mitigation is suitable for steep slopes [35]. As of now Allseas does not own a tool like this.

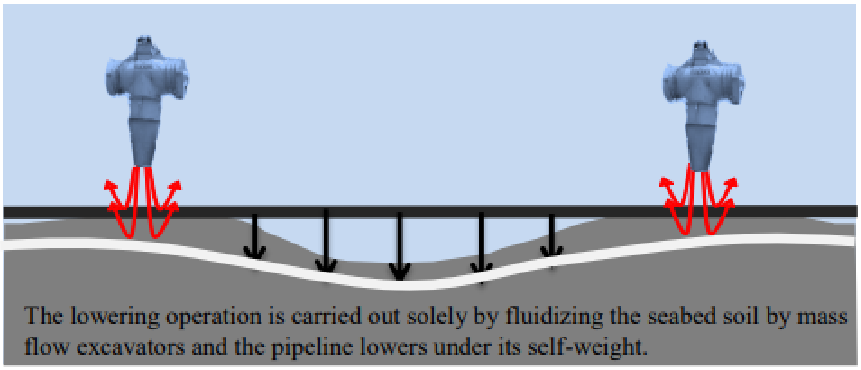


Figure 4.8: Free Span Mitigation - Mass Flow Excavation [9]

- **Peak Shaving**

When dredging is required for a project, this scope of work is subcontracted by Allseas to a dredging company. Peak shaving is used to smoothen the seabed before pipeline installation and remove the peaks on the seabed as it can be seen in Figure 4.9.

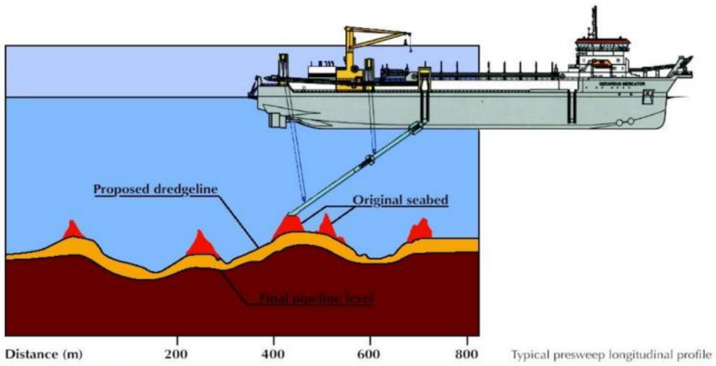


Figure 4.9: Peak Shaving with Trailing Suction Hopper Dredger [36]

With suitable equipment this method is applicable to smoothen the edge at the shoulder of a steep slope as well [13]. To minimize the volume that has to be dredged only a small corridor is excavated. The installed pipeline then lays in a trench which leads to a reduction of the hydrodynamic loading. Reducing span length leads to lower bending moments and changed natural frequencies. Different dredging equipment can be used for different soil conditions and water depth ranging from 30m (cutter suction dredger) to 2000m (grab dredger) [36].

- **Buoyancy Modules / Ballast Modules**

Buoyancy modules can be attached to the pipeline on the pipe lay vessel during installation. They are designed to reduce the self-weight of the pipeline and control the movement of the pipe to mitigate buckling [40]. These modules influence the natural frequencies of the pipeline and therefore VIV. They are suitable for steep slopes, any soil type and water depth. The tough exterior protects the buoyancy module during handling, deployment and operation (Figure 4.10a) [40]. One disadvantage is however that due to an increased diameter, the drag force and therefore the hydrodynamic loading is increased.

To improve the stability of the pipeline at the shoulders of a free span additional weight is an optional solution. The additional weight can be applied by installing ballast modules similar to that shown in Figure 4.10b [41]. An alternative to weight modules is to apply a concrete coating to the pipe joints to increase the weight. These ballast modules are however not considered as free span mitigation as such but rather mitigation for on bottom stability.


Figure 4.10a: Buoyancy Module [40]

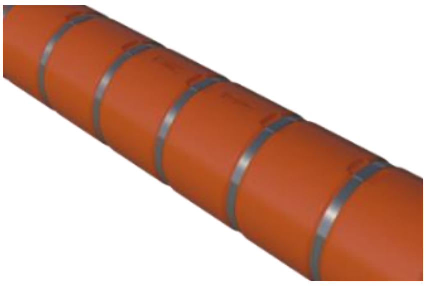


Figure 4.10b: Ballast Module [41]

- **VIV Strakes**

VIV strakes are designed with computational fluid dynamics to suppress vortex induced vibration (VIV) [18]. These strakes consist of one or more fins (Figure 4.11) that cause the vortices to break into shorter and weaker segments which shed randomly such that the pipeline is not excited into vibration close to one of its natural frequencies [19]. These strakes can be installed on the pipeline on the pipelay vessel during S-lay or J-lay [18]. They are a suitable solution for steep slopes, any soil type and water depth. VIV strakes "can suppress VIV very efficiently" but the drag of the pipeline is increased [19]. An experimental study showed further that "modest amount of marine growth, equal to 60% of the fin height are capable of negating most of the suppression benefits" [42]. Managing marine growth is possible with anti-fouling coating, copper based coating, or cleaning via water blasting with a ROV [43]. These mitigations are expensive but below 200m water depth not necessary. Marine growth requires sun light to grow which is only sufficient in the photic zone as it can be seen in Figure 2.2 [12] [43].


Figure 4.11: Helical VIV Strake [18]

4.1 Deficiency Analysis

In this section deficiencies of the different conventional free span mitigations, which are presented in the previous section, are discussed. Analysing operational deficiencies of the current systems is part of the Needs Analysis of the new system concept [4]. The following deficiencies have been identified:

- Performed by Allseas

  Not all free span mitigations can be performed by Allseas. For example, for earth moving procedures specialized dredging vessels are required which are operated by dredging companies. This specific scope of work is then subcontracted.

- Steep Slope

  The mitigation should also be suitable for steep slopes like escarpments at continental shelf breaks. Supports, for example, might not be stable on a steep slope.

- All Soil Types

  Is the mitigation suitable for all kind of soil types? Especially the dredging tools are only suitable for a specific range of soil types and strength.

- During Pipe Installation

  Not all of the mitigations measures can be taken during the installation of the pipeline itself. They require post or pre-lay seabed interventions which extends the timeline of the project.

- Deep Water

  Due to the capability of equipment, some procedures are not suitable for areas with deep water depth which are defined here as 2000m.

- Reduce Bending Moments

  A free span causes high bending moments at the shoulders and in the middle of the pipe. This can be mitigated by for example supporting the pipeline where it is not supported by the seabed.

- Reduce Hydrodynamic Load

  The mitigation leads to a reduced hydrodynamic loading which act on the pipeline due to currents, waves and relative pipe motions DNV-St-F101 4.3.3.

- Reduce VIV

  VIV can be influenced by changing the natural frequencies of the pipeline by, e.g. changing the span length with supports, or influence the flow around the pipeline. "Partial supports may be applied to increase the stiffness of the system, and hence the natural frequencies." [7]

- Environmental Robust

  The mitigation can be altered by the environment. For example scour can cause instability of supports and marine growth impacts the efficiency of VIV strakes.

In the table below it is summarised which conventional free span mitigation has which deficiency. It can be noted that where the free span mitigation can be carried out by Allseas the capacity of the company's equipment is used as reference.

Table 4.1: Deficiencies of Conventional Free Span Mitigations

	Per-formed by Allseas	Steep slope	All Soil types	During instal-lation	Deep Water (2000m)	Reduce bending moment	Reduce hydro-dynamic load	Reduce VIV	Environ-mental robust
Rock/soil dump	No	No	Yes	No	Yes	Yes	No	Yes	Yes
Concrete mattresses	Yes	No	Yes	No	Yes	Yes	No	Yes	Yes
Grout bags	Yes	No	Yes	No	No	Yes	No	Yes	No
Mechanical supports	Yes	Yes	Yes	No	Yes	Yes	No	Yes	No
Mechanical Trenching	Yes	No	No	No	No	Yes	Yes	Yes	Yes
Jetting	Yes	No	No	No	No	Yes	Yes	Yes	Yes
Ploughing	No	No	No	No	No	Yes	Yes	Yes	Yes
Mass flow excavator	No	Yes	No	No	No	Yes	Yes	Yes	Yes
Peak shaving	No	Yes	Yes	No	Yes	Yes	Yes	Yes	Yes
Buoyancy modules	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes	Yes
VIV strakes	Yes	Yes	Yes	Yes	Yes	No	No	Yes	No

Table 4.1 shows that all of the conventional free span mitigations have one or more disadvantages. Only three of the introduced mitigations are suitable for steep slopes and can be performed by Allseas. These are mechanical supports, buoyancy modules and VIV strakes (highlighted in grey in Table 4.1). Buoyancy modules and VIV strakes can be installed onto the pipe during the pipelaying process. Allseas used these mitigations in previous projects in combination with concrete weight to stabilize the pipe at the slope shoulder after detailed assessment during mechanical pipeline design. Both solutions have the disadvantage that they add drag to the pipe which increases the hydrodynamic loading. The hydrodynamic loading on the pipe is also not mitigated by the third free span mitigation for steep slopes, mechanical supports, as well because it only provides support to reduce bending moments and influence the natural frequencies of the pipeline. Mechanical supports, as well as VIV strakes are not considered environmental robust as explained in detail in the previous section.

From this analysis of the operational deficiencies it can be concluded that there is a need for a new free span mitigation concept which can be performed by Allseas, is stable on steep slopes and does not have the deficiencies that the above mentioned mitigations have. Additionally, it should be performed during the installation of the pipeline. By defining these needs part of the second part of the first research question is answered which is about what is needed to address the current limitations. One idea which is explored within this thesis is the possibility to bend the pipeline to mitigate the free span. Different variations of this general idea are discussed in the following chapter 4.2 offshore opportunities.

Regardless of the systems concept architecture which is developed at a later state of the concept development, the new system concept should satisfy a set of different operational requirements which are presented in chapter 5.1. The operational requirements are based on the Needs Analysis performed in the cause of the literature review.

4.2 Offshore Opportunities

In the following different methods and ideas about pipe bending for subsea pipelines are presented. This includes present methods which are used for different purposes than free span mitigation as well as ideas and concept designs for underwater pipe bending as free span rectification.

4.2.1 Offshore Pipe Bends in General

In this section ideas and methods are introduced which are not being used as free span mitigation. These concepts could serve as starting point for new system concept ideas for free span mitigation.

- **Prefabricated joints**

The idea of welding prebent joints into the line pipe, like it is being done for onshore pipelines, has been proposed in the patent of the first under water pipe bending machine in 1998. The idea is to "bend the joints onboard the pipelay vessel before the assembly". In this patent it is however stated that "it has been found that laying a pipeline with bends in it causes excessive stress in the pipeline" and that it is not always easy to calculate the required shape and exact positioning of the bends far enough in advance [10]. A specific challenge which in mentioned in a case study about bending pipelines during Reel-lay is that a single bend tends to rotate by about $90°$ in 300m to 400m water depth [48].

In a new article about subsea bending published 2022, written by engineers from the same company, it is briefly mentioned that it is not possible to incorporate bend joints in the pipeline string when using the S-lay or J-lay method [13].

- **Jumper lines**

Installing structures made out of prebent pipe sections is done for small connections between subsea structures like wells and manifolds for example. These structures are called Jumper lines. Typically, Jumpers are U-, L- or Z-shaped but other shapes are possible as well. Jumper lines are limited in size as they are installed in one piece from an installation vessel as shown in Figure 4.12. The weight of the structure and the available deck space of the installation vessel limits its size. [44]

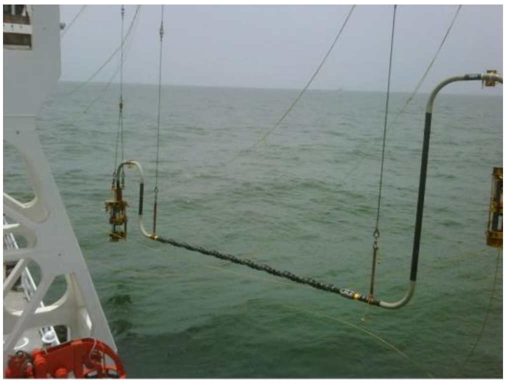


Figure 4.12: Jumper Installation [14]

- **Planned lateral buckling**

Lateral buckling is defined as the buckling of the pipeline in the horizontal plane. It can occur due to thermal expansion and compressive axial forces. The pipeline can be laid in snake-lay configuration such that in case of pipeline extension, the bends compensate for axial forces and it does not buckle in an uncontrolled manner. A pipeline installed in a typical snake-lay configuration is shown in Figure 4.13a. [45]

A variation of the snake-lay is to install buckle trigger structures to induce a horizontal bend in the pipeline. The pipeline is installed according to the selected installation method along the selected route. When the pipeline rests on the trigger, the installation vessel changes the lay direction by about $8°$ to $12°$. This way a bend is created in the pipe around the vertical stopper (part c) Figure 4.13b) which is smaller than the curvature achieved by snake-lay without this additional structure. [45]

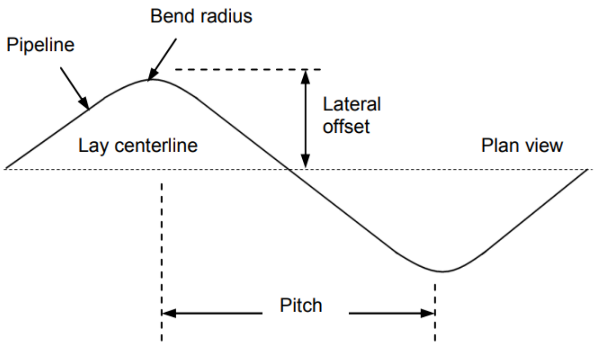


Figure 4.13a: Typical Snake-Lay Configuration [45]

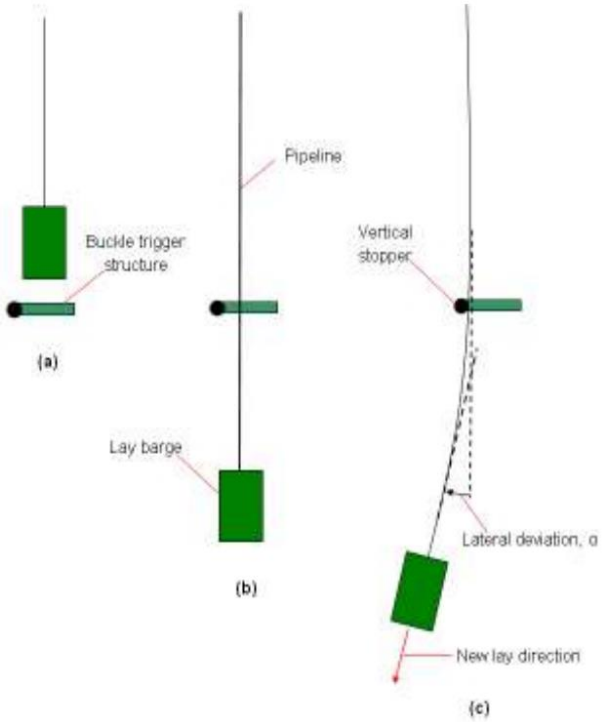


Figure 4.13b: Method of Executing Planned Lateral Buckles [45]

- **Straightener system Reel-lay**

A system with which submarine pipelines are bent is used when installing pipelines with the Reel-lay method. For this installation method the pipe is prefabricated in an onshore facility in long stalks and then spooled onto a reel on a specialized installation vessel. Offshore the pipe is spooled off the reel when it is installed. As it can be seen in Figure 4.14 the pipe is guided through a pipe aligner, 40, a straightener, 50, tensioner, 60, and hang-off clamp, 80, which are all installed on a ramp, 34, on the installation vessel.

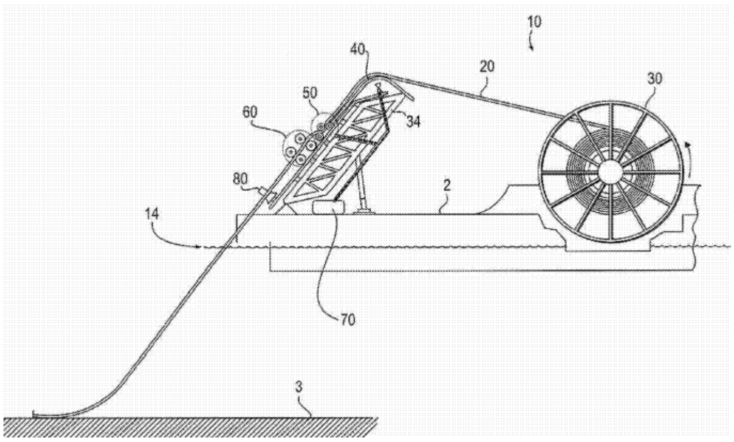


Figure 4.14: Side View of Reel-lay System [46]

The straightener performs reverse bending which removes residual curvature of the pipe and straightens it on the vessel before it is lowered to the seabed. The three-roll bending principle, which is introduced in section 3.1, is applied to the straightener system. Before a new patent from 2012 was published, the settings of the straightener have been determined onshore while the pipe was spooled onto the reel onboard the vessel.

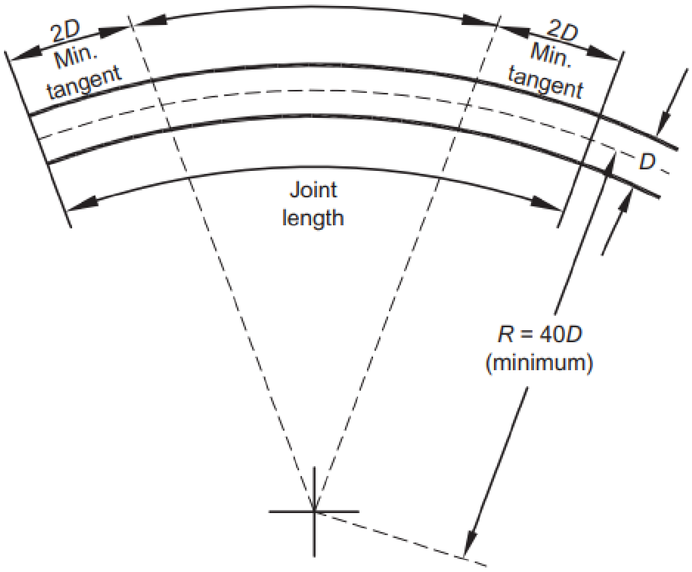
A new application of this concept is introduced in the patent filed in 2012 [46]. The rollers of the straightener can be adjusted as the laying proceeds such that non-straightened parts are introduced into the pipeline. These non-straightened sections provide a mitigation for uncontrolled, undesirable lateral buckling [46]. The settings that can be controlled according to this patent are the engagement and disengagement of the rollers, their relative spacing and positions, and the amount of pressure applied by the rollers [46]. For the Reel-lay installation method the pipe diameter is limited to 20” [47] as the minimum bending radius has to be maintained while spooling the pipe onto the reel.

4.2.2 Pipe Bending as Free Span Mitigation

The standard DNVGL-ST-F101 (section F.5.3.2) from 2017 *permits field "i.e. cold" bends* for offshore pipelines as long as certain requirements are met [8]. These conditions are:

- The bend does not exceed permanent strain of 1.25% (corresponding to 40D bend radius [13])
- The material requirements are met after bending
- Bending machines should provide sufficient support to the pipe cross-section to prevent buckling or wrinkling of the pipe wall and to maintain coating integrity
- Distance from the end of bend to first girth weld should be at least 1.5 times the pipe diameter or 500mm

These conditions are similar to the requirements for onshore field bends according to ASME B31.4 and ASME B31.8 codes which are shown in Figure 4.15.



Notes:
1. Bend radius (*R*) depends on method of bending and pipe *D*/t
2. Bends should be of uniform radius with curvature evenly distributed over the length of the pipe joint
3. Bends should be free from buckles, wrinkles, cracks, or other mechanical defects
4. Ovality after bending should not exceed 2.5%
5. Longitudinal welds shall be near the neutral axis of the bend
6. No bend should be made within 2*D* of a previously made girth weld.

Figure 4.15: Bending Requirements for Onshore Field Bends [5]

Two papers have been published which both present a case study in which pipe bending as free span mitigation is investigated. The first case study from 2015, focuses on the Reel-lay method, for which the three-roll cold bending method is used to bend the pipe such that it follows the topography. The example for the case study used here is an iceberg scar with the dimensions of the free span length of 85m and a span height of 10m. Three different scenarios A, B and C have been investigated which can be seen in Figure 4.16 below. [48]

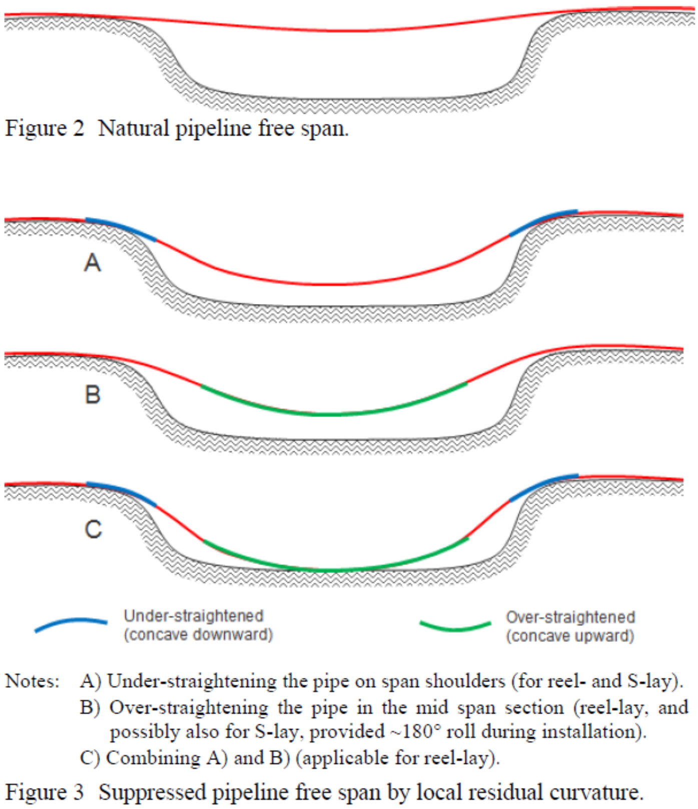


Figure 4.16: Three Scenarios Investigated for Span Mitigation during Reel-lay [48]

In scenario A the pipeline is bent downward at the free span shoulders only. In scenario B the pipeline is bent upward at midspan. Scenario C combines the first two with bends at shoulders and midspan. It has been shown through calculations that the bending moment has the lowest utilization for scenario C which can be seen in Figure 4.17 and is therefore the preferred method for suppressing the span in this case. At the left shoulder the bending moment utilization for this case C is lower than the peak of the bending moment utilization of the normal lay graph which represents the pipeline without bends (right side Figure 4.17). The bending moment is not affected by 5m and 10m tolerance which was proved in an additional study. [48]

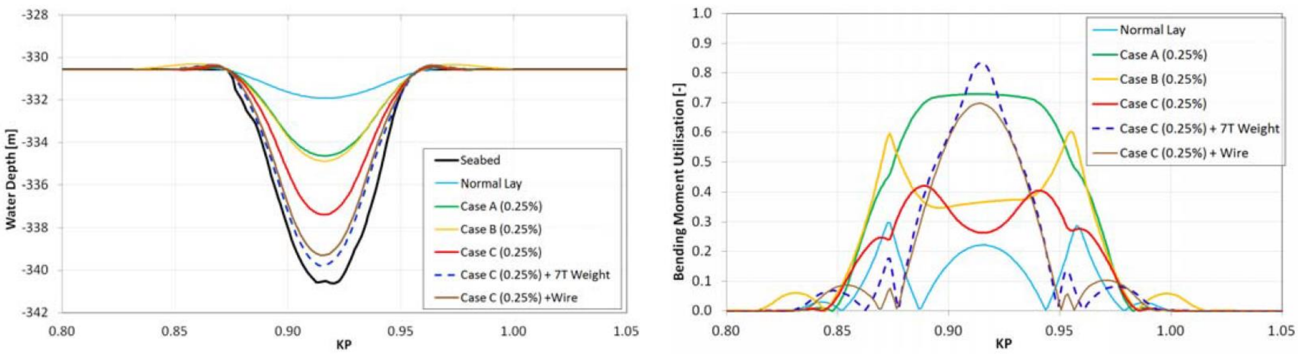


Figure 4.17: Utilization of Bending Moments for Three-Roll Bending [48]

By reducing the free span height by bending, the cost of the conventional free span mitigation is reduced by approximately 10% according to Endal et. Al. Less rock volume is required to support the pipeline after bending compared to the conventional free span mitigation, rock dumping as a single solution, which the new method is compared to.

As described in the previous section the rollers of the straighteners can be adjusted during pipeline installation with the Reel-lay method. In Endals paper it is suggested to adjust them such that bends are introduced to the pipeline during the pipeline installation. These bends, depending on the roller positions can be upward concave or downward concave. Further, it is mentioned that if the pipeline is laid over a single shoulder, the bend tends to rotate, approximately $90°$ in 300m to 400m water depth. Therefore, roll control and contingency measures should be assessed for all cases. [48]

The second case study, published by an Italian offshore construction company Saipem in 2022, is about a pipeline crossing a shelf break. Here, two solutions are compared as well, whereas the conventional method is to dredge the free span shoulder to smoothen the radius such that the minimum bending radius of the pipe is maintained. Depending on the environmental conditions and trenching requirements like water depth, accuracy, soil type and soil volume different trenching methods have been investigated. However, it has been concluded in this paper that the "excavation works exceed the capability of the available technology" such that new tools need to be developed or available tools need to be upgraded. [13]

The second solution for this free span mitigation is again to introduce bends in the pipeline as it can be seen in Figure 4.18. It has been estimated that the bending moment is "minimized according to the performed local buckling check" [13]. This can be seen from the results which are low for the as laid scenario in the lower graphs compared to the peak in the upper graph (Figure 4.18). The upper graph shows the local buckling check for the as laid scenario. According to Pigliapoco the low values of the unity check allow for a large residual bending capacity of the pipeline to "withstand unexpected accidental loads and provide robustness against design data uncertainties and installation tolerances." [13]

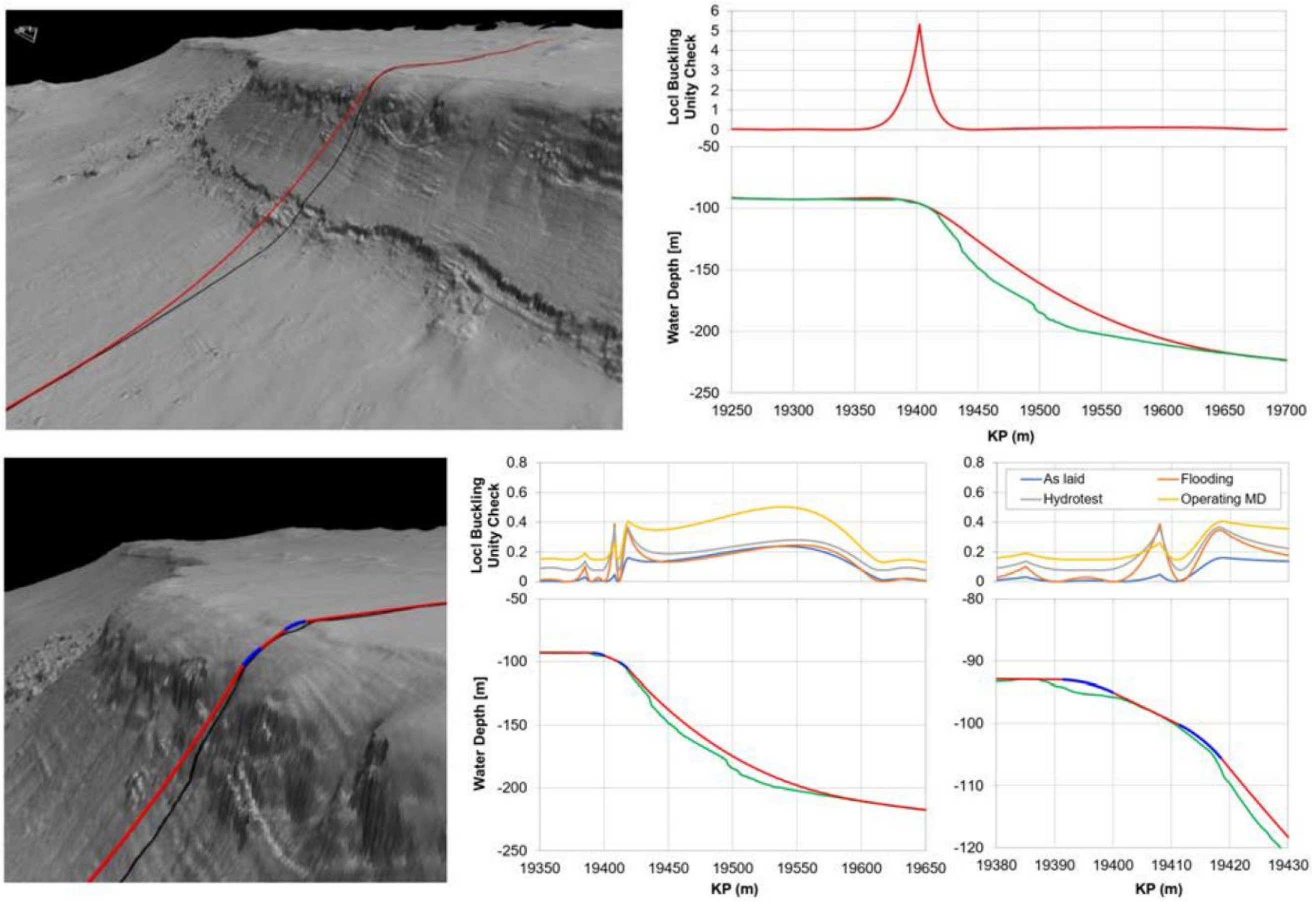


Figure 4.18: Pipe Bending Solution at Shelf Break [13]

These tolerances are specified in a sensitivity analysis. It shows that a slope failure up to 10m does not result in local buckling. Additionally, the local buckling check is still satisfied when the bend is installed +- 2m from the estimated position and the bend angle is $2°$ bigger or smaller than the calculated angle (Figure 4.19).

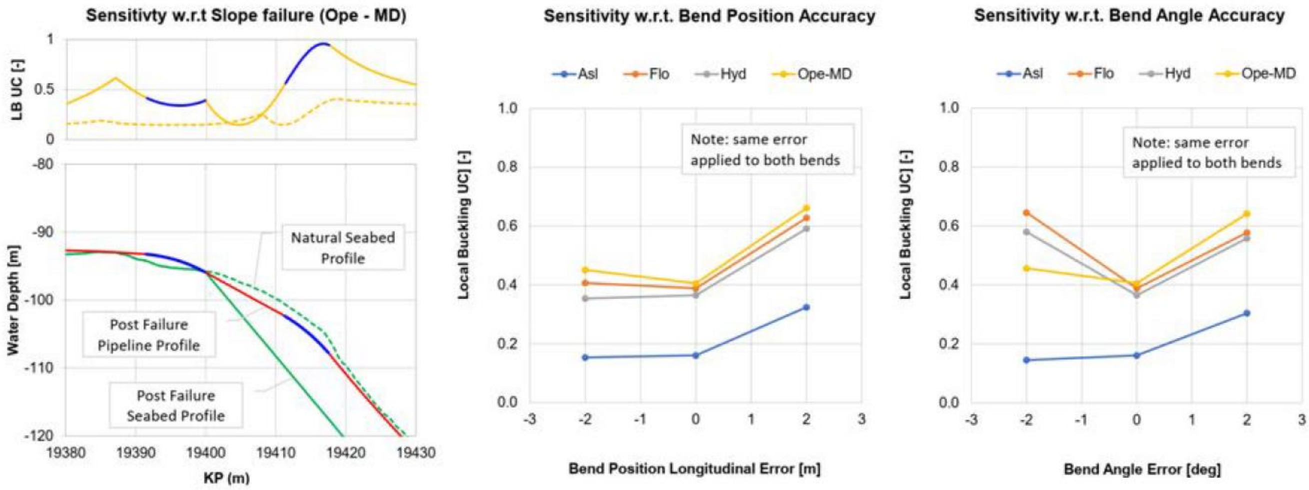


Figure 4.19: Results of Sensitivity Analysis Underwater Cold Bends [13]

The case studies present pipe bending as solution at different locations where different soil types are present. The case study from 2022 was performed for a continental shelf break at which the soil is described as "a rocky skeleton of cemented coral reef material combined with sand filled voids" [13]. In the second case study however, the analysis for pipe bending as free span mitigation is performed for an iceberg scar where the soil is a soft clay [48]. It can therefore be assumed that this new free span mitigation is suitable for different soil types and is stable on steep slopes.

What has not been investigated in both case studies is the effect of bent pipe sections on hydrodynamic loading and VIV. Therefore a brief example calculation has been performed to prove that the hydrodynamic loading is decreased in the free span. The idea is that the current velocity is lower when getting closer to the seabed because of the bottom roughness of the seabed. This reduced velocity results in a smaller drag force in the Morison equation (equation 2.2) which is introduced in section 2.1.2. The DNV-RP-F109 provides the following equation 4.1 to calculate this reduced velocity [49]:

$$V = V(z_r) \cdot \frac{\left(1 + \frac{z_{0a}}{D}\right) \cdot ln\left(\frac{D}{z_{0a}} + 1\right) - 1}{ln\left(\frac{z_r}{z_{0a}} + 1\right)} \cdot sin(\theta_c) \qquad 4.1$$

As an example a current of 3m/s which has a relative angle to the 32'' pipe of $90°$ and a reference height above the seabed of 3m is selected. The drag force of the free spanning pipeline with a height above the seabed of 3m is reduce from 2600N/m to 1600N/m when the pipe lays on the seabed. Additionally, the effective diameter is not increased like it is the case for the other free span mitigations which can be installed during installation, the VIV strakes and the buoyancy modules. As the free span length is also reduced by introducing the bends the pipe section becomes more stiff and the natural frequencies are higher [7]. Therefore the fatigue due to VIVs is mitigated as well.

A procedure for bending the pipeline during S-lay installation is introduced in the second paper. The patented method can be summarized as follows: The pipelay vessel instals the pipe until the touchdown point reaches the escarpment edge where the stresses in the pipe start increasing. Then the pipelay process stops and the tension in the pipe is held constant. A bending machine is deployed from a support vessel and lowered onto the pipe. The machine attached to a ROV performs the bend such that it introduces little bends in small steps. When the bend is completed the machine detaches from the pipe and the pipelaying resumes. [13]

In the patent of this method it is stated that it is also feasible to bend the pipeline after installation. To avoid damaging the pipe and to reduce stresses until it is bend, temporary supports, for example in form of buoyancy modules, are installed during installation [50].

The bending procedure itself is described in the following when two types of bending machines are introduced. The detailed steps of this methods are described in the published article and in the patent as this method is claimed as new invention [13], [50].

4.2.3 Underwater Pipe Bending Machines

How an underwater pipe bending machine can look like was defined first in a patent from 1998. The machine is designed to introduce vertical bends in the pipeline. This patent was eventually withdrawn because examiners found that this invention "did not comply with the requirements of unity of invention and related to several inventions or groups of inventions". [10]

As it can be seen in Figure 4.20a and Figure 4.20b below the tool consist of a chassis, 1, at which a lifting means, 5, is attached. This lifting means can be used to attach the machine to a ROV which provides power and is used to manoeuvre it. When the machine is lifted over the pipeline just before the pipeline is lowered onto the seabed it sits on reels, 2 and 3, on which it can roll along the pipe. Once in position the pipeline is clamped between the bending restrain, 12, and the clamp, 17. Then the hydraulic cylinder, 8, moves wedge, 7, such that the bending shoe, 6, is forced down. The bending shoe, 6, and the clamp, 17, have a concave cylindrical shape such that they fit the pipeline. The bending radius is the same as the curvature of the bending restrain, 12. The final radius of the pipe section is achieved by bending repeatedly very small bends along the pipe section in a distance that can be as small as a few centimetres. Legs with wheels or skids attached to the machine which extent to the seabed can be used to stabilise the machine or to lift the pipeline from the seabed such that the machine can engage the underside of the pipe to close the clamps. [10]

This machine is explicitly developed to reduce free spans. It is stated that "for a typical pipeline on a typical seabed the maximum free span can be considerably reduced with only a small number of bending operations without excessively straining the pipe."

A second version of the invention is in form of a ROV which has buoyancy modules to stay upright, and thrusters to move along the pipeline. The operation of this second form of the bending machine is the same as described above including the legs which stabilize the machine. By extending the hydraulic cylinders which lower the leg to the seabed unevenly this version of the machine might be used to bend the pipe horizontally or upward. "However, for most purposes the additional versatility of such an arrangement is not thought to justify the additional complication." [10]

Additionally, it is proposed to use a mandril which is a device lowered into the pipeline on the pipelaying vessel before it is needed. It remains connected to the vessel via a tether rope which acts as an umbilical to the mandril. Once it is at the location of the bend hydraulic cylinders expand and press staves and rings against the inside of the pipe wall. Support provided to the pipe wall from the inside reduces the risk of buckling of the pipe while it is bent.

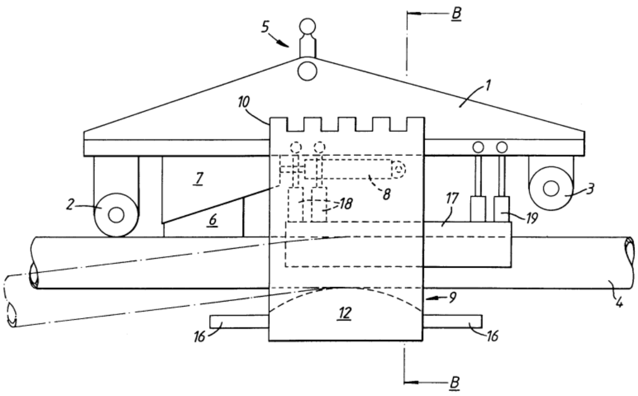


Figure 4.20a: Side View of Underwater Pipe Bending Machine [10]

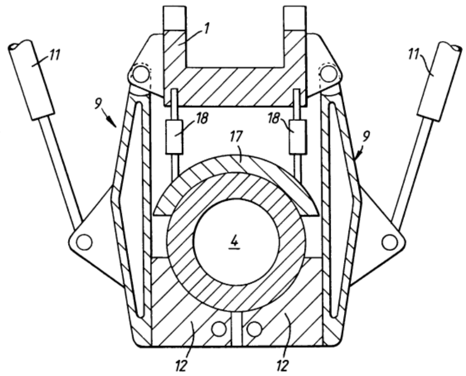


Figure 4.20b: Sectional View of Underwater Pipe Bending Machine [10]

In the article from 2022 a pipe bending machine is presented which seems similar to the tool patented in 1998 [13]. As stated before the method of the bending procedure is patented not the bending machine itself. An overview of the recent design of pipe bending machine and additional equipment used is shown in Figure 4.21. In contrast to the previous design where the machine is lowered to the seabed to the bend location with a cable attached to the lifting means, a deployment frame, 10, is introduced to overcome the splashing zone more safely. The bending machine is then attached to an work class remote operated vehicle (WROV), 8, through a docking plate, 12. Buoyancy modules, 11, enable the machine to be approximately neutral in water and not transfer load to the pipeline due to its self-weight. With the help of cameras, lighting system and instrumentation the lowering onto the pipe as well as the bending process can be monitored. Trimming tanks, 13, are installed at the front and back of the machine. By moving fluid between these tanks the pitch of the machine can be controlled during landing on the pipe. Rollers allow the machine to move along the pipe during the bending process as the total bend comprises of a series of small bends. Once in position the pipe is clamped by activating the pin-up cylinder, 7, which pushes the wedge, 6, to force the pin-up shoe downwards. Then the inboard and outboard cylinders (3 and 4) push the stiff-back, 2, down, thus bending the pipe around the die, 1. When the bend is complete the bending machine detaches from the pipe and is lifted by the WROV which lands the machine on the deployment frame for recovery. This new machine does not make use of a mandrel and the bending machine connected to the WROV are stable enough that no legs are required to extend to the seabed for stabilization.

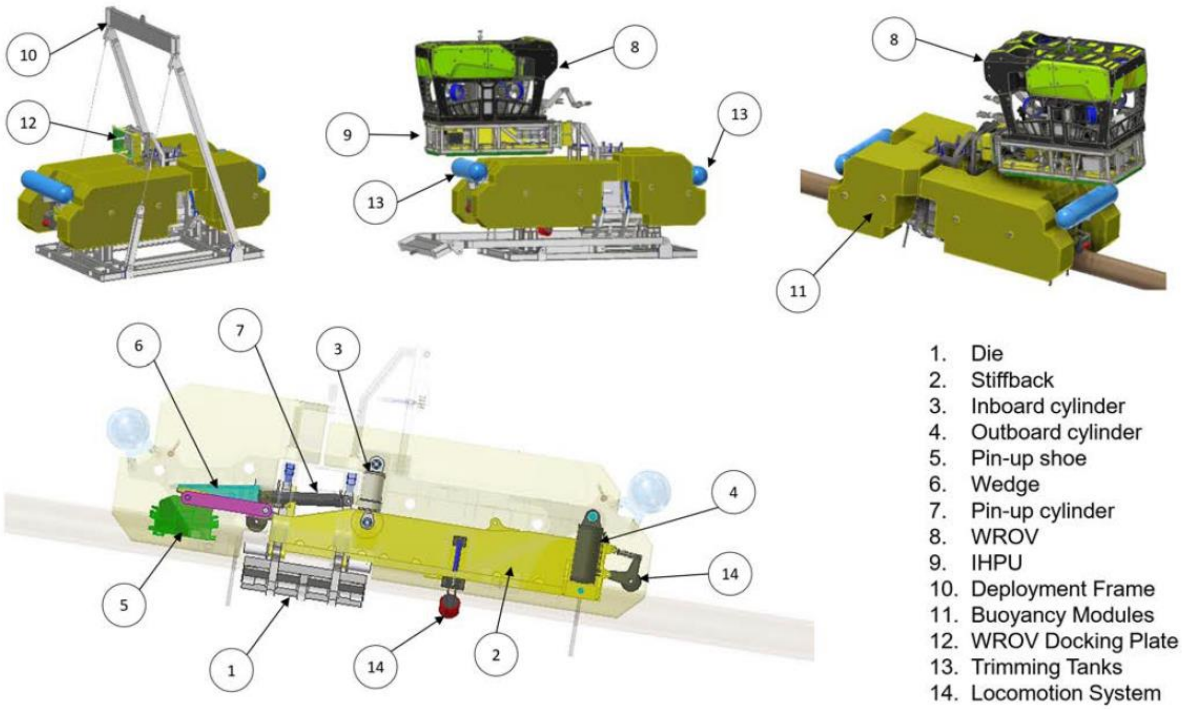


Figure 4.21: Underwater Cold Bending Equipment [13]

4.2.4 Bending with Sleeves

Instead of using underwater bending machines to bend the pipe there are several patents which use sleeves consisting of a number of segments, installed around the pipe to introduce controlled bends into the continuous line pipe. One example of this sleeve invention was patented in 1973 [51]. This patent is however expired. More recent versions of bending sleeves are presented in Appendix C. This sleeve comprises of a number of cylindrical segments in a certain spacing which are interconnected with hinges, as it can be seen in Figure 4.22. The inside of the sleeve is made from suitable antifriction material to facilitate relative movement of the segments and flowline. Loose fit of the sleeve segments provides support for pipe walls to prevent kinking and unwanted deformation. The end segments are secured on the pipe by reducing their inner diameter. These sections are connected by a hydraulic cylinder which when activated retracts. Pipe is bent when the bending moment applied by the cylinder is greater than the tension in the pipe which is controlled by the pipe lay vessel to control the buckling during installation. The curvature of the pipe bend is restricted by the number and length of segments and depend on the diameter of the pipe. When the bend is completed the sleeve is left in place as protection. [51]

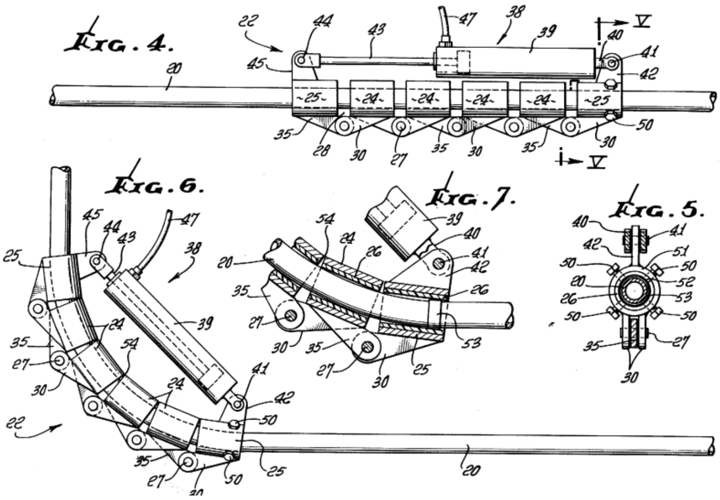


Figure 4.22: Bending with Sleeve Segments Bent by Cylinder [51]

5
CONCLUSIONS NEEDS ANALYSIS

In this literature review the first two research questions have been answered: The state of the art onshore is discussed in section 3, the state of the art offshore is presented in sections 4 and 4.2 and what is needed to address current limitations can be concluded from the analysis in section 4.1. These three parts combined answer the first research question. The operational requirements which are asked for in the second research question are presented in section 5.1.

From the deficiency analysis, performed as part of this literature review on the conventional free span mitigations, it can be concluded that a new system concept for free span mitigation at steep slopes which can be performed by Allseas during pipeline installation, is desirable. Although different solutions have been found for previous projects where free span mitigation was required for steep slopes, these had to be engineered for the specific situation as no standard mitigation is available to mitigate all types of loading. These three types are bending moments at free span shoulder and midspan, VIV, and hydrodynamic loading. Additionally, it is demonstrated that the available conventional free span mitigations have different disadvantages (Table 4.1). A new system concept design as standard solution would reduce engineering time significantly as no individual solution has to be found for each free span.

Bending the pipeline can be such a new system concept as it is shown in section 4.2. Within two different case studies it has been proven that the bending moment in the pipe is reduced at the free span shoulder when the pipeline is bent. According to these papers the bent pipe section is not sensitive to slope failure of up to 10m and has lay tolerances up to +-2m. As both of these case studies have been presented at different conferences of the offshore industry by well-known and experienced companies of this industry and industry standards have been used to calculate the bending moments before and after bending, the results can be assumed to be reliable. An example calculation, according to the DNV-RP-F109 standard, shows that hydrodynamic loading is lower when the free span height is reduced. From the reduction of the free span length it is concluded that the free span becomes more stiff and therefore fatigue due to VIV is reduced as well.

As presented in section 4 many different ideas about underwater bending machines, and methods have been published but as of now it is not known of a project where vertical pipe bending has been realized such that the submarine pipeline follows the topography. Bending offshore pipelines is state of the art for straightening the pipeline when they are installed with the Reel-lay method. Pipelines are also bend horizontally to prevent uncontrolled lateral buckling. Performing field bends in general is permitted according to DNVGL-ST-F101 standard.

Onshore, bending oil and gas pipelines has been done successfully for many years. In contrast to subsea pipelines, onshore pipelines follow the topography such that there are no free spans created. Two different approaches are being used which are welding prefabricated bends, delivered from an off-site facility, into the pipeline, and bending pipe sections at the building site. The functionality of these field bending machines shows that offshore infield bending might be feasible as those onshore oil and gas pipelines are similar in dimensions, coating and material as offshore pipelines.

As it is concluded that the development of a new system concept should be taken to the next stage, the Concept Exploration, operational requirements have been defined based on the findings of the deficiency analysis and the review of existing pipe bending systems.

5.1 Operational Requirements

Operational requirements describe the overall objective of the new system as a whole and refer to its mission and purpose [4]. A new system concept to make free spans acceptable at escarpments, where supports installed by Allseas so far are not stable, shall satisfy the following operational requirements:

1) The new system concept shall provide free span mitigation which can be carried out by Allseas with the support vessel and pipelay vessel that are required for the pipeline installation itself.

2) The new system concept shall allow for a procedure which can be carried out as close to the pipeline installation as possible, ideally during installation. This is required as a longer pipe length is needed when the pipeline follows the seabed topography.

3) The new system concept shall enable the reduction of the length and height of the free span especially at steep slopes like continental shelf breaks, such that the free span becomes acceptable according to the DNV standard.

4) The new system concept shall perform reliable in weather and metocean conditions which are limited by the operability of the pipelay- and support vessel. Additionally, it shall be suitable for pipeline movements during installation.

5) New system concept shall be suitable for deeper waters as well as shallower waters.

6) The new system concept shall be suitable for a range of pipe sizes from 16" to 32" which are typically too stiff to follow the topography. It shall be feasible for a range of coatings and pipe materials.

7) The pipe and coating shall not be damaged if a bend is introduced, due to the bending itself and due to the machine.

8) If the pipe is bent the bend shall at least satisfy the requirements stated in the DNVGL-ST-F101 standard section F.5.3.2.

9) If specific parameters are defined to measure the success of the bend, an adequate monitoring system shall be available to check these parameters.

Concept Exploration

6
CONCEPT DEVELOPMENT

In the Concept Exploration phase a set of functional requirements is derived from the operational requirements found in the Needs Analysis. Based on these different concept ideas are developed. In the following these system concepts and their subfunctions are presented. As final step of the Concept Exploration phase the best three concepts are selected by performing a trade-off analysis. These three concepts are subsequently taken to the next phase, the Concept Definition.

6.1 Functional Requirements

The functional requirements which all possible system concepts need to meet are derived from the operational requirements presented in section 5.1. Two different types of functional requirements have been identified. The pipeline and coating requirements must be met as they are stated in ISO and DNV standards. The vessel requirements, environmental conditions and other additional requirements do not necessarily need to be satisfied as there might be alternative solutions. When for example the machine is too large to be deployed with the *Lorelay* a support vessel can be used for this function.

Vessel requirements	**1.1**	The bending machine shall be deployed from the pipelay vessel or support vessel with the available cranes and deck space.
	1.2	The limiting capacities of the pipelay vessel are the ones from *Lorelay* as she is the smallest pipelay vessel which can install pipelines of the size 2” to 34”. She can operate in water depth ranging from 18m to about 1600m. [52]
	1.3	If the machine is deployed from onboard the pipelay vessel in the way that it is clamped to the pipeline in the firing line the available space for the machine is 573mm in height and 1100mm in width. If the weight exceeds 12.5ton then the length is required to have a minimum value of 6300mm. The maximum limiting weight is 20ton. If the machine is attached to the pipeline after the last station in the firing line the width is limited to 1700mm.
	1.4	If the machine is deployed by crane the limiting weight in air is 300t which is the capacity of *Lorelays* special purpose crane SPC at a radius of 14m. If it is deployed with the AHC winch the limiting weight is 150t. [52]
	1.5	If the machine is deployed through a moonpool in the middle of the vessel the machine size shall be sufficiently small. The dimension of the smallest moonpool from the support vessels, *Oceanic* and *Fortitude*, is 4.8m x 4.9m. The main moonpool on *Oceanic* is 7.2m x 7.2m and 8.4m x 8.4m on *Fortitude*. [53] [54]
Environmental requirements	**2.1**	The machine shall be manoeuvrable at steep slopes such as shelf breaks.
	2.2	The machine shall be clamped securely to the pipe such that it follows its movements during bending. Further it shall be built such that it is not damaged when deploying it through the splash zone.

	2.3	The machine shall be suitable to operate in sea state of Hs 3m and Tz 7s which is the limiting sea state for pipeline installation for *Lorelay*, *Audacia*, and *Solitaire*. If the machine is deployed by crane this shall have an active heave compensation system (AHC) such that the machine does not damage the pipeline while approaching it.
	2.4	The machine shall be operational in a current of 1.5knot. This is the current speed at which the maximum bottom tension for the pipeline installation is determined. These parameters depend on the vessels thruster system capacity. It can be noted that if less bottom tension is required the thrust capacity can be used to navigate in stronger currents.
	2.5	The ROV requirement is that interfaces should be elevated minimum 1.5m above seabed to avoid interference due to seabed disturbance. Additionally, there needs to be sufficient space between the pipeline and the seabed to allow for clamping.
	2.6	The mechanism which provides the force for bending as well as all other components shall be suitable for shallow as well as deep water up to 2000m. If a hydraulic system is used, the needed pressure must also be available in deep water (the hydraulic power unit HPU has to be on the machine).
Pipeline requirements DNG-GL-F101	**3.1**	The bending system shall to provide a bending moment of **16006.00 kNm** (pipe size 32", wall thickness 39mm, water depth 45m, 40D minimum curvature). The calculation of this bending moment is presented below in section 6.2.
	3.2	The bending radius shall not exceed 40D to ensure that the maximum allowable strain of 1.25% as stated in DNG-GL-F101 is not exceeded [13] [30].
	3.3	The pipe shall be bend such that it follows the topography. The achievable angle per pipe joint is described below in section 6.6.
	3.4	The pipe shall not buckle because of the weight of the machine. 1.25 ton/m for 4m long machine is acceptable according to OrcaFlex calculations. That is a total submerged weight of about 5ton.
	3.5	500mm or 1.5D whichever is larger is the minimum distance of the bend to the weld.
Coating requirements ISO 21809-1	**4.1**	The maximum angle per bend shall not exceed "an angle of 2.0° per pipe diameter length" as this is the flexibility requirement for polyolefin coating systems according to ISO 21809-1 [55].
	4.2	The pressure per area at the contact elements shall not exceed 10MPa as this is the stress at yield for PE top layers (class A) [55]. PP coating allows for 18MPa.
Additional requirements	**5.1**	The bending machine shall have a communication system, or data transfer system, such that it is included in the installation process like an additional stage in the firing line.
	5.2	If the machine shall be handled by a ROV in water, the submerged weight is limited to an upper range of 100kg to 200kg.
	5.3	The machine shall have sensors with which the bending radius can be determined as well as cameras to visually monitor the pipe to detect any cracks or wrinkles.

6.2 Calculation of Required Moment

The required bending capacity of the machine is based on the maximum required bending moment. This is why the required moment to bend a 32" pipe with the material grade X65 is calculated. This pipe size is the largest diameter to be bent according to the previously defined operational requirements. An example pipeline from one of Allseas completed projects has the wall thickness of 39mm. This pipeline is installed in different water depths along the pipeline route. It varies between 45m and 2000m.

Analytical calculations are compared to numerical calculations which are done with two different standard software used by Allseas. The first software is OrcaFlex, a package which is based on finite element analysis with which static as well as dynamic analyses can be performed of offshore marine systems. This software is known as a fast and robust system with a user friendly interface. Training is provided as well as a support service. Additionally, it is easy to interface it with MATLAB, Python or enabling OrcaFlex being integrated into third party software, for example for data post processing. As it is used by Allseas for some time, there are simulations of pipeline installations available, which can be used and adjusted to fit the parameters of the simulation done in this thesis. This saves time and less potential for errors. The second software is BendPipe an in-house developed program to calculate non-linear material properties of the pipe.

Onshore, the pipelines are installed following the topography by bending the joints before they are welded into the line pipe. As discussed in section 4.2.1 this approach is not applicable for offshore pipeline installation as the bend would rotate while it is lowered to the seabed and it can be straightened due to the self-weight of the suspended pipe from vessel to seabed. Therefore the common idea of all the system concepts as they are introduced in the following are based on the principle to bend the pipeline close to the seabed.

Additional assumptions for the calculations are:

- The bending radius is limited to 40D (40 times the pipe diameter) to ensure that the limiting permanent strain of 1.25% is not exceeded.
- Ovalization is neglected as the bending moment is reduced when the cross-section deforms.
- The external hydrostatic pressure is neglected as it would act in the direction of the required bending moment and would therefore reduce it.

In Figure 6.1 it can be seen that not only a bending moment is required to overcome the pipe bending stiffness but that there is additionally a bending moment present in the sagbend which acts in the opposing direction than the bending direction. Therefore the total required bending moment is the summation of two parts (equation 6.7).

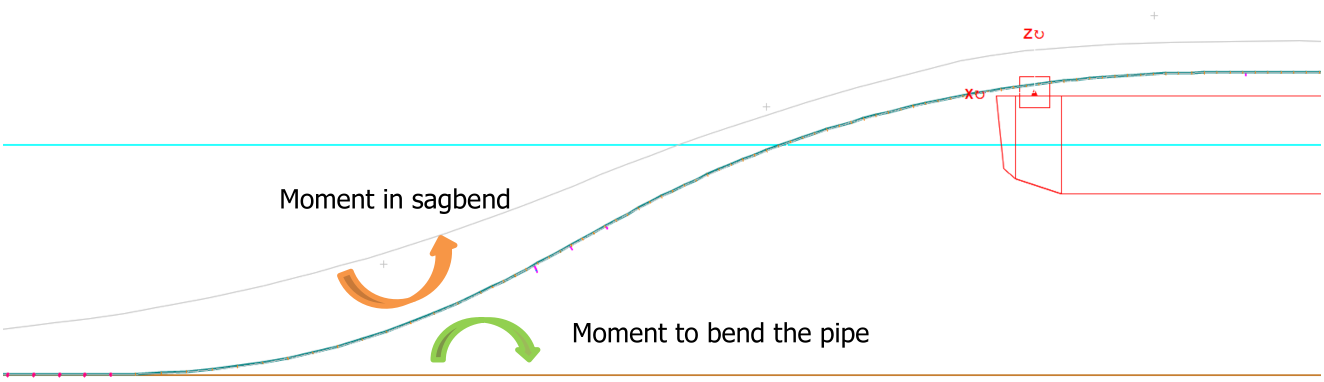


Figure 6.1: Two Parts of Required Bending Moment

The first moment is the moment which is required to bend the straight pipe section showed as the green arrow in Figure 6.1. It is dependent on the pipe properties and the bending resistance. An analytical approach to calculate this moment is to use the equation 6.1 below which results in the moment required to deform the pipe plastically:

$$M = W \cdot [\sigma_T \cdot (1 + s_{ext})] = 9632.86kNm \qquad 6.1$$

With

$$I = \frac{\pi \cdot (OD^4 - (OD - 2 \cdot t)^4)}{64} = 7.114 \times 10^9 mm^4 \tag{6.2}$$

$$W = \frac{I}{0.5 \cdot OD} = 1.75 \times 10^7 mm^3 \tag{6.3}$$

To ensure that the pipeline does not buckle this moment estimated above is compared to the maximum bending moment capacity due to pure bending. Different similar equations can be found in the literature whereas the following, equation 6.4, provides the most conservative result [56]:

$$M_b = SMYS \cdot OD^2 \cdot t \left(1 - 0.002 \cdot \frac{OD}{t}\right) = 11322kNm > 9632kNm \tag{6.4}$$

The yield stress is assumed as SMYS = 458MPa at 0.75% strain which is also the value which can be found in the DNVGL-ST-F101 standard for a X65 material. Figure 6.2 represents the stress strain curve for this material. As it can be seen, the curve becomes more and more constant with increasing strain such that the stress at 0.75% is a close enough approximation for the allowable strain for bends, which is 1.25%.

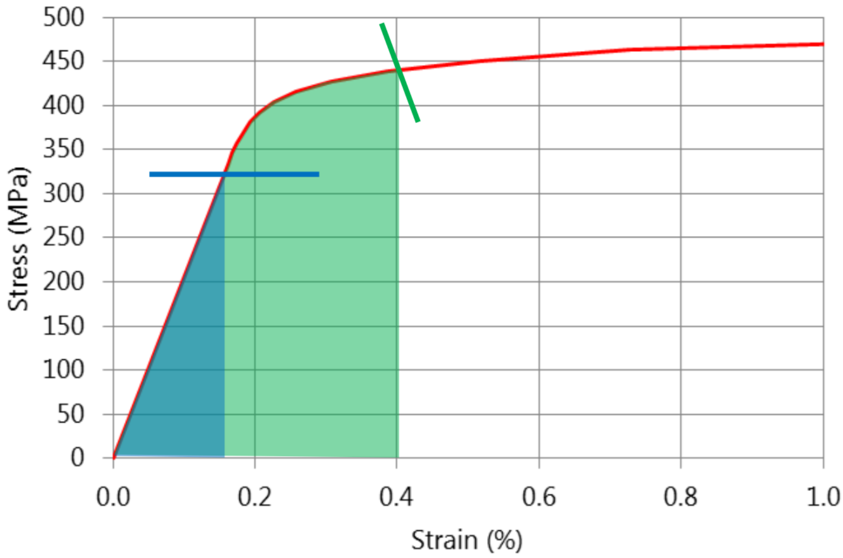


Figure 6.2: Stress-Strain Curve for X65 [3]

This analytical result of the required moment is compared to the result obtained with the in-house software BendPipe. It can be noted that the minimum bending radius which can be input in this software is 55m instead of 32m which is the 40D radius of the 32” pipe. That is because this software assumes that the maximum allowable strain is 0.75%. As discussed before, this is also the strain for which the yield stress is assumed in the analytical calculation. Therefore these two results are comparable. The maximum moment which the software provides is *10336kNm*. The detailed output of the BendPipe calculation can be found in Appendix D.

The second part of the total moment is the moment which is needed to overcome the sagbend moment close to the seabed during installation. This sagbend moment is represented by the orange arrow in Figure 6.1. This moment acts in the opposite direction, upwards, than the direction the pipe is being bend, which is downwards. Assuming a linear approach, this moment can be calculated with equation 6.5.

$$M = EIk = 6072.9kNm \tag{6.5}$$

With

$$k = \frac{1}{R} \tag{6.6}$$

The bending radius of 246m is equal to the sagbend radius obtained from OrcaFlex static simulation for the 32” pipe in 45m water depth. This shallow water depth is assumed for this calculation as the bending moment is larger compared to sagbend moments in deep water. The analytical calculated moment in the sagbend is then 6072kNm which is close to the moment which is calculated with the software which is *5670kNm* in the sagbend. The numerical moment is smaller in this case because here the pipe is under compression in the sagbend.

To consider the highest bending moments the required bending moment is the summation of the moments obtained from the two different computer software. The requirement bending moment which is used as basis for the design for the different concepts is therefore 16000kNm as it can be seen in equation 6.7.

$$required\ moment = 5670kNm + 10336kNm \approx \mathbf{16000kNm} \tag{6.7}$$

These assumptions and calculations have been reviewed by two experts in this field: A senior R&D engineer from Allseas whose expertise lays in developing new machines and applications and an assistant professor from Delft University of Technology of the department Ship and Offshore Structures. His expertise is in steel, ocean structures, fatigue and welds.

6.3 Coating Requirement

There is a range of different coatings available for submarine pipelines. These include according to DNVGL-ST-F101 the coatings listed below whereas the first three coatings are external corrosion protection:

- 3-layer polyethylene (PE)
- 3-layer polypropylene (PP)
- Single layer fusion bonded epoxy
- Liquid epoxy
- Concrete weight coating
- Thermally insulating coatings

In the ISO 21809-1 standard requirements for different polyolefin coatings (3-layer PE and 3-layer PP) are specified. In the following it is estimated if the requirements of the PE and PP coatings are met when the forces of the different bending machines are applied. The focus is first laid on these coatings as it is state of the art onshore to bend pipe joints with these coatings. Additionally, the epoxy coating becomes like glass and concrete is also considered too brittle to be bent. As it is the case for onshore pipelines it is considered that pipelines with insulating coatings should only be bent with internal pipe bending concepts.

The first possible failure the coating can experience is cracking due to bending if the material is not flexible enough. In the ISO standard it is specified that a coating system is qualified when the flexibility of the system is such that there is "no cracking at an angle of $2.0°$ per pipe diameter length" [55].

It is assumed that if the angle per bend is smaller than $2.0°$ then the coating is not damaged. For the 32" pipe with a bending radius of 40D and a distance of 32" between the bends the bend angle is calculated as shown in equation 6.8:

$$angle\ per\ bend = tan^{-1}\left(\frac{OD}{40D}\right) = 1.43° < 2.0° \tag{6.8}$$

The second possible failure of the coating is that it can be damaged because of indentation of the contact element of the machine. A coating is qualified when the indentation depth is less than 0.4mm when a force of 10N/mm^2 is applied over a predefined time. The property which describes this coating capacity is the stress at yield which can be seen in Figure 6.3 below. The coating class A, B and C are depending on the layer thickness [55].

Properties	Unit	Test method	Requirements		
			Class A	Class B	Class C
Density of black compound	g/cm³	ISO 1183 (all parts) or ASTM D792 or ASTM D1505	≥0,930	≥0,940	N.A.
Density of the base resin (not black compound)	g/cm³	ISO 1183 (all parts) or ASTM D792 or ASTM D1505	≥0,920	≥0,930	≥0,890
Carbon black content	%	ISO 6964	2 – 3	2 – 3	N.A.
Carbon black dispersion	—	ISO 18553	Max Grade 3	Max Grade 3	N.A.
MFR	g/10 min	ISO 1133-1	Within manufacturer's specification		
Strain at break at 23°C ± 3°C[a]	%	ISO 10350-1 ISO 527-2	≥600	≥600	≥400
Stress at yield at 23 °C ± 3 °C[a]	MPa	ISO 10350-1 ISO 527-2	≥10	≥15	≥20

Figure 6.3: Minimum Requirements for PE/PP Top Layer [55]

Based on this requirement the contact area of the system concept is adjusted such that it is large enough to distribute the force sufficiently to not damage the coating. It is assumed that for PP coating 18MPa is the limiting stress and for PE 10MPa respectively.

It can be noted that a solution for concrete coated pipe can be to weld a few sections of PP or PE coated pipeline into the line pipe where the estimated bends are positioned. This is considered feasible because one of the functions of concrete coating is to add weight to the pipeline. To ensure on bottom stability the thickness of the coating can be increased a bit to account for the joints without coating. A similar idea is presented in section 4.2.4 where sleeves are proposed to be used for bending of concrete coated pipelines as well.

6.4 The Functions

Based on the external and internal onshore bending machines as well as the straightener system of a Reel-lay installation system the following subfunctions have been identified which a pipe bending concept should be composed of.

- A body which differs in how the machine approaches the pipe
- A clamp which provides the reaction force to the bend force
- A bending mechanism which provides the bending force
- Locomotion system to manoeuvre the machine from bend to bend
- A bend die which is the moving contact element to bend the pipe

For every subfunction different solutions have been investigated. These solutions are described in the following and are summarized in Table 6.1 below. This specific table is called a morphology matrix [57]. This matrix provides an overview of all the possible combinations of the different solutions for each subfunction. Some combinations of solutions do not make sense, like for example combining a clamp solution for clamping the machine inside the pipe with a ROV body which is approaching the pipeline from the outside. The solutions in parentheses written in italic are ideas which are dismissed based on calculations or operational reasons. Based on dismissed solutions and logical combinations the number of concepts is narrowed down.

Table 6.1: Morphology Matrix – Subfunctions of Bending Machine

		solutions					
subfunctions	*Body*	ROV approaching from above	ROV approaching from the side	*(internal / inside the pipe)*	over stinger/ roll over pipe		
	Clamp	*(soft robotics)*	mechanical outside bent half shell	*(mechanical inside, like mandrel)*	*(roller)*	like a tensioner	*(Slings)*
	Bending mechanism	hydraulic cylinder	*(screw press solution with motor)*	*(rack and pinion)*	*(wedge with horizontal cylinder)*		
	Locomotion system	tracks (similar to tensioners or tank tracks) on/in pipe	*(thrusters)*	both rollers and thrusters	*(self-folding propellers and tracks)*	Handover clamps	
	bend die	like a tensioner	like a half shell convex or concave	*(roller)*			

Subsequently, different solutions are described and in some cases why they are dismissed, such that in the end twelve concepts (presented in Table 6.2) found as combinations of solutions of the subfunctions are considered feasible. A trade-off analysis is used to find the top 3 concepts of these twelve.

6.4.1 The Machine Body

Different deployment methods and different approaches to clamp the machine around the pipe influence the design of the machine body. In the following three different solutions are presented.

- **ROV body**

A first solution is to design a new ROV which is deployed from a support vessel or the pipelay vessel itself. It is designed such that it can be lifted into the water with one of the available cranes, a launch and recovery system (LARS), or a winch. Although the *Lorelay* has a LARS for a ROV which can monitor the pipeline installation a special purpose crane SPC is more likely to be used to deploy the bending machine from this pipelay vessel. This is currently the case for trenching operation. The SPC is then used to deploy the *Digging Donald* such that the ROV can still be deployed with the LARS system to enable monitoring at the same time. The maximum lift capacity of this SPC and therefore the limiting weight in air of the ROV concept is 300t. Current subsea tools are however lighter in air than these 300ton. As reference, a WROV weighs about 5ton in air [58]. It can be noted that it is preferable to design a small and light machine as, for example, the price for the umbilical increases when a larger minimum breaking load capacity is required. Another factor to consider is the limiting submerged weight. As the machine becomes heavier more buoyancy modules are required to achieve the same weight in water.

The allowable submerged weight on the pipeline is determined by the strain in the pipe. To determine this parameter a simulation is performed in OrcaFlex. A file from an old project is used as basis. The model includes the pipelay vessel *Solitaire* with the stinger attached and the pipeline as it is suspended from the vessel over the stinger to the seabed in the S-lay shape. In this case, the *Solitaire* is the pipelay vessel because she is more likely to be used in the chosen water depth than the *Lorelay*. A model is chosen which already simulated the pipeline installation in similar water depth such that the stinger radius is not modified. The water depth is changed to 2000m as it is the deepest water depth according to the functional requirement 2.6. Together with the smallest pipe size of 16” with a wall thickness of 2cm (operational requirement 6) the scenario, which is most sensitive to buckling is simulated. The wall thickness of 2cm is chosen to achieve a D/t ratio which is within the range of $15 \leq d/t \leq 45$ for which the buckling check according to DNVGL-ST-F101 is valid. After the buckling check is passed and the required tip separation is met the machine is added to the model. It is then modelled as a stiffener, which is a 4m long pipe section which inner diameter is equal to the outer diameter of the pipe. As shown in pink in the left bottom drawing in Figure 6.4 the machine is placed close to the seabed where the pipeline is supposed to be bent (functional requirement 2.5). The weight for which the strain is still acceptable is 1.25ton/m. As it can be seen in the graph on the left (Figure 6.4) the strain in the overbend is then still below the typical limit of 0.35% and in the sagbend where the machine is placed it is below the typical limit of 0.15%. As shown in the top left the, buckling check is also acceptable. Assuming an overall length of the machine of 4m the total limiting submerged weight is then 5ton.

A similar simulation has been performed for a 6m long machine which results in the same limiting submerged weight of 5ton. The estimated weight here is 1.85ton/m.

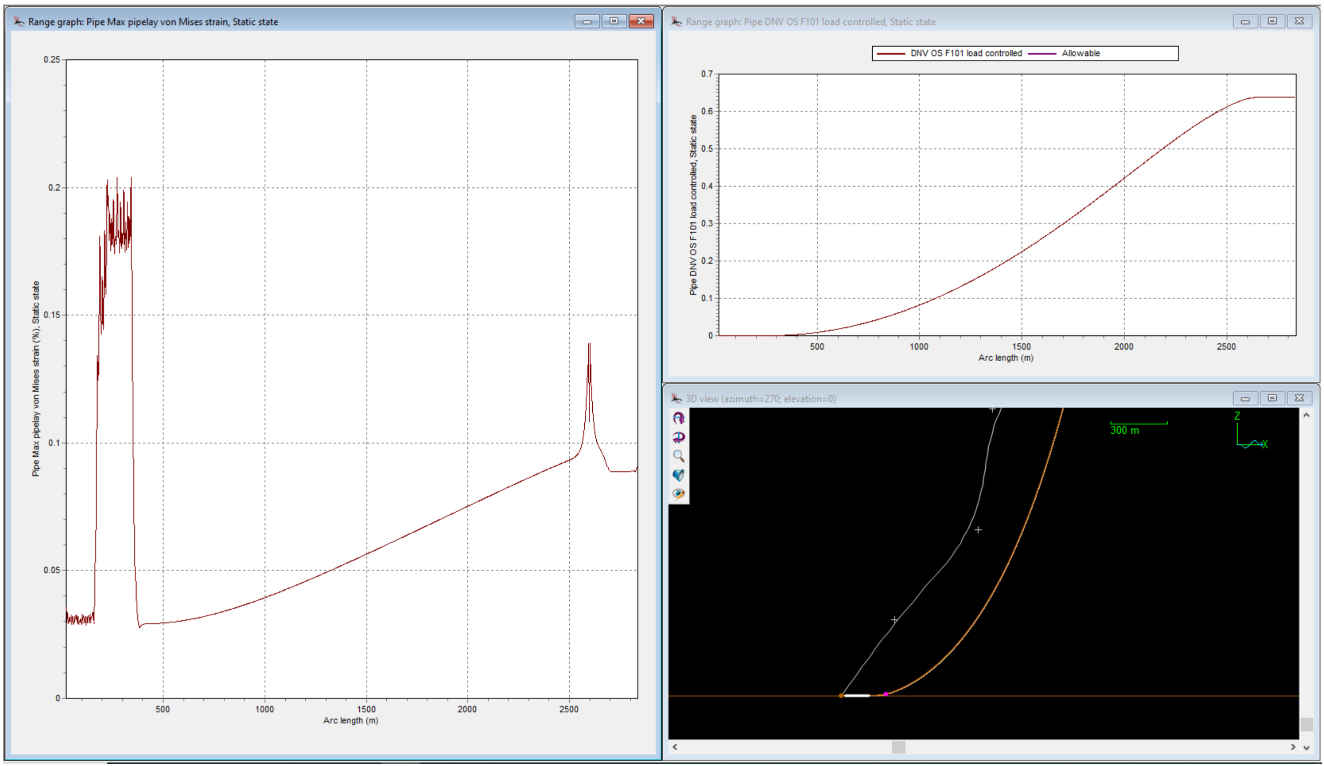


Figure 6.4: Load Controlled Buckling Check and von Mises Strain - 4m Long Machine

One option of the ROV body is to design a new ROV such that every part is fitted onto the machine like thrusters, HPU unit, and sensors. An alternative is to use an AUXROV which is a unit at which tools can be attached, like grabs or mass flow excavators as it can be seen in Figure 6.5. The AUXROV then provides the sensors and thrusters and power to the tool. The interface between the AUXROV and the tool is a universal docking plate [59].

Figure 6.5: AUXROV by Aleron [59]

The bending machine with a ROV like body can approach the pipeline section close to the seabed either from above or from the side as it can be seen in the schematics in Figure 6.6. When the ROV approaches the pipeline from above the clamp needs to reach around the pipe. Here as an example a half shell formed clamp is connected to a pivot point about which it can be rotated. Once the clamp is located under the pipe it is lifted by hydraulic cylinders to clamp it. The body on which all the elements are mounted is then located above the pipeline.

When the ROV approaches the pipeline from the side the clamp simply has to be moved up and down as it can be seen on the right side of Figure 6.6. The ROV body can then be composed of one or two parts of which in any case one part is located on the side of the pipeline. There might be a part added above the pipeline when more space is needed for, for example buoyancy or when the machine is then floating more stable in the water.

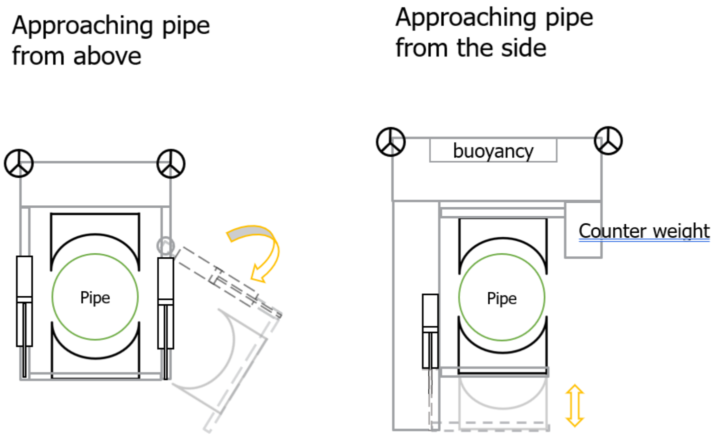


Figure 6.6: Schematics - ROV Concepts

Examples of systems which make use of rotation or vertically movable parts are tensioners which open by rotation of the upper track or vertically moving the upper track respectively as presented in Figure 6.7a and Figure 6.7b.


Figure 6.7a: Tensioner - Opening by Rotation [60]


Figure 6.7b: Tensioner - Opening by Moving Upper Track Vertically [61]

- **Internal bending concept**

A second idea is to launch the machine from the pipelay vessel by inserting it into the pipeline. For onshore pipelines there are internal bending machines on the market which are mainly used for bending pipe joints with external insulation which shall not be damaged. This concept is presented in section 3.2.2. The offshore internal bending machine stays connected to the vessel via an umbilical with the possibility to attach an additional winch wire to compensate for the weight of the machine. By paying out the winch wire and the umbilical the machine is lowered until it reaches the bend section which is in the sagbend region. It can be noted that when installed in deep water the suspended pipeline can be close to vertical in the middle of the water column.

Simulations with the software package OrcaFlex show that the pipeline integrity is not compromised when deploying the machine inside the pipeline. Details of this calculations can be found in Appendix E.

A machine which is inside the pipeline during offshore pipeline installation is a buckle detector as it can be seen in Figure 6.8. It is a gauge plate made from aluminium which is mounted on a frame with rollers. The buckle detector is inserted into the pipe and connected to the vessel by a wire. Buckles are detected when the pulling force is increased. The wire is connected to the line-up clamp such that new joints can be welded to the line pipe. This device is considered a traditional approach to detect buckles in the pipeline. However, it is not the preferred solution to detect buckles, as it is a frequent problem that the wire brakes and the buckle detector gets lost in the pipeline. Retrieving the buckle detector is difficult and time consuming. Because of this experience with the buckle detector "nothing should be put inside the pipeline during installation" [62]. These described experiences in the article written by employees of Det Norske Veritas are the same as the field engineers described at Allseas as well.

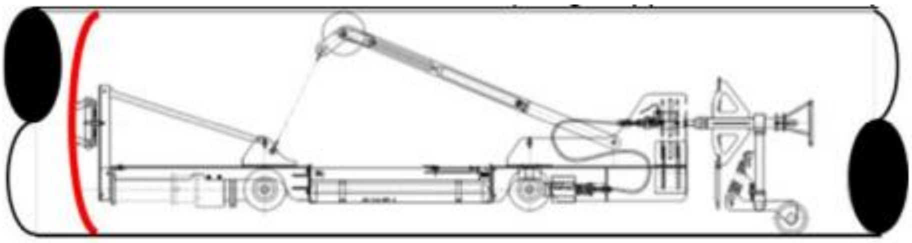


Figure 6.8: Buckling Detector with the Gauging Plate in Red [63]

Following this reasoning of the operational difficulties, the approach to deploy the bending machine inside the pipeline during installation is considered as too risky.

- **Deploy the machine on top of the pipeline**

The third solution to deploy the machine is to set it on top of the pipeline on the pipelay vessel. Depending on the contact elements, the clamp and the bend die, the machine rolls on the pipe over the stinger, is clamped to pipe and reaches the sagbend as the pipe is installed, or it uses the "handover" locomotion system to move along the pipe.

Figure 6.9: Typical Stinger

As the pipe is supported by roller boxes on the stinger it needs to be considered in the design of the machine that the clamp is either suitable to be rolled over the roller boxes on the stinger or that the clamp is not engaged when the machine is on the stinger and clamps the pipe when the machine leaves the stinger.

As the machine is deployed over the stinger, the width of it is limited by the stanchions which are on either side of the roller boxes as presented in Figure 6.9. However, based on the procedure how to lift the machine onto the pipe the lifting capacities and available space inside the vessel is governing in this case when using the *Lorelay* and not modifying the vessel.

The procedure to lift the bending machine onto the pipeline is based on the method of installing inline structures on the *Lorelay*. The machine can be lifted from the storage area on deck with the SPC through a hatch onto a trolley which is called the *Mercedes*. Then the machine is brought to a position in front of the first tensioner. There it can be lifted with one or two overhead cranes onto the pipe. Once placed on the pipe the machine then needs to pass the remaining 2 tensioners before being guided over the stinger. Because of this process the dimensions are limiting for the system concept which is deployed over the stinger. These are summarized in the functional requirement 1.3.

Additional considerations include that none of the onboard cranes can reach the stinger tip to place the machine on the pipeline after the stinger tip. This means that if the machine is not able to crawl back over the stinger a support vessel is needed to recover it from the pipeline. When deploying the pipe over the stinger it needs to be ensured that the machine on the pipe stays upright. This can be achieved with buoyancy modules which influence centre of buoyancy.

6.4.2 Mechanism for Bending

In the following different approaches for the bending mechanism are presented. These mechanisms shall be suitable to function in shallow as well as deep water and provide sufficient capacity to apply the required bending force.

- **Sizing of hydraulic cylinders**

A first idea to apply the force is to make use of hydraulic cylinders. The force, which the hydraulic cylinders need to provide, is depended on the machines' leaver arm which varies among the different concept ideas and can be changed in order to optimize the design. It can be estimated by dividing the required bending moment by the lever arm. The calculation of the bending moment is described in section 6.2.

The size of the cylinder can either be estimated by simply dividing the force by the pressure of the hydraulic system. Common high pressure hydraulic systems have a pressure of 700bar. This results in the area of the cylinders piston as it can be seen in equation 6.9 below.

$$F = P \cdot A \tag{6.9}$$

Often the cylinder size is specified by the weight they can lift. The required tonnage can be estimated by dividing the force by 9.81 m/s^2. The result is then the first selection criterion for the hydraulic cylinder. The required stroke is calculated according to the standard geometry equation 6.10 for a segment of a circle:

$$h = 2r \cdot sin^2 \left(\frac{\alpha}{4}\right) \tag{6.10}$$

For the minimum bending radius of 32.5m (40D of 32" pipe) and a leaver arm of for example 4m the stroke is 0.06m. It can be noted that the shorter the leaver arm the smaller the stroke becomes, but at the same time the force, that the cylinder needs to provide, increases.

To account for the spring back the results of the BendPipe calculation is used. Part of the output, which can be found in Appendix D, is the residual radius. When comparing this residual radius with the minimum radius which serves as input then a factor can be derived as division of these two radii. To achieve a more accurate factor four different calculations are performed within which the input minimum radius is changed in steps of 5m from 55m to 70m. Then the geometric mean of the obtained factors of these four calculations is the coefficient which is used to calculate the minimum radius to achieve a residual radius of 40D. It can be noted that the largest stroke is required for the smallest pipe size of 16" as defined in the operational requirements. The minimum radius is then 9.2m to achieve a residual radius of 16m. The required stroke of the cylinder when the leaver arm is 3m for this minimum radius is 12cm according to equation 6.10.

From the available cylinders on the market with the right capacities the diameter and weight of the cylinder can be found. One example of suitable hydraulic cylinders are high tonnage double acting cylinders which have a capacity of up to 1000ton [64]. This example is a "normal onshore" cylinder. The size of the cylinder and the length of the leaver arm can influence the more detailed design of the bending machine concept.

The sizing remains the same for cylinders which are used for subsea applications. In general, the differences between subsea cylinders and the once used on land are the materials. Special seals make them water tight and operable in deep water. Suitable coating protect the cylinder against corrosion and the hydraulic fluid needs to be bio degradable. [65]

To operate a hydraulic cylinder a hydraulic system with different components is needed. The hydraulic system for subsea machines is a closed system which includes a pump, valves, and a reservoir or tank, and the cylinder itself as it can be seen in Figure 6.10. The pump, reservoir, and valves are combined the hydraulic power unit (HPU). The fluid is pumped from the tank to the cylinder following the red line in Figure 6.10, and the fluid on the side of the rod flows out of the cylinder back into the tank. If the reservoir would be located on the vessel, the pump would need to overcome the additional hydrostatic pressure for pumping the fluid up to the vessel. In 2000m water depth that are additional 200bar. The solution is to place a HPU on the machine itself. This is state of the art for ROVs with hydraulic systems. Allseas operates for example a WROV which has the optional IHPU which can provide 10kpsi which is approximately 690bar. It has hydraulic arms and can operate in water depth up to 2000m.

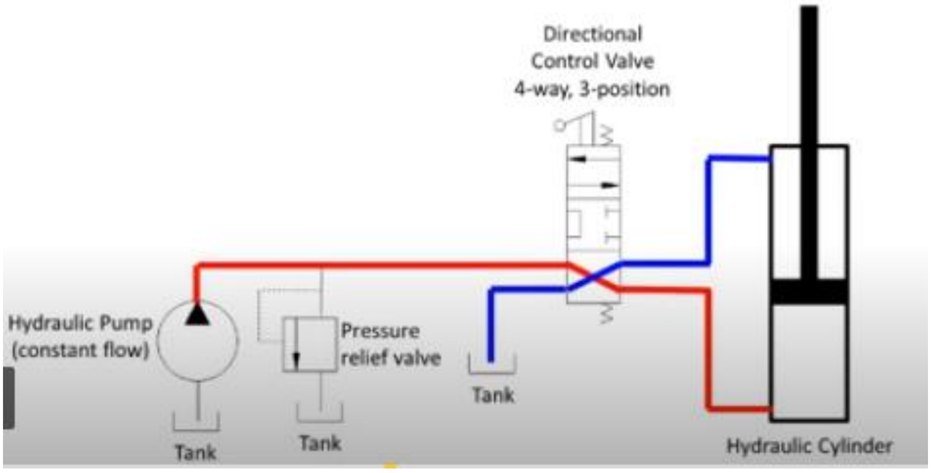
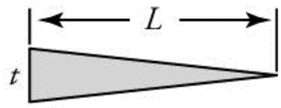


Figure 6.10: Schematic - Hydraulic System [66]

The hydraulic system is the preferred solution for the subfunction bending mechanism.

- **Screw press approach**

The second concept is to use the screw press principle which is shown in Figure 6.12. The screw in general has a higher mechanical advantage than the similar simple machine the wedge. This is because the screw can be seen as inclined plane wrapped around a cylinder. The mechanical advantage of a wedge is its length divided by its height as it can be seen in Figure 6.11 [67]. So the longer the wedge and the smaller the height the better the mechanical advantage and that is exactly how a screw is built.



$$MA = \frac{L}{t}$$

Figure 6.11: Mechanical Advantage of a Wedge [67]

The general idea how to implement this mechanism into the bending machine is to use a motor as power source which torque is then converted through a gear wheel into tangential force, turning the rim of the screw press. The screw subsequently translates the rotational motion into the vertical force required to bend the pipe.

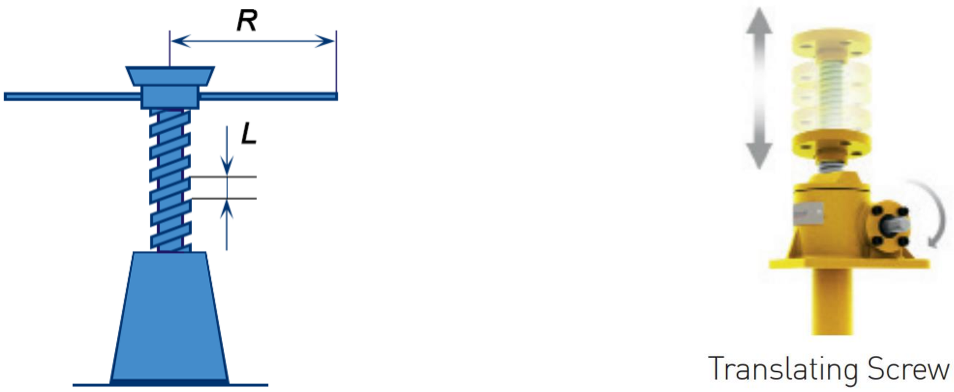


Figure 6.12a: Screw Press - Schematic [68] [69]

A company which offers screw jacks is Morskate. Their models are available up to a capacity of 2000kN on request. They can be combined in for example a t-configuration as it can be seen in the Figure 6.12b below, where the gearbox is located in the middle of two screw jacks.

Figure 6.12b: Screw Press - T-Configuration [69]

However, compared to the hydraulic system these screw jacks are very heavy. A comparable hydraulic cylinder with the capacity of 100ton and a stroke of 6" (150mm) weighs 61kg [70]. One Screw Jack with the capacity of 1000kN and a stroke of 150mm weighs 553kg including the gear box [69]. As there is such a size and capacity difference the screw jack solution is considered less desirable than the hydraulic system.

- **Rack and pinion**

The rack and pinion system is investigated as it is suitable for high forces. It is used for example for jack-up vessels to move its legs up and down such that it can stand on the seabed and the hull is lifted above the waterline. Racks are mounted on legs which are moved up and down over pinions which are powered by motors (Figure 6.13). As the force of the rack (F_r) is known as the force required for bending the torque which needs to be provided by a motor can be calculated as presented in equation 6.11.

$$T_p = F_r \cdot r_p \tag{6.11}$$

The following estimation of the torque according to equation 6.11 is performed to find the order of magnitude of the required torque and therefore a first approximation of the size of the system. The required force is obtained by dividing the bending moment from equation 6.7 by an assumed leaver arm of 3m. The resultant force is then 5335kN. When the radius of the pinion is 0.2m then the required torque according to equation 6.11 is about 1000kNm. From this equation it can be seen that the bigger the radius of the pinion becomes, the larger the needed torque when the force remains the same.

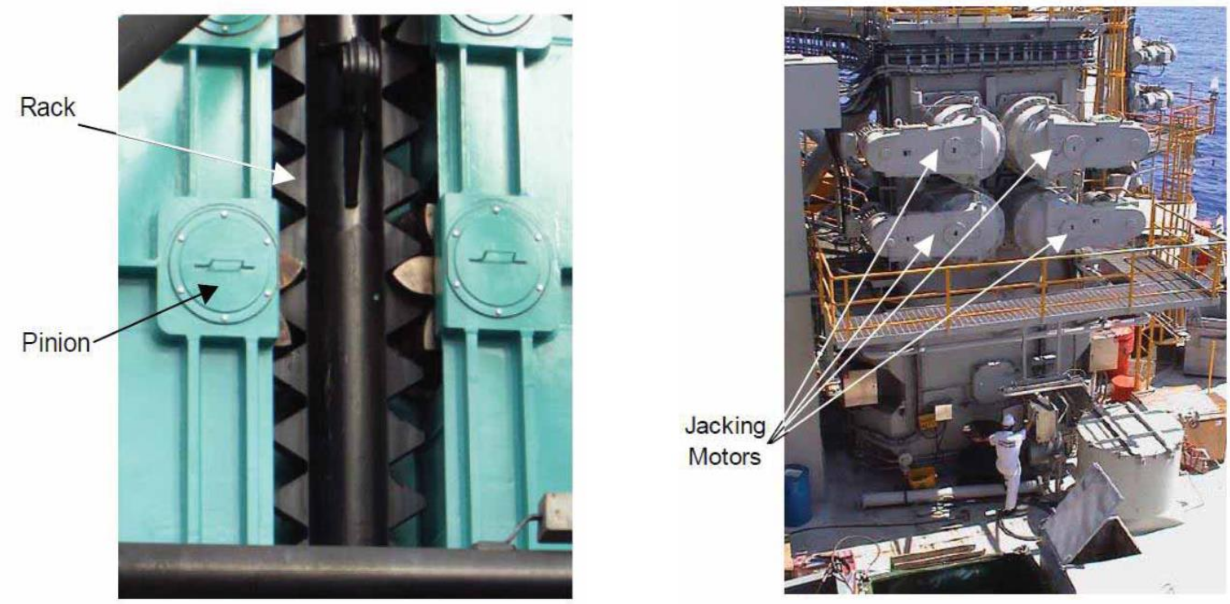


Figure 6.13: Rack and Pinion - Jacking System [71]

A jack-up drive system which is specifically designed for the purpose of powering jack-up legs has the capacity of 560ton [72] which is within the range of capacity what is needed as the 5335kN required bending force are equal to about 544ton. The reason why this solution is not considered feasible is the size of the system. As it can be seen in Figure 6.14 one of these drives is almost 3m long and 1.7m wide. Implementing these large drives into a design of a bending machine plus the motors that are needed additionally is not considered desirable especially in comparison to the much more compact hydraulic system.

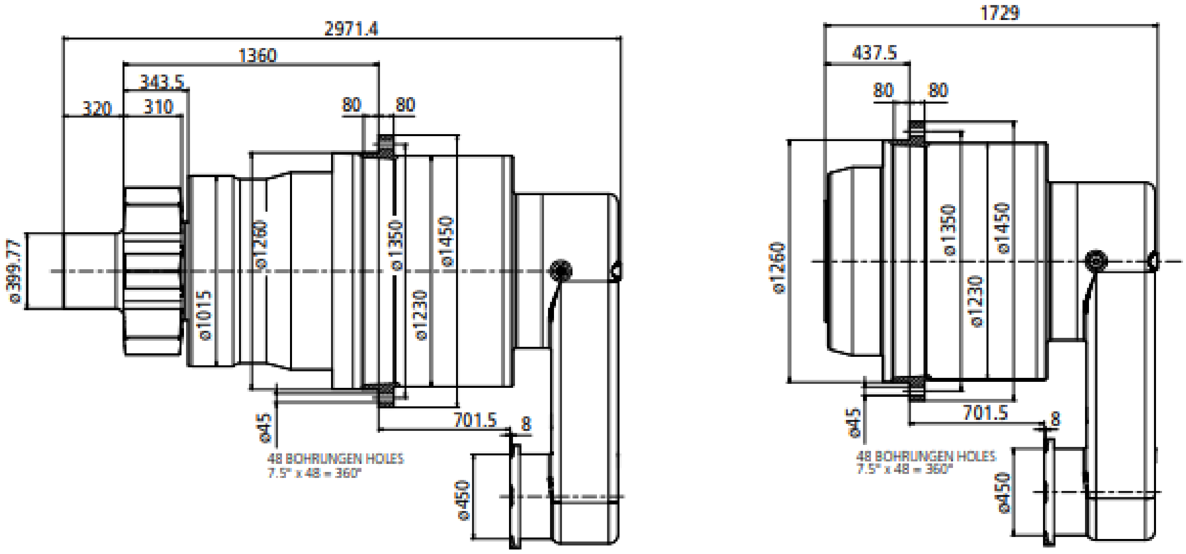


Figure 6.14: Jack-up drive – Overall Dimensions [72]

6.4.3 Clamping the Pipe and Bend Dies

Before the pipe is bent it is clamped between a upper and lower part which are on opposing sides of the pipeline. The clamp is the element of the machine which is located on the seabed side of the pipe. When the pipe is clamped the machine is fixed in position during bending.

- **Soft robotic arms**

One idea to clamp the pipe and hold the machine in place is to use soft robotic arms. There are different soft robotic gripper designs being described in published articles, and tested (Figure 6.15a and b). As this is a relatively new field of robotics, much research is still done for example in scaling up current designs. One example is the gripper design which uses a bellow soft actuator. The actuator is the blue element in Figure 6.15a which shows a hand like gripper with four “fingers”. The silicone actuator is hollow inside and when air or fluid is pressurized inside, the access material in the bellows unfolds which leads to the asymmetric motion to form around the object. The soft actuator can be reinforced in order to increase the operating pressure. Often soft robotics use principles which can be found in the nature like a gripper which is based on octopus tentacles (Figure 6.15b).

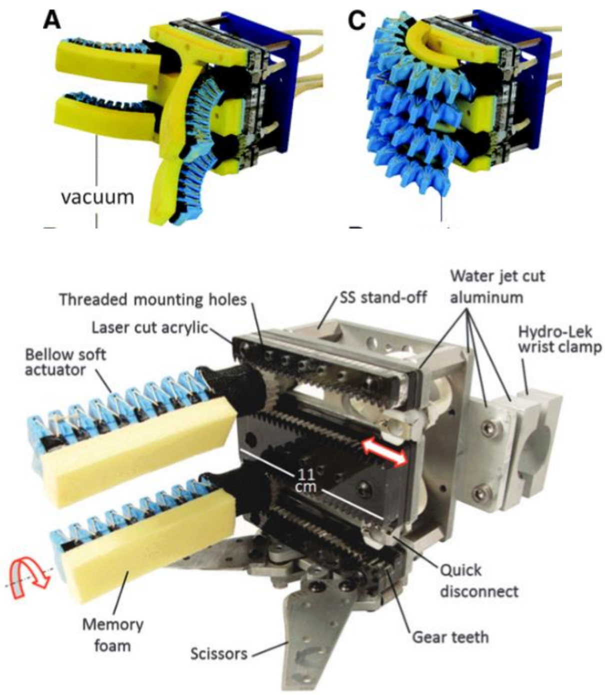


Figure 6.15a: Soft Gripper with Bellow Type Actuator [74]

Figure 6.15b: Tentacle Gripper [73]

The advantage of soft robotics is that these grippers can easily adapt to different shapes and sizes of the objects they are gripping [74]. In this case it means that when making use of this technology the clamp can adjust to different pipe sizes without the need to exchange the contacting element of the mechanical clamp.

As it can be seen in Figure 6.15 most of the available soft robotic actuators are of size on a centimetre scale, as the palm of the gripper hand is 11cm wide [74]. In a study from 2021 experiments were made to upscale soft robotics arms [75]. These are using fluid-driven origami-inspired artificial muscles (FOAM) actuators. These actuators use fluid instead of air and instead of pumping fluid in, to make the arm bend, a vacuum pressure is created. According to this study higher forces can be produced with this system. As it can be seen in the Figure 6.16 the elements have a length of 0.86m. In one of the experiments it is determined that these arms can bend by $30°$ and have a contractile force of about 194N which is about 19kg. These parameters show that also the upscaled soft robotic arms do not have the capacity yet to be used as clamp for a system concept for underwater pipe bending. To be suitable as a clamp the robotic arm should be capable to reach half way around a 32" pipe. This means that it needs to be able to bend $180°$ instead of only $30°$ and be 1.20m long which is almost twice as long as the arms in the study.

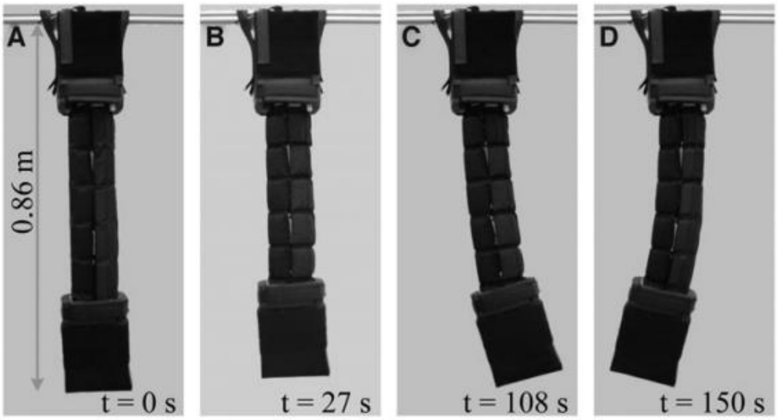


Figure 6.16: Upscaling of Soft Robotic Arms [75]

- **Slings**

An additional idea is to use slings as clamps like they are used as rigging to attach a load to a crane. These slings are made out of polyester which is a softer material than metal and therefore less likely to damage the coating. They are available with a high capacity as they are used for heavy lift operations as well. The company DAWSON offers for example round slings with a capacity of up to 1000ton [76]. According to the estimated 500ton from the rack and pinion solution for the bending mechanism (section 6.4.2) the slings capacity is sufficient to provide reaction force to the bending moment.

To implement slings as clamps on a bending machine, a ROV arm can be used to take one end of the sling, reach around the pipe and hook it into a shackle or something similar. There are ROV friendly hooks available which have handles such that they can be operated with a ROV arm (Figure 6.17).

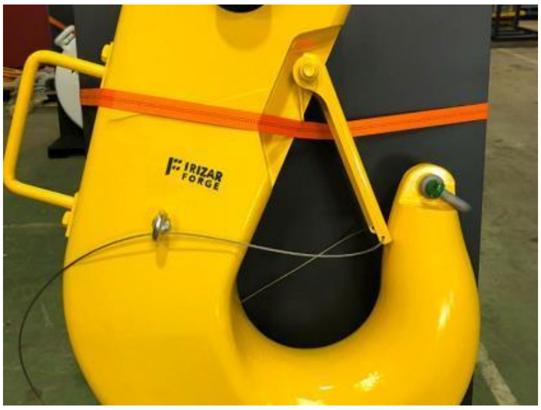


Figure 6.17: ROV Hook [77]

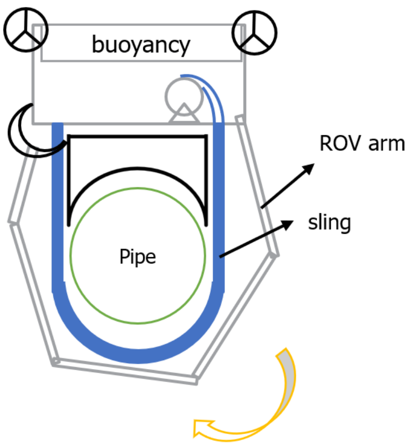


Figure 6.18: Schematic - ROV Concept with Sling

The slings are light enough to be manipulated by a ROV arm, a 10m long 500ton sling for example weighs about 53kg [76]. A manipulator which has a sufficient lifting capacity is the Titan 4 Manipulator [78]. It can lift 122kg at full extension and is 1.9m long. This length is not quite sufficient to reach around half the circumference of an 32" pipe which has the length of 1.27m plus the straight distance on both sides which are at least the radius so 2 times 40cm. The hook and the base of the arm can be however placed such that they are closer to the centre of the pipe when the machine extends down on both sides of the pipe.

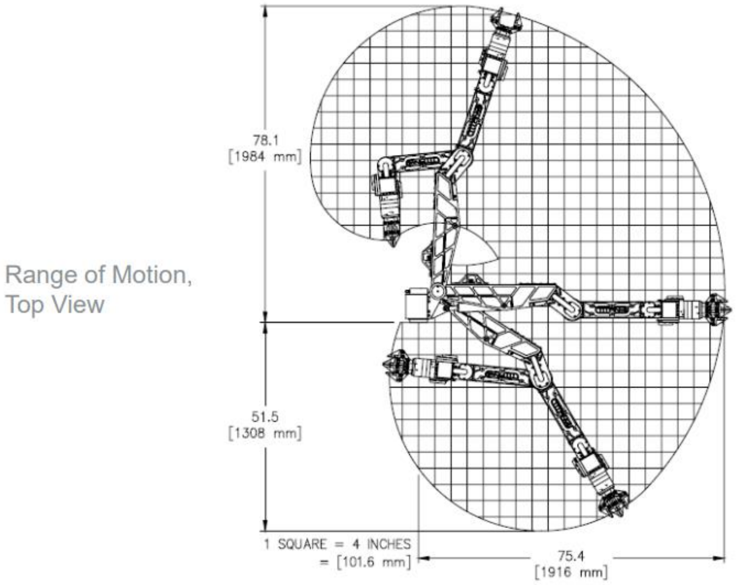


Figure 6.19: ROV Manipulator Titan 4 [78]

This option is ultimately dismissed because the range of motion of the ROV arm can be limiting. The pipe might be in the way when the arm tries to reach the hook on the opposing side of the ROV. One idea to overcome this difficulty is to customizing a ROV manipulator such that it has enough pivot points. The decisive reason why slings are not further considered as clamping option is that they elongate by 3% at rated capacity. For a 2.5m long sling, which is approximately the length that is needed if the arrangement of the elements is similar to the schematic in Figure 6.18, that is 7.5cm.

These 7.5cm have to be considered when sizing the hydraulic cylinder. When considering a bending machine which is 4m long which results in a stroke of hydraulic cylinders of 6cm then the additional 7.5cm mean that the required hydraulic cylinder needs to be twice as big. Therefore, it is assumed that a rigid clamp is the better alternative to this sling clamp.

- **Half shells**

A third idea is to use concave half shells as a clamp which inner diameter is equal to the pipe outer diameter. These elements are used in onshore vertical pipe bending machines as it can be seen in Figure 3.3. To make this clamp suitable for different pipe sizes exchangeable parts can be mounted onto the shell which have the outer diameter equal to the inner diameter of the shell and the inner diameter equal to the outer diameter of the pipe. Figure 6.20 shows a schematic of the clamp with the blue part which represents the exchangeable pads. This system is used for tensioners on pipelay vessels as well such that not the whole tensioner needs to be replaced when installing a new pipe size but only the pads on the track shoes (Figure 6.21a).

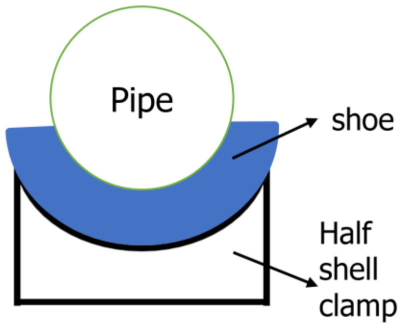


Figure 6.20: Half Shell Clamp

The half shell can also serve as a bend die when this element is used to distribute the force from the bending mechanism onto the pipe. Similar to when the half shell is used to clamp the pipe the half shell is formed concave for external system concepts. Half shells are used as bend dies for onshore vertical bending machines as introduced in section 3.2.1. For this machines the half shell bend die is used in combination with the bending mechanism of hydraulic cylinders which is also a possible combination for the underwater bending machine concept. Thus, this combination can be considered as proven concept.

- **Rollers / tensioners**

Another idea is to use rollers as a clamp which can be used for roll bending. However, the diameter of this roller needs to be really large, more than 2.36m, in order to have a sufficiently large contact area to prevent damaging the coating. The required area is estimated according to the allowable force per unit area as described in section 6.3. When the one side of the rectangular area is of length 0.45% of the total circumference of the 32” pipe

(1.15m) then its length is 1.8m when allowable force is 10N/mm². Taking this length as the required arc length of the roller the radius can then be calculated according to the following equation 6.12.

$$c = \alpha \cdot r \tag{6.12}$$

So assuming that the angle is $90°$, and the arc length is 1.8m then the radius of the roller is 1.18m which means a diameter of 2.36m. It can be noted that because of the relation in the above described equation the roller diameter becomes larger as the angle gets smaller.

An alternative to a simple roller is to use an element similar to a tensioner which are used for pipelay operations on the vessel during S-lay to hold the end of pipe on the vessel (Figure 6.21a). Connecting several rollers with a track increases the contact area of the clamp to the pipe. The length and the width of the tensioner can be sized such that the load distribution becomes acceptable according to the allowable pressure per unit area of the coating as presented in section 6.3.

One concept idea of using tensioners as bend dies is that when using two of them to roll over the pipe they can have the additional function for providing the bending force. Two tensioners are then placed on top of the pipe. One additional tensioner serves as clamp. The pipe is bent by adjusting the tensioners in orientation such that they are inclined and the outer rollers are forced down onto the pipe. The machine then rolls along the clamped pipe and the pipeline bends following the three-roll bending principle as introduced in section 3.1. A similar principle of roll bending with adjustable tensioners is used in the straightener system on Reel-lay vessels [79] as it can be seen in Figure 6.21b.

Figure 6.21a: Tensioner [2]

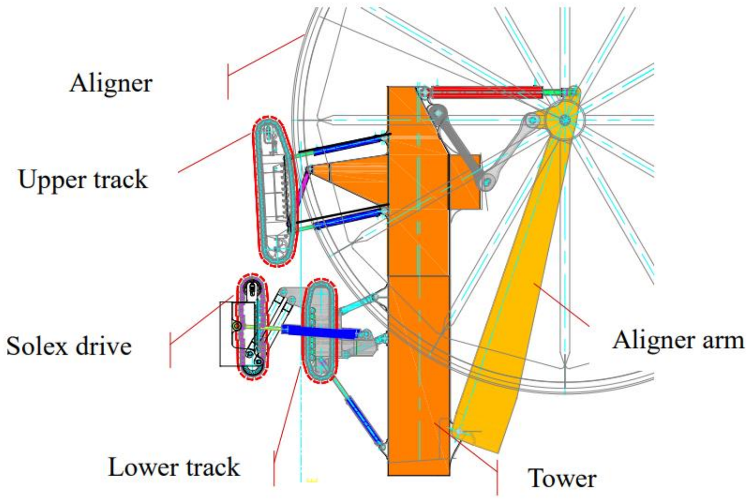


Figure 6.21b: Tensioners in Reel-lay Straightener System [79]

The tensioner as bend die or clamp can also be combined with rigid clamp or bend die options. Then the bending process is again producing little bends in a sequence. The tensioner provides the additional function of moving the machine from bend to bend.

To investigate if the tensioner as element is a feasible combination with the machine body that is deployed over the stinger from the pipelay vessel, the required height of the tensioner is estimated by comparing the bending stiffness of the pipe to the stiffness of the body of the tensioner. The body of the tensioner is designed as a rectangular shaped cross-section. The bending stiffness is EI, the Young's modulus multiplied with the moment of inertia. As the Young's modulus for steel is the same for the pipeline and the material of the machine, it is sufficient to only compare the moment of inertia.

The tensioner is modelled as a box with vertical plates evenly distributed inside it, to improve the stiffness of this body. The dimensions of the box are selected such that the tensioner fits through the envelope given for the *Lorelay* when deploying the machine over the stinger according to functional requirement 1.3. The selected size of the box is therefore 790mm x 500mm with three vertical plates inside whereas the thickness of all plates is assumed as 35mm. The resulting moment of inertia is significantly lower, 5.8×10^9 mm^4, than the moment of inertia of the 32" pipeline, which is 7.1×10^9 mm^4.

A different approach to build the tensioner body is to assume two standard I-Beams as basis which are welded together such that they build a box shape. According to the moment of inertia of the cross-section, a HEM800 profile with a height of 814mm provides sufficient stiffness. The height of this cross-section is larger than the limiting height of 573mm for the total machine. Both approaches show that a machine with a tensioner of sufficient stiffness is too large to be deployed over the stinger of the *Lorelay*.

6.4.4 Locomotion System of the Machine

The machine once attached to the pipe shall be able to move along it. This is necessary because the total bend is achieved by introducing little bends in a sequence in the pipe joint such that the pipe and coating are not damaged.

- **Tracks**

Tracks can be used for the external machine options as well as internal bending machine. As the internal bending concept is dismissed, only external tracks are presented here. These tracks are similar to tensioners which are introduced in section 6.4.3. The difference between the tensioner on the vessel compared to the ones part of a bending machine is that the tensioners in the firing line are stationary and the pipeline moves past them. Now the tensioner is used to move the machine along the pipeline.

Tracks or tensioners can fulfil different functions. Tracks might be used as a clamp and as a solution to move from bend to bend at the same time. Another possible combination is to have a track or tensioner element which moves the machine over the pipe and serves also as bend die.

- **Thrusters**

A common propulsion for ROVs are thrusters. An example of a hydraulic thruster is shown in Figure 6.22. These are reliable components which can be mounted on the ROV concepts and concept for which the machine is deployed over the pipeline from the end of the firing line. ROVs can fly to any direction. This means that a machine manoeuvred with thrusters can fly from bend section to bend section and that the orientation of the machine on the pipe can be adjusted by using the thrusters as well.

Figure 6.22: Hydraulic Thruster SA420 [80]

However, once the machine arrives at a section which is supposed to be bent the machine attaches to the pipe with clamps. The total bend in one section needs to be carried out as little bends in a sequence, or using roll bending, to not damage the coating and meet the limiting strain of 1.25%. Therefore, the machine needs to be able to move along the pipe within this bend section. Because of pipe movements and ROV movements it is considered faster and less risky to damage the pipe coating when the machine only engages to the pipeline once and then rolls or crawls from bend to bend until the total bend is completed. Therefore only thrusters as locomotion solution are dismissed.

When using thrusters the positioning needs to be considered carefully in the more detailed design as it influences the stability of the ROV concepts. The required capacity is dependent on the drag force on the machine and the umbilical caused by current.

- **Self-folding propellers**

This solution to manoeuvre a machine is suitable for any concept which assumes that the machine rolls over the pipe from the vessel to the sagbend. The idea is that instead of normal thrusters which are a reliable propulsion for under water, it is made use of a different kind of propellers. These self-folding propellers as depicted in Figure 6.23 can be used underwater as well as in air. At the top of Figure 6.23 it can be seen that the propeller blades are folded under water and expand in the transition between water and air until they are completely unfolded in air. The propeller unfolds because of the of the increased rotational speed in air compared to water which increases the centrifugal force such that the propeller unfolds. When transitioning from air to water the propeller passively folds inwards as the fluid force pushes the blades inward.

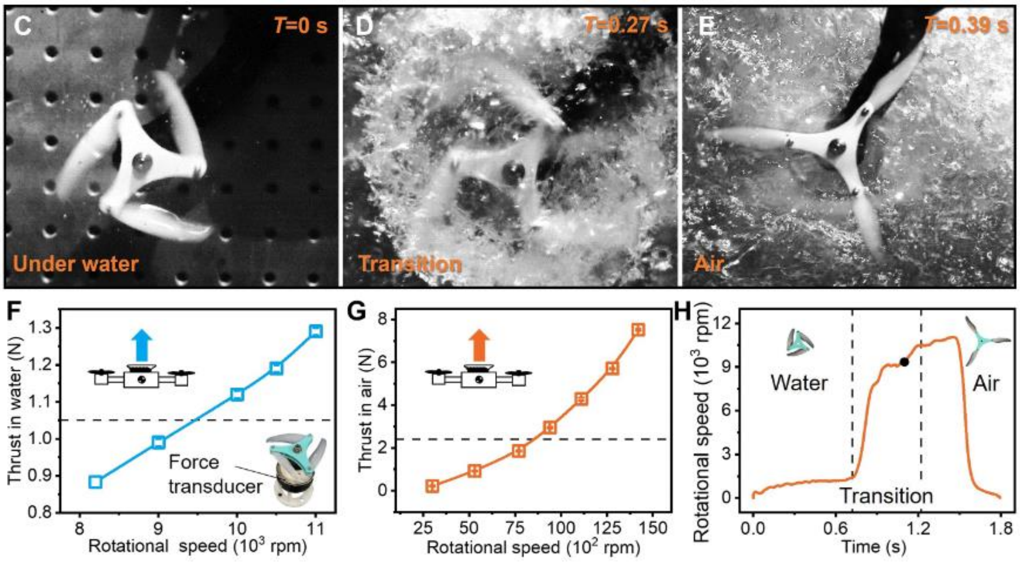


Figure 6.23: Self-folding Propellers [81]

Additionally to using these different propellers, it is proposed to use these in a frame similar to the frame which is mounted on the VertiGo robot which can drive on vertical walls as well as horizontal ground (Figure 6.24). The propellers can be adjusted in direction such that they create downward pressure to press the robot against the wall while the second propeller additionally creates the thrust to move the robot forward.

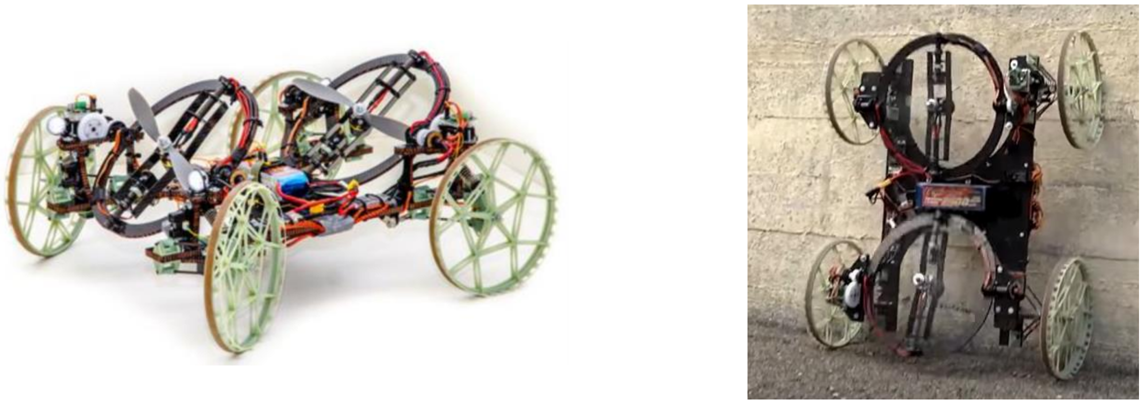


Figure 6.24: VertiGo - Robot Driving on Vertical Walls [82]

The idea for the pipe bending machine is to combine these two components, the self-folding propeller and the direction adjusting frame of the VertiGo robot. This makes it possible to control the machine on the pipe in roll direction as well as moving it along the pipe. This is the case when the machine is submerged as well as when it is on the stinger. The self-weight of the machine on the pipe can also be reduced by creating an upward thrust of the unfolded propellers in the unsubmerged part of the pipeline.

To check if the reduction of weight of the machine in air justifies the use of this more complex propulsion system a simulation is done in OrcaFlex. This analysis is similar compared to the one done in section 6.4.1 to determine the maximum allowable weight of the machine on the pipeline. The vessel and water depth of 2000m are the same, as well as the 16'' pipeline. The machine is modelled as a 6m long stiffener with a weight of 0.82ton/m which resembles a total weight of 5ton, as well. However, in this simulation the machine is modelled on the stinger instead of in the sagbend region. The buckling check of the load controlled as well as the displacement controlled condition and the strain in the pipeline are analysed. The load conditions which are characterised as displacement controlled in the limit state design allow for exceedance of yield strength under certain circumstances, if all relevant limit state conditions are acceptable for the applied load. This buckling check is used for the overbend region within Allseas. The load controlled buckling check is performed for the sagbend region. In Figure 6.25 the results of this analysis are presented. It can be seen that the strain in the pipeline is the most critical value, compared to the buckling checks, as it is closed to the limiting value of 0.35%. However all three parameters are acceptable over the total pipeline length. Additionally, in the bottom right drawing it can be seen that most of the stinger is submerged. That means that buoyancy can also be used to reduce the weight of the machine when it is on the stinger not only on the submerged pipeline. Because of these two findings it is concluded that a weight reduction by propellers in air is not necessary.

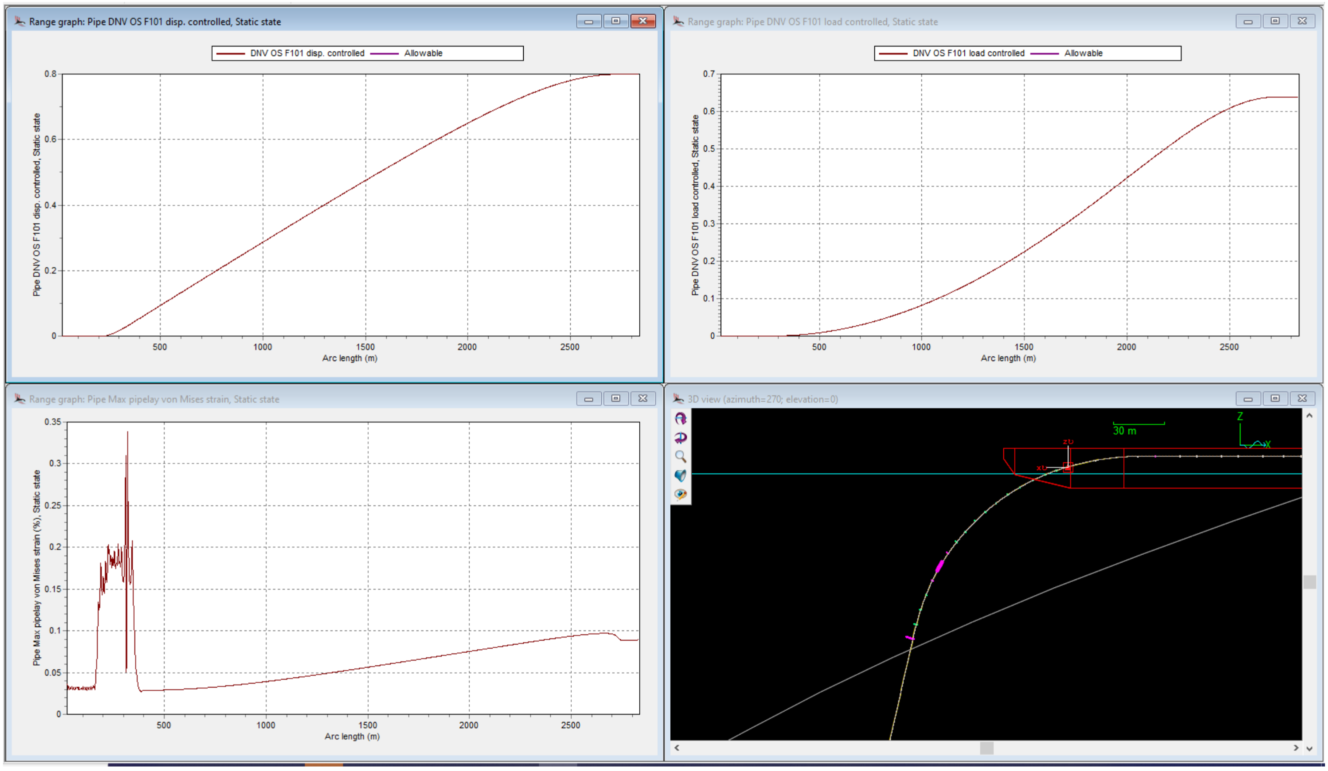


Figure 6.25: OrcaFlex Simulation - Machine on Stinger

- **Handover clamps**

This locomotion system is only compatible with half shells as clamps. For this system these clamps clamp around the total circumference of the pipe. The outer clamps only open and close, whereas the middle clamp additionally needs to be able to move forward and back, in direction along the pipeline, between the outer clamps. The two outer clamps fulfil the function of the bend dies, and the middle clamp provides the reaction force to the bending force.

The working principle of the machine movement can be described as follows: All three clamps are closed for bending, then the middle clamp opens and moves towards the outer clamp, in the direction of movement of the machine. There it closes and the outer clamps open. Then, while being closed, the middle clamp moves back against the direction of movement of the machine and pulls the machine along the pipeline forward. The outer clamps close again and the middle clamp can move to its initial position. This sequence is repeated until the machine reaches the new bend position.

Although this mechanism is not used in context of pipe bending machines, there are some examples of machines which use this principle for moving the machine along a pipe or move a wire with the machine. A first example introduced here is a pipe climbing robot, a climbing crane with which onshore wind turbines are build. As shown in Figure 6.26 the tower of these wind turbines are composed of a number of elements which are mounted on top of each other. As the tower grows the crane climbs up such that it can lift the next element on top of the tower. The total weight of this crane moving along the pipe is 270 tons. It can be noted that the clamp force is not only transferred to the tower element by friction and contractile force. In this case there are holes in every tower element into which small cones, mounted on the outer clamp, lock to secure the crane in its working position. As the bending machine is under water and buoyancy can be used to reduce the weight this additional locking mechanism is considered not needed. It is also not feasible as holes in a pipeline make no sense.

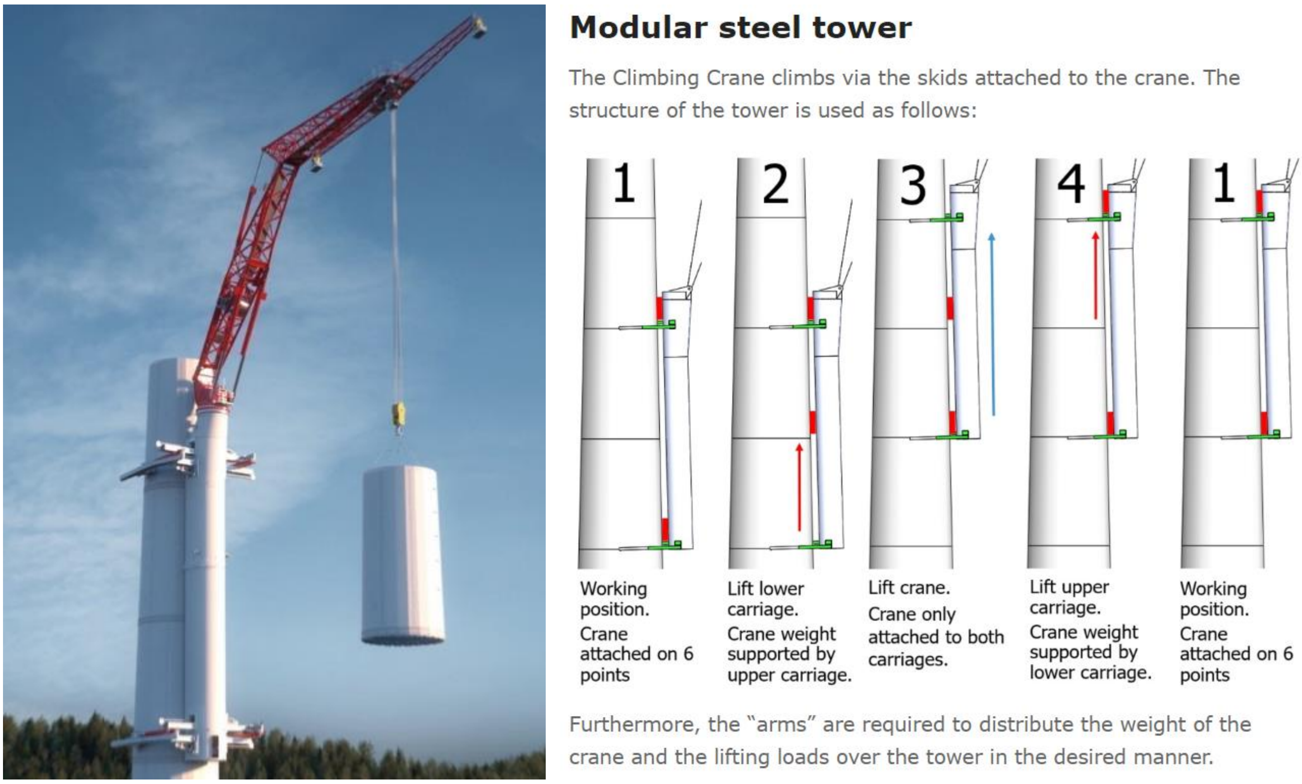


Figure 6.26: Climbing Crane [83]

A second example where this "handover" principle is used is the linear winch. There are two different working principles for these winches. One is a intermitted version, which means that the wire is clamped, then pulled a bit after which the clamp moves to its initial position to perform the next pull. With two clamps which are moved in the frame by hydraulic cylinders it is possible to pull the winch wire in a continuous motion. These winches have the capacity of up to 800ton of pulling force the maximum wire size is then 152mm and the maximum pulling speed 3m/min [84]. Faster linear winches from a different manufacturer have a maximum pulling speed of 6m/min at the same pulling capacity [85]. The large pulling force of the winch is transferred to the wire over the wedge formed clamps by friction. Additionally to the winch itself which is shown in Figure 6.27 a HPU unit with an electric motor, hydraulic tank, and a control unit, is required to provide power for the hydraulic linear winch.

Figure 6.27a: Picture of Linear Winch [85]

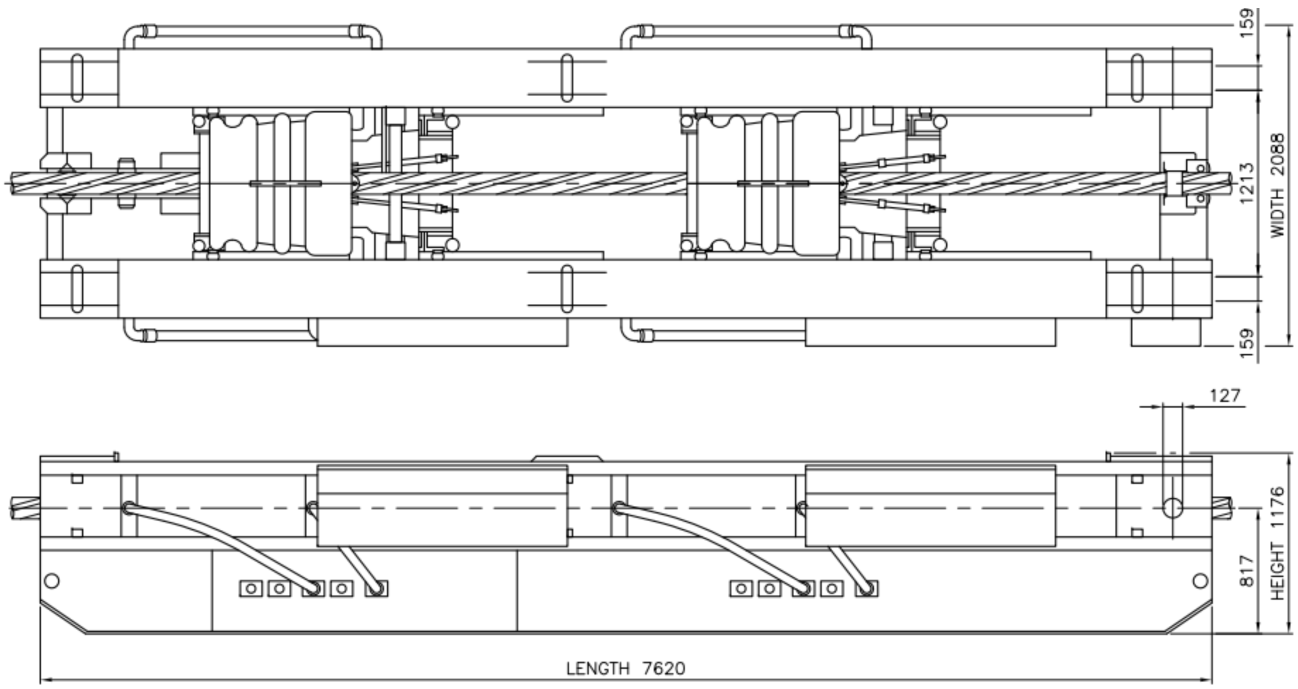


Figure 6.27b: Continuous Linear Winch 300ton [84]

6.4.5 Conclusion of Solutions of Subfunctions

A summary of the different solutions for the subfunctions identified can be found in Table 6.1. When combining one solution of each subfunction a variety of different combinations of elements of the bending machine can be found as it can be seen in the following in Table 6.2. These are the twelve combinations of subfunctions which are considered feasible according to the calculations and considerations presented in the previous section.

In this table the bending mechanism as identified subfunction is not listed as the only reasonable option identified is the hydraulic cylinder to provide the bending force. It is therefore the same for all the combinations. Further, can be seen that the first 5 concepts compared to concept 6 to 10 do have the same possible combinations of clamp, locomotion system and bend die. The only difference between these groups of five is that the ROV approaches from above in the first five concepts whereas it approaches the pipe from the side for the second five concepts.

Table 6.2: Twelve Concepts for Trade-off Analysis

	Body	**Clamp**	**Locomotion**	**Bend Die**
1	ROV approaching from above	tensioner	thruster and rollers	tensioner
2	ROV approaching from above	tensioner	thruster and rollers	half shell
3	ROV approaching from above	half shell	handover clamps	half shell
4	ROV approaching from above	half shell	thruster and rollers	tensioner
5	ROV approaching from above	half shell	thruster and rollers	half shell
6	ROV approaching from the side	tensioner	thruster and rollers	tensioner
7	ROV approaching from the side	tensioner	thruster and rollers	half shell
8	ROV approaching from the side	half shell	handover clamps	half shell
9	ROV approaching from the side	half shell	thruster and rollers	tensioner
10	ROV approaching from the side	half shell	thruster and rollers	half shell
11	rolling on pipe	half shell	thruster and rollers	half shell
12	rolling on pipe	half shell	handover clamps	half shell

6.5 Trade-off Analysis

The trade-off analysis is used to find the top three concepts out of the remaining twelve as shown above. Two tools are combined to find the weights for the criteria and to calculate the final score of each concept. The weights are found by using the pairwise method and the scoring is done using a formal trade-off analysis [4].

6.5.1 The Criteria

The following eight criteria are defined to rate the concepts [4].

A) Effort adapting to different pipe sizes 16'' to 32''
Scored according to the number of exchanged parts like for example tensioner pads.

B) Risk of damaging the pipe
This criterion focuses on the part of the bending process when the machine is deployed. It is evaluated how large the risk is to add additional strain to the pipe or that the pipeline buckles due to weight of the machine.

C) Risk of damaging the coating
This criterion evaluates if the coating is damaged easily when the machine approaches the pipeline or when it moves from bend to bend. Vessel movements or current also influence the deployment of the machine and therefore the risk of damaging the coating.

D) Risk of new technology
The risk is low when the elements or combination of elements are already used for pipe bending machines.

E) Risk of violating a patent because the machine is similar
Scored according to number of similar elements/ similar combination of elements compared to patented concept.

F) System robustness
Scored according to number of fragile elements or connections. The more simple the system concept the higher the score.

G) Maintenance
Scored according to the number of maintenance intensive elements. The less maintenance the better.

H) Ease of transport and storage
Rated according to estimated size. Larger and heavier machines are more difficult to handle.

6.5.2 The Pairwise Method

To find the weights for the trade-off analysis the pairwise method is used. In a matrix the criteria are compared to each other. As it can be seen in Table 6.4 the criteria are listed in the first row as well as in the first column in the same order. To each cell are values assigned to indicate if a criterion in a row is more or less important than the criteria in the columns of that row. The cell where the same criterion is written in the row and column is given the value 1. All other cells are filled in with values from 1 to 5 and from 1/2 to 1/5 according to the following rating scheme presented in Table 6.3.

Table 6.3: Pairwise Method - Rating Scheme

more important	
1	same
2	slightly more important
3	moderately more important
4	a lot more important
5	way more important

less important	
1	same
1/2	slightly less important
1/3	moderately less important
1/4	a lot less important
1/5	way less important

As it can be seen in Table 6.4 the matrix is anti-symmetric to its diagonal. This means that if the first row criterion A is way less important than B it is rated 1/5 and the mirrored cell about the diagonal gets the reciprocal value which is 5 in this example. The weight of the criterion is then calculated as percentage such that the sum of all weights of all criteria is equal to 100. The table of the pairwise method and therefore the rating and weights of the criteria can be seen in Table 6.4 below.

Table 6.4: Pairwise Method – Comparison Chart

Criterion	A	B	C	D	E	F	G	H	**geometric mean**	**fraction**	**percentage/ factors for MCDA**
A	1	0.20	0.20	0.50	2.00	0.25	0.33	1.00	0.44	0.05	4.64
B	5	1	2	4	5	4	5	5	3.27	0.34	34.24
C	5	0.5	1	4	5	4	5	5	2.68	0.28	28.09
D	2	0.25	0.25	1	3	0.5	0.33	5	0.67	0.07	7.05
E	0.50	0.20	0.20	0.33	1	0.33	0.25	0.25	0.34	0.04	3.59
F	4.00	0.25	0.25	2.00	3.00	1.00	3	5	1.24	0.13	12.98
G	3.00	0.20	0.20	3.00	4.00	0.33	1	5	0.90	0.09	9.43
H	1.00	0.20	0.20	0.20	4.00	0.20	0.2	1	0.39	0.04	4.04
										1.00	100.00

The rating of the concepts are the result of a discussion with a senior installation engineer which have been subsequently reviewed by a senior research and development (R&D) engineer. These two engineers gave their input for the trade-off analysis as well presented in the following section 6.5.3. By exchanging different point of views and arguments the values presented above have been agreed upon.

6.5.3 The Top 3 Concepts

In the trade-off analysis the different concepts are rated according to the criteria presented in section 6.5.1. Each concept is given a value for each criterion as it can be seen in Figure 6.28. The values are ranging from 1 to 5, whereas 5 means superior and 1 means poor. To achieve a more reliable result the scoring is done additionally by two engineers who have a different perspective on the concepts and expertise in this field. Engineer R is a senior installation engineer with 12 years offshore experience as field engineer who has a more practical point of view. Engineer A is a senior R&D engineer who has experience with new concept developments. Including these two ratings three different trade-off analysis with slightly different ratings are evaluated.

By multiplying the values with the weight of the criterion, determined with the pairwise method, the weighted score of all the criteria is calculated like it can be seen in Figure 6.28. The sum of these weighted scores and the geometric mean of the weighted scores for each concept are the values which determine the top 3 concepts of each of the three trade-off analyses. The results of the individual trade-off analyses are shown in Appendix F.

For each alternative ...			
Selection criteria	Weights	Value	Score = weight × value
1	w_1	v_1	$w_1 v_1$
2	w_2	v_2	$w_2 v_2$
3	w_3	v_3	$w_3 v_3$
4	w_4	v_4	$w_4 v_4$

Figure 6.28: Weighted Sum Integration of Selection Criteria [4]

A sensitivity analysis is performed to confirm the results of the trade-off analyses. It shows that the highest scored concepts do not change when changing the weight of each criterion sequentially and recalculating the study. The weights are changed to 20% above the original weight and 20% below the original weight. This percentage is chosen as variations of about 20% should be considered because of uncertainties in the assignment of weighting and scores [4].

The concept which has clearly the highest values for the score and the geometric mean in all of the trade-off analyses is concept 5. To find the top 3 of the highest scoring concepts the results are analysed in more detail in Table 6.5 below. These 6 concepts are the once which represent the three highest scores of the three different trade-off analysis. Which combination of the different solutions for the subfunctions these 6 concepts are, can be seen in Table 6.2.

Table 6.5: Evaluation of Trade-off Analyses Results

concept number	score			geometric mean			total score	total mean
	s	r	a	s	r	a		
2	384.88	0	0	26.68	0	0	384.88	26.68
3	443.05	437.6	0	36.58	38.57	0	880.65	75.15
5	458.5	471.15	417.08	37.07	39.03	38.68	1346.73	114.78
10	0	423.03	388.99	0	42.45	36.77	812.02	79.22
11	372.7	328.88	422.89	37.07	39.03	35.29	1124.47	111.39
12	0	0	486.71	0	0	41.57	486.71	41.57

The different highest scores of the concepts are listed on the left. Next to the score the highest geometric means are listed. In yellow and green respectively are the three highest of these values highlighted. On the right the sum of the scores and the sum of the geometric means are listed for the concepts. It can be seen that concept 5 has the highest total score and the highest total geometric mean. Second place as it can be seen on the right side of the table is concept 11.

Looking at the total geometric mean concept 10 would be a good choice as a third concept to look into. But the only difference to concept 5 is that the ROV approaches from the side instead from above. Concept 5 and 11 are both concepts with two half shells as clamps and bend dies their locomotion system is composed of thrusters and rollers. To look into a different system concept as well in the Concept Definition phase, concept 12 is selected as it is a concept which has the handover system as locomotion. This concept has the highest score of the individual trade-off analyses. Additionally, it is the only concept next to concept 5 which has the highest score and geometric mean at the same time in the individual trade-off result as it can be seen in the last row in the left parts in Table 6.5.

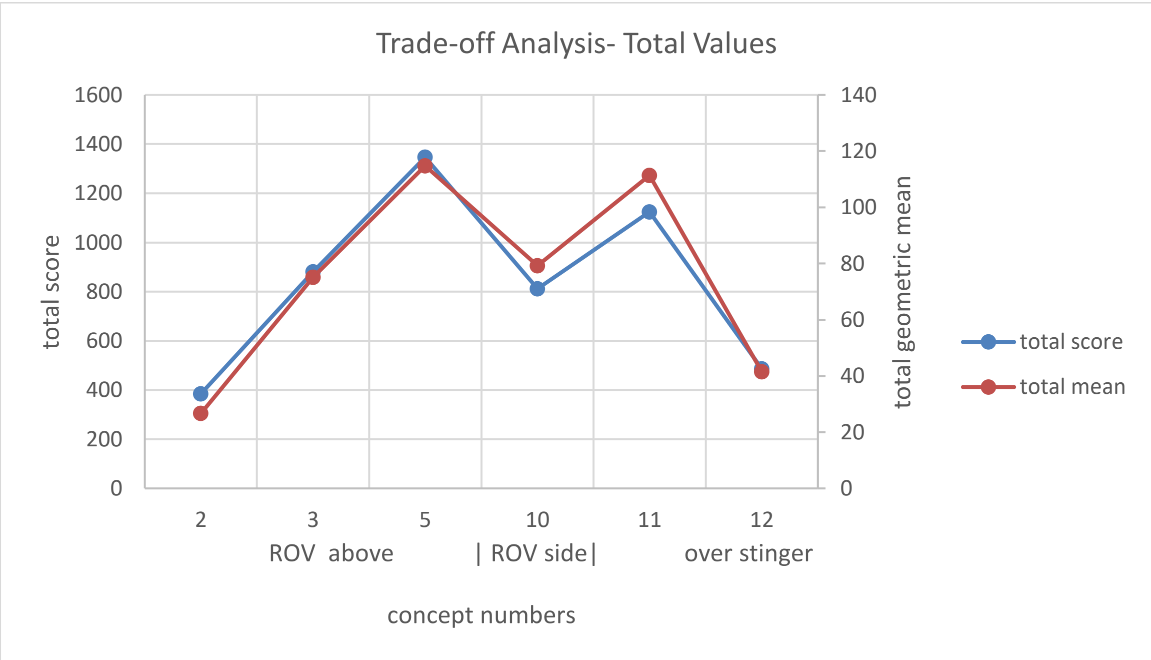


Figure 6.29: Trade-off Analyses - Total Values

It can be noted that all the concepts which have a tensioner as element do not score very high because tensioners are heavy compared to half shells, they consist out of many elements which makes them the least simple solution. It takes also considerably longer to change all the pads on the tensioner shoes compared to less pads on the half shells to adjust to different pipe sizes.

Further it can be noted that the concepts based on the ROV approaching from the side are the less favourable solution compared to the ROV solution approaching the pipeline from above. Figure 6.29 shows that 3 concepts of the ROV solution from above are scoring high compared to one concept of the ROV body approaching from the side.

In the following the three different concepts with the best scores are presented:

1. Concept 5: ROV from above with half shells
2. Concept 11: rolling over pipe with half shells
3. Concept 12: rolling over pipe with handover clamp

6.6 Concept 5

The idea of this system concept is to design a ROV which bends the pipeline with a similar principle like the onshore vertical pipe bending machines. Table 6.6 shows the path in the morphology matrix which combines the solutions of the subfunctions for this concept.

Table 6.6: Morphology Matrix - Concept 5

		solutions					
subfunctions	*approaching pipe/ deployment to bend section*	from pipelay or support vessel: ROV approaching from above	from pipelay or support vessel: ROV approaching from the side	*(from pipelay vessel: internal / inside the pipe)*	from pipelay vessel: over stinger/ roll over pipe		
	clamping (below pipe/ inside)	*(soft robotics)*	mechanical outside bent half shell	*(mechanical inside, like mandrel)*	*(roller)*	like a tensioner	Slings
	actuator/mechanism for bending	hydraulic cylinder	*(screw press solution with motor)*	*(rack and pinion)*	*(wedge with horizontal cylinder)*		
	manoeuvre the machine from bend to bend	tracks (similar to tensioners or tank tracks) on/in pipe	(thrusters)	both tracks and thrusters	self-folding propellers and tracks	Handover clamps	
	bend die / contact element (above pipe)	like a tensioner	like a half shell convex or concave	*(roller)*			

This system concept uses the mechanical advantage of the leaver arm. With a leaver arm of 1.5m to each side the force which needs to be provided by the actuator, hydraulic cylinder, is 5335kN which is equal to 543ton. A schematic of a first design of this concept is shown in Figure 6.30.

The ROV can be deployed from the pipeline installation vessel as well as a support vessel. It is lifted with one of the available cranes and flies then down to the sagbend. It is continuously connected to the vessel over an umbilical. Once it reaches the bend section it approaches the pipeline from above. The clamp, 5, which is a concave formed half shell moves up to clamp the pipe between the lower clamp, 1, and the upper half shells which are the bend dies, 5. The thrusters, 3, keep the machine in the correct orientation. The ROV concept creates a number of bends in a row to achieve the total required angle.

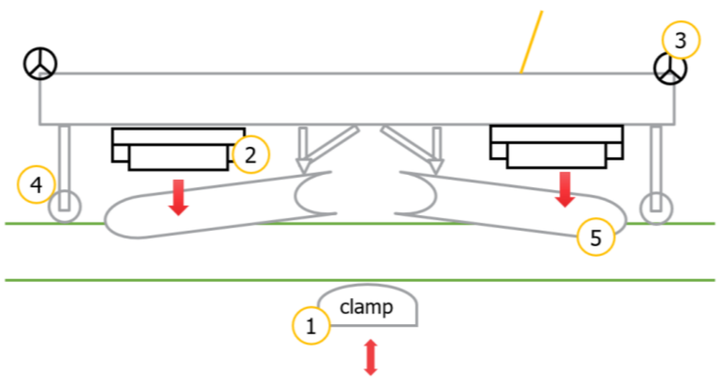


Figure 6.30: Schematic – Concept 5

The needed bending force comes from a hydraulic cylinder, 2. To navigate from bend to bend within the same bend section rollers, 4, extent onto the pipe to maintain the contact between the machine and the pipe to avoid the risk damaging the coating. This can happen when the ROV collides with the pipeline due to the movement of the pipeline due to vessel motions.

The angle of these little bends depends on the distance between the bends and the bending radius. Assuming a bending radius of 40D and a distance between the bends of 30cm the angle per bend is $0.52°$. The 30cm distance is chosen based on the recommendation of the onshore vertical bending machines [30]. The total angle per pipe diameter length is still below 2.0° as it is stated in the coating requirements. The required angle is produced by creating a number of bends in a row. The total achievable angle per 12m joint is about 17° for a pipe size of 32" when considering the required distance to the welds of 1.5D [8].

As stated in section 6.4.1 there are different options how to build this ROV which are designed in detail in the next phase the Concept Definition. If the ROV is light enough in water then it is possible to fly it around in the water in any direction. Then the machine is a new ROV with all necessary sensors and power systems on the body. There is also the option to design the tool and instead of designing a new ROV body the AUXROV can be used as this element has all the power supply, thrusters and sensors needed. This AUXROV can be simply connected to the tool with a docking plate.

Depending on the submerged weight, it needs to be considered if a support vessel is needed. If the machine gets too heavy it is the better solution to lower it with a crane directly over the bend section where only the orientation needs to be slightly adjusted.

6.7 Concept 12

One version of the machine which can be deployed from the pipelay vessel is this concept which uses the "handover" locomotion system. Concept 12 which scored high in the trade-off analysis is the combination of solutions of subfunctions presented in Table 6.7.

Table 6.7: Morphology Matrix - Concept 12

		solutions					
subfunctions	*approaching pipe/ deployment to bend section*	from pipelay or support vessel: ROV approaching from above	from pipelay or support vessel: ROV approaching from the side	*(from pipelay vessel: internal / inside the pipe)*	from pipelay vessel: over stinger/ roll over pipe		
	clamping (below pipe/ inside)	*(soft robotics)*	mechanical outside bent half shell	*(mechanical inside ,like mandrel)*	*(roller)*	like a tensioner	Slings
	actuator/mechanism for bending	hydraulic cylinder	*(screw press solution with motor)*	*(rack and pinion)*	*(wedge with horizontal cylinder)*		
	manoeuvre the machine from bend to bend	tracks (similar to tensioners or tank tracks) on/in pipe	(thrusters)	both tracks and thrusters	self-folding propellers and tracks	Handover clamps	
	bend die / contact element (above pipe)	like a tensioner	like a half shell convex or concave	*(roller)*			

This machine can be clamped to the pipeline on the vessel with the half shell clamps and bend dies. As there is no space on the *Lorelay* to set the machine on the pipe after the last tensioner, the machine is lifted from the deck with the SPC on the *Mercedes* trolley. The trolley rolls the machine inside close to the beadstall. With the two overhead cranes the machine is lifted onto the pipeline after the first tensioner. Like an inline structure it then passes all the following stations in the firing line. Eventually, the machine is guided over the stinger like an inline structure as it can be seen in the Figure 6.31.


Figure 6.31: Inline Tee on Stinger [14]

The advantage of using the half shells for clamping is that a similar half shell system is used to clamp the roller guides to the pipeline. In Figure 6.32b there are elements like roller guides on each side of the pipeline to stabilize the machine against rolling on the stinger if needed. Once below water thrusters, 3, can be used to keep the machine in the right orientation relative to the pipeline. A more robust solution is to use buoyancy to keep the machine upright under water. The locomotion system is then used to move the machine to the sagbend as well as from bend to bend in the bending section as the bending principle on how to introduce the bends into the pipeline is similar as previously introduced in the ROV concept. The middle clamp, 1, moves in horizontal direction to fulfil the function for the locomotion system, at the same time it acts as clamp during bending to provide the reaction force. The outer clamps, 4 and 5, have two functions as well. They are serving as bend dies and part of the locomotion system. The bending force is provided by hydraulic cylinders, 2. This locomotion system can then be used to retrieve the machine after the bend is completed such that it can climb along the pipe back onto the vessel.

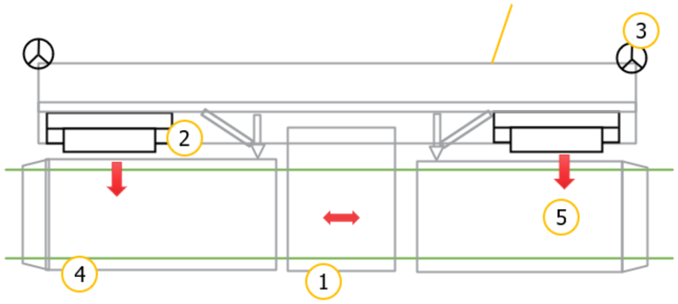


Figure 6.32a: Schematic – Concept 12 Side View

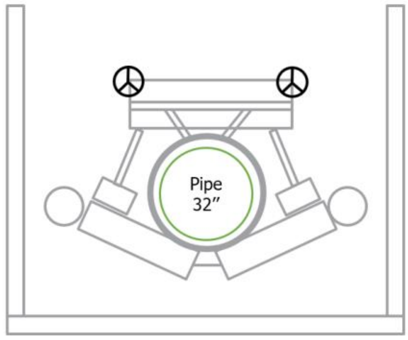


Figure 6.32b: Schematic - Concept 12 on Stinger

6.8 Concept 11

Similarly to the two previous concepts, this concept can be presented as path in the morphology matrix as well, as show in Table 6.8.

Table 6.8: Morphology Matrix - Concept 11

		solutions					
subfunctions	*approaching pipe/ deployment to bend section*	from pipelay or support vessel: ROV approaching from above	from pipelay or support vessel: ROV approaching from the side	*(from pipelay vessel: internal / inside the pipe)*	from pipelay vessel: over stinger/ roll over pipe		
	clamping (below pipe/ inside)	*(soft robotics)*	mechanical outside bent half shell	*(mechanical inside ,like mandrel)*	*(roller)*	like a tensioner	Slings
	actuator/mechanism for bending	hydraulic cylinder	*(screw press solution with motor)*	*(rack and pinion)*	*(wedge with horizontal cylinder)*		
	manoeuvre the machine from bend to bend	tracks (similar to tensioners or tank tracks) on/in pipe	(thrusters)	both tracks and thrusters	self-folding propellers and tracks	Handover clamps	
	bend die / contact element (above pipe)	like a tensioner	like a half shell convex or concave	*(roller)*			

This concept is a version of concept 5 which uses half shells as clamp, and bend dies, and rollers as locomotion system. The difference to the previous concept is in the deployment method which is here the one where the machine is guided over the stinger. The machine, when on the *Lorelay*, is lifted onto the pipeline like concept 12. In contrast to concept 12, the machine is not clamped to the pipe until it leaves the stinger. As it can be seen in Figure 6.33b the clamp, 1, is rotated above the machine body such that it does not interfere with the rollers on the vessel and stinger. Once the last roller box is passed, the clamp rotates around the pipe and the machine rolls on the rollers, 4, to the sagbend. The pipe is bent by the hydraulic cylinders, 2, which force the half shells, 5, which are connected to a pivot point in the middle of the machine, downward (Figure 6.33a). This machine completes the total required bend as little bends in a sequence.

As the machine is connected to the vessel via an umbilical on a winch the machine can be retrieved by pulling the machine with the winch such that is rolls on the pipe back onto the vessel.

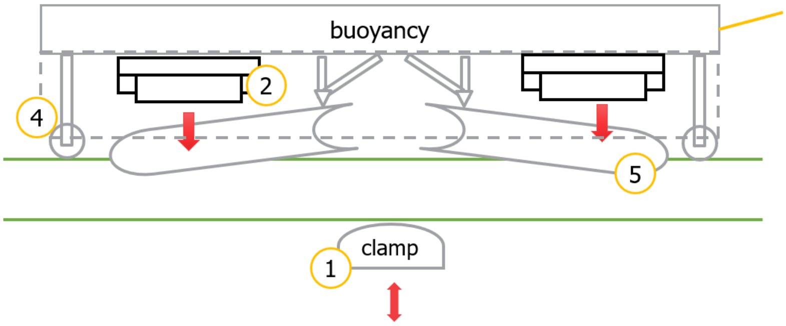


Figure 6.33a: Schematic - Concept 11 Side View

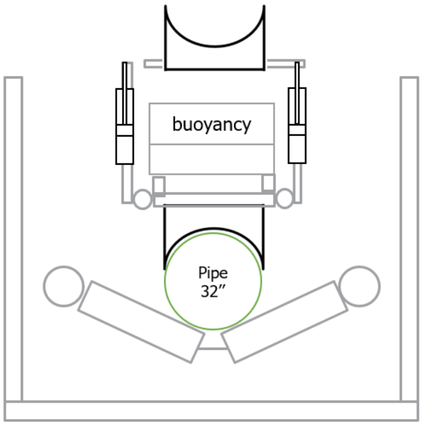


Figure 6.33b: Schematic - Concept 11 on Stinger

7

CONCLUSION CONCEPT EXPLORATION

In this chapter the results of the Concept Exploration phase are presented. Following the systems engineering approach the first step is to find functional requirements from the operational requirements, defined in the Needs Analysis. These functional requirements, listed in section 6.1, are valid for all different concept designs which are explored in this design stage. Defining the requirements answers part of the research question c).

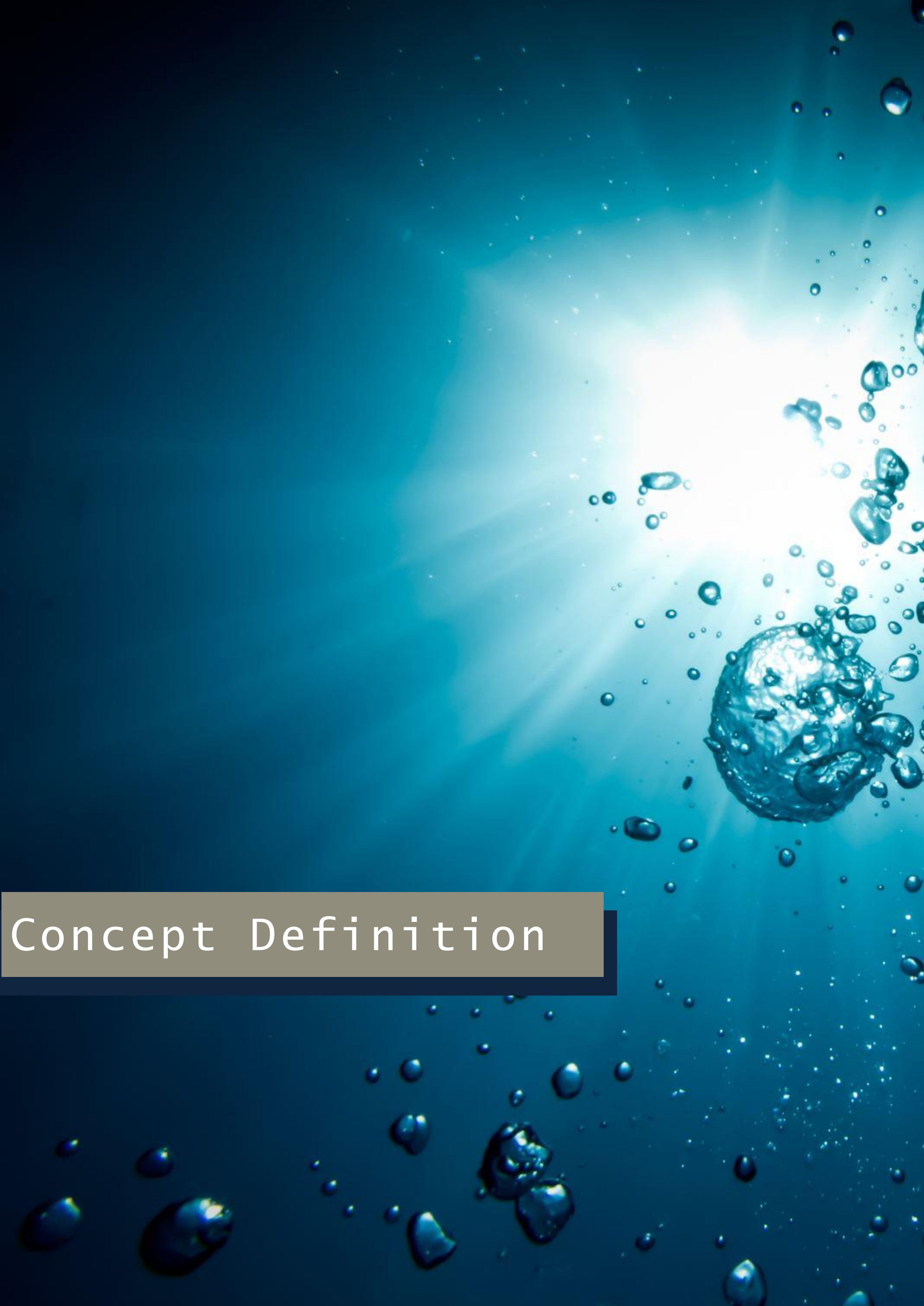
In the following five different subfunctions have been found, inspired by different onshore pipe bending machines and principles presented in section 3. These subfunctions are listed below.

- Machine body
- Mechanism for bending
- Clamp
- Bend die
- Locomotion system

With this result a next part of research question c) has been answered. For these subfunctions different solutions are investigated. These solutions come from different fields like for example the soft robotics arms or the linear winch like locomotion system. All solutions for these subfunctions are summarized in a morphology matrix in Table 6.1.

Alternative different concepts have been found by combining a solution of every subfunction. As not all combinations are judged feasible twelve concepts are found using this method which are presented in Table 6.2. To narrow down the number of concepts for the next phase the Concept Definition a trade-off analysis has been performed. The criteria after which the concepts are evaluated are presented in section 6.5.1. To achieve a more reasonable scoring the concept scores are multiplied with a weight given to each criterion, as not all criteria are of similar importance. This weight is found with the pairwise method (Table 6.4).

The top three concepts out of the twelve feasible ones found are the ones which are further investigated in the following design stage the Concept Definition. With these concepts the remaining part of research question c) is answered which asks for “alternative possible concepts for an underwater cold bending system for marine pipelines” and the Concept Exploration phase is concluded.



Concept Definition

8 DETAILED DESIGN

In this phase of the concept development the Concept Definition, the most preferable of the concepts, which have been filtered out in the previous phase, Concept Exploration, are designed and presented in more detail. These winning concepts of the trade-off analysis are two concepts with the rotation clamp and one concept with a handover system. For these two clamping mechanisms it is analysed what the preferred solution for deployment and therefore the design of the body is. By looking into this aspect in more detail the three concepts are narrowed down to two.

3D drawings are produced to visualize the system concepts in more detail. The contact elements namely the clamp and bend dies are sized according to the required contact area to prevent damage of the coating. Additionally, where feasible, commercial off the shelf parts are used. Examples are the hydraulic cylinders which have the correct capacity to bend the pipe, and beams which have standard cross-sections.

In this Concept Definition phase two different software are used. 3D drawings of the concepts are generated with the software SolidWorks with which also different FEM analyses are performed. SolidWorks is chosen as software because it has a user friendly interface and the file format is compatible with other CAD software as well. As the 3D modelling is the main focus to see the interaction of different components and the FEM analysis secondary this software is chosen over FEM software at this design stage. OrcaFlex is used to evaluate the influence of a bend section on a pipeline installed on a seabed with steep slopes.

During this phase it is determined how the normal pipeline installation procedure is influenced by adding the additional step of bending predefined sections. A storyboard for the remaining two detailed concept designs is developed. To answer research question e) an estimate of the required time to bend one 12m pipe joint for both concepts is presented below.

With the information from the detailed design and analysis of this phase the concepts are compared and the most promising one is found in section 8.6. Thus, research question d) is answered in the following as well.

8.1 Deploying the Machine

As presented in the previous section 6.4.1 there are several options to deploy the machine which has influence on the shape of the machines' body. These are

- Machine attached to the pipe on the pipelay vessel and deployed over the stinger
- A ROV which can fly around freely in the water, deployed from support or pipelay vessel
- Tool attached to an AUXROV which is deployed from a support vessel

For the remaining two clamping concepts, the rotation clamp and the handover system, this aspect is analyzed in more detail in the Concept Definition phase and the results presented in the following.

When deploying the machine over the stinger the question arises how to recover it when the bend is completed. Based on the coating requirement the outer clamps of the handover locomotion system are 60cm long (Figure 8.13). This length exceeds the available space between the rollers of the roller boxes on the stinger.

As the clamps do not fit between the rollers to clamp around the pipe on the stinger, climbing up with the handover clamp system is not a preferred solution. To resolve this problem the number of clamps can be increased to distribute the force sufficiently on the pipe coating. This however results in more parts and therefore a less simple system. Further, the 3m long machine is too short to reach over the total length of the roller box, which would avoid reaching in between the rollers entirely. It can be noted that a 3m long machine provides sufficient leaver arm to apply the force with hydraulic cylinders of conventional size. In section 8.3 the weight of the machine is calculated which is sufficiently low such that the use of the second overhead crane inside the vessel is not necessary (functional requirement 1.3) and therefore the length of 6.3m does not have to be met. In general it is desirable to design the machine as small, light and simple as feasible.

During normal pipeline installation, welding one additional joint to the pipeline, including testing and coating, takes approximately between 3min and 5min depending on numerous factors like the pipe size, type of coating and number of welding stations. If welding the joint takes less than 5 minutes then the handover system is too slow to climb back up the pipe again. The calculation of the travel time of this concept presented in detail in section 8.4. One option is to wait for the machine and lay the pipeline slower than possible until the machine is back on the vessel. Assuming a travel distance of 2000m from the bend section back onto the vessel, the approximated duration is 13 hours in which the pipeline installation is slowed down.

In order to enable the rotation clamp concept roll up the pipeline, after the bending procedure is completed, additional clamps or rollers are required. This is the case because the suspended pipe becomes steeper with increasing water depth which leads to the pipeline being close to vertical in the middle of the water column.


Figure 8.1: Cable Car Grip [86]

An additional idea to enable rolling the machine up the stinger is to install steel ropes on top of the stanchions of the stinger. These can be used to clamp the machine to it, with arms which have rollers at the end. The arms with the rollers look similar to the ones used for cable cars as it can be seen in Figure 8.1. The cable car arms clamp to the rope and the rope is being moved. However, on the stinger the cable is fixed and the clamps need to be powered rollers which provide the power to move the machine on the pipe. The machine sits then on additional rollers on the pipe to minimize friction. The arms on either side clamped to the rope additionally provide stability to the machine against movement in roll direction. This idea is however no longer pursuit because the distance between the roller box and the top of the stanchions varies along the stinger from roller box to roller box by about 5.5m. This means that the arms need to be flexible in their length like for example telescope arms. This idea results in a more complex system of the machine and modifications to the stinger. Consequently, it is more convenient to detach the machine from the pipe right after the bending is completed.

The second option to deploy the machine is to design a ROV which can be deployed from either the pipeline installation vessel or a support vessel. Then, the ROV attached to teaser management system (TMS) is deployed from the vessel with a crane, LARS or winch until it is close to the seabed. The ROV then detaches from the TMS and can fly around freely attached with a teaser line to the TMS. This requires that the machine is light enough in water such that the thruster capacity is sufficient to not only keep it stationary but also move it in a current. The thruster capacity of a large WROV, for example the Schillig Robotics UHD-II ROV is 1000kgf in the vertical direction and 1200kgf in the horizontal direction [58]. The limiting submerged weight is then assumed as about 100kg as it can be seen in equation 8.1.

$$\frac{1000kgf}{9.81\,m/s^2} = 101.93kg \qquad 8.1$$

Although the payload of the UHD-II ROV is 300kg the smaller submerged weight is assumed to allow for current velocity [58]. If installing buoyancy modules on top of the machine with the surface area of 3m x 1.3m, the height of the buoyancy modules, with the buoyancy per element of 11.85kg [87], is approximately 4m.

The last option investigated in this thesis for the remaining clamping solutions is the AUXROV attached to the bending tool as presented in section 6.4.1. The assembly is then deployed from a support vessel right above the bend section. It is not required for the machine to fly from the pipelay vessel where it is deployed to the bend section which can be several hundred meters away, as it is the idea of the ROV concept.

The advantages are:

- The AUXROV is a proven system which performed reliable in various projects so far. (Centurion launched the AUXROV in 2016). If designing the ROV such that all needed parts are assembled on the machine then designing, testing and verifying of all these elements as a new system is required additionally to the design and testing of the bending mechanism itself.
- If an element of the ROV system fails then the AUXROV can be exchanged and bending procedure can be continued. The ROV concept, on the other hand, needs to be repaired and during this time no bending can be performed.
- The AUXROV can be used for other tools as well (Figure 6.5).
- As the machine is deployed directly above the bend section from a support vessel, the submerged weight is only limited by the allowable weight on the suspended pipeline, which is about 5tons. This means that less buoyancy is required compared to the ROV concept where the submerged weight is limited to 100kg. The tools weight in air can be up to 30ton which is the capacity of the lifting frame of the AUXROV [59].

The AUXROV is connected to the tool via an interface plate which is locked with a hydraulic locking pin. A magnetic sensor and software feedback show when the lock is fully engaged. Safety mechanism are built into the software to ensure that the tool cannot be disengaged without using a password. What is not verified at this point is if there is a fail-safe mechanism if the hydraulics fail which enables the operator to disengage the tool. In case of failure of the hydraulic system it would be a great advantage to be able to at least recover the AUXROV instead of losing both the AUXROV and the tool [88]. As it is possible, according to the brochure, to customize the AUXROV to a certain extent like adding additional cameras or sensors, it is assumed that it is possible to integrate the function of releasing the tool if the hydraulics fail [59].

Because of this presented reasoning the AUXROV concept is selected as the most preferable option for both clamping concepts, the rotation clamp and handover system. The previously three concepts are therefore reduced to two concepts with the same deployment principle but with different clamping mechanisms.

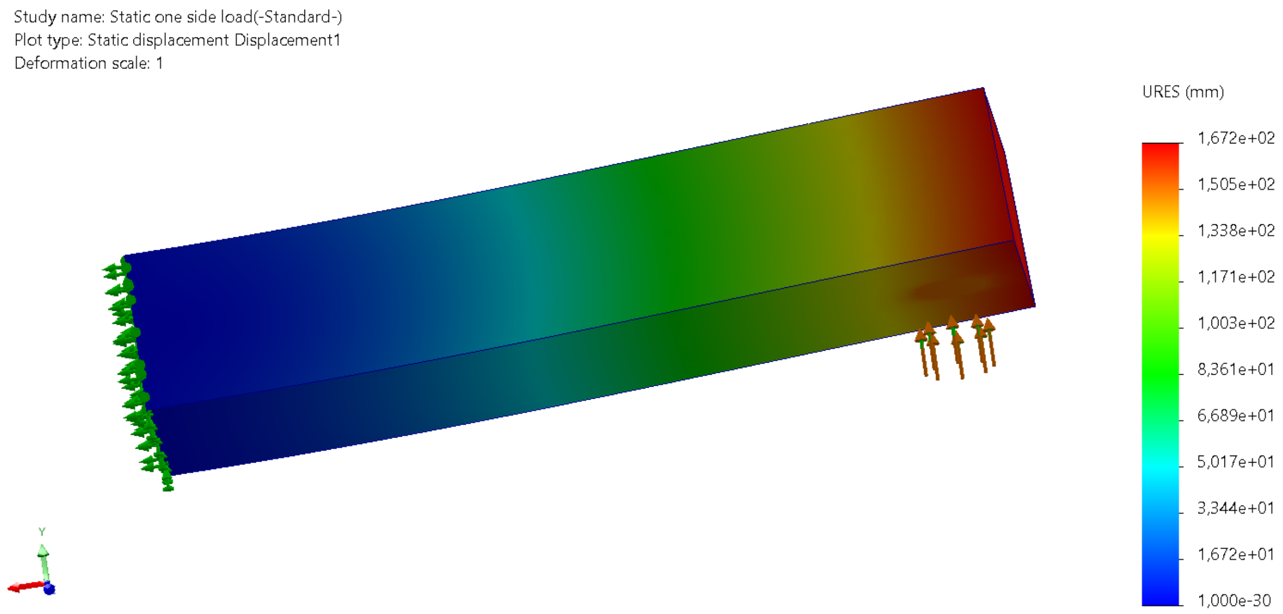
8.2 One vs. Two Cylinders

The bending principle of the two remaining concepts is a version of press bending, a cold pipe bending procedure as introduced in section 3.1. In conventional press bending the middle part is the bending tool which is moved and the outer parts are fixed rollers (Figure 3.2a). In the design of the tool used for underwater pipe bending the moving and fixed parts are switched: Two outer bend dies are forced down while a clamp in the middle of these remains fixed.

Vertical bending machines which are used for field bends during onshore pipeline installation however have only one bend die and the pipe is clamped at the end of it as described in section 3.2.1. This distinction is the reason why an additional simple analysis is performed with SolidWorks. The goal is to see the difference of the deformation of the machine body when using one instead of two cylinders. In theory a smaller leaver arm results in less deformation.

As it can be seen in Figure 8.2 a solid body with the approximate dimensions of the machine body is modelled. In the first scenario this body is clamped on one side and a pressure is applied on the projected surface of the cylinder body to represent the reaction force of the hydraulic cylinder on the body. In the second scenario the body is clamped in the middle and two pressure areas are modelled on the ends of the body as reaction forces during bending.

The results presented in Figure 8.2 show that the use of two cylinders at opposing ends of the machine results in less deformation than applying the combined load of both cylinders on one side. This output verifies the assumption of the smaller lever arm leading to less displacement. Compared to the onshore vertical bending machine this is an improvement because it means that the body is more stiff. The added stiffness is favorable as there is no soil to provide reaction force like it is the case for onshore machines. Additionally, this result leads to the assumption that a concept design where the force is distributed to two cylinders has the potential to bend pipelines with larger diameters.



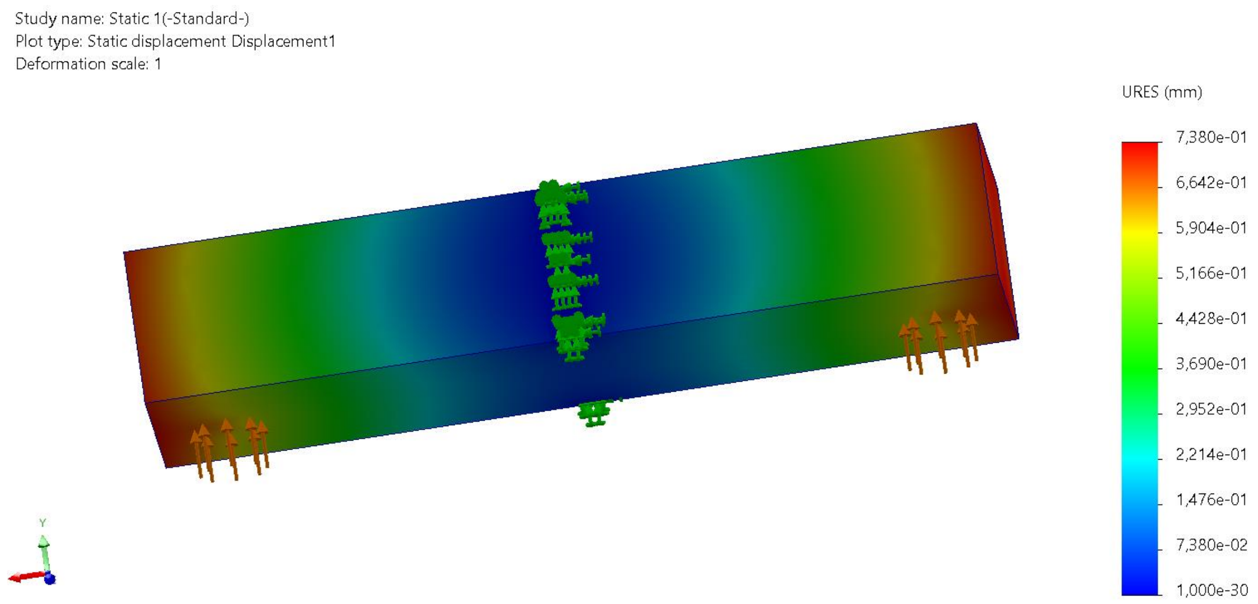


Figure 8.2: FEM Results - One vs Two Cylinders

8.3 Design Characteristics

In this section the remaining two concepts are presented in more detail. This includes drawings which have been generated while sizing different elements and FEM analysis to verify the structural integrity of parts and assemblies of the concepts.

8.3.1 The Rotation Clamp Concept

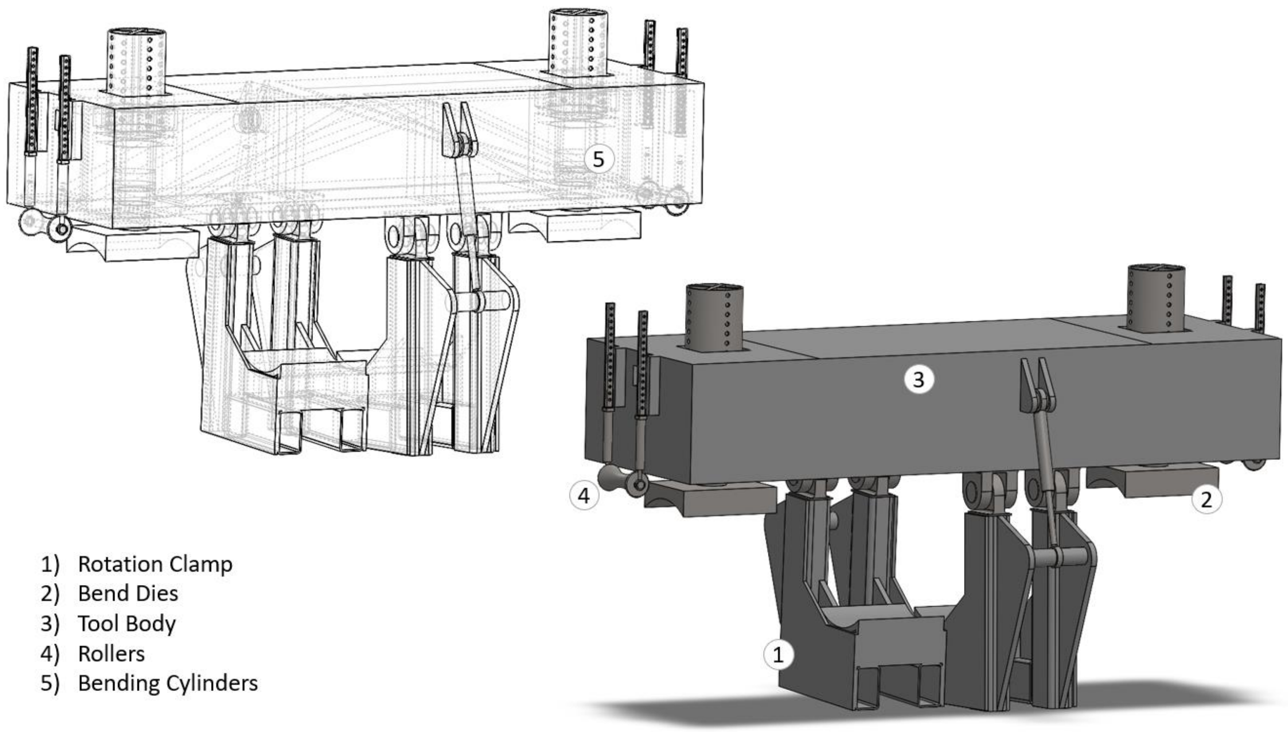


Figure 8.3: 3D Drawing - Rotation Clamp Concept

Using this tool the pipe is being bent when the rotation clamps in the middle of the tool clamp around the pipe and the outer bend dies are forced down by hydraulic cylinders, adapting the cold bending procedure press bending as presented in section 3.1. How this system concept impacts the normal pipeline installation is described in section 8.4.

The rotation clamps are made out of two rectangular steel tubes, with the dimension 250mm x 150mm x 20mm, which are welded together in a $90°$ angle. They are element number 1 in Figure 8.3 and are the fixed supports which provide the reactive force to the bending force provided by the hydraulic cylinders on either side of the clamps. To increase the brackets stiffness, such that they are not bent open during bending, steel sheets are welded to one side, and another smaller one is welded into the corner. Two of these brackets connected by the contact element to the pipe form the main structure of one clamp. The contact element is a half shell which dimensions are estimated such that the area is large enough to distribute the reaction force of the two 300ton bending cylinders to not damage the coating. As presented in section 6.3 the allowed force per unit area is 10MPa. With a total reaction force of 5335kN the estimated area is 0.51m x 1.1m. Distributing this area to now two clamps the length of each clamp is about 0.6m. The decision to use two clamps reaching around the pipe from opposing sides has been made to improve the tools stability during bending. Using only one clamp results in the clamp tending to bend open instead of the pipe being bend. This change has the additional effect that the forces, which are bending and tensile forces, are now distributed over 4 vertical beams of the clamps, instead of 2.

The rotation clamp design is based on the opening mechanism of the tensioner in Figure 6.7a. Based on the weight of the clamps a 10ton double acting cylinder with long stroke is selected to open and close the clamps which have a weight of about 860kg each [89]. The static linear FEM analysis shows that the part itself is stiff enough to resist the reaction forces of the 300ton cylinder while bending the pipeline. This analysis is chosen as no large displacements are expected. In SolidWorks a large displacement solution is used if the ratio of the maximum displacement over the characteristic length of the model is larger than 10% and the stiffness of the material changes during loading [90]. As it is chosen to use two clamps, the force which is modelled onto the contact area is 2667kN, as shown as pink arrows in Figure 8.4. The upper end of the vertical rectangular steel tubes is considered fixed in position as the machine clamps around the pipeline during bending and cannot move in any direction relative to the pipe. As the material of the pipeline is assumed to be X65 the material of the machine is chosen as S460N with a yield strength of 460MPa. The stresses in Figure 8.4a are mostly in an acceptable range except the green marked corners which can be optimized in a later design stage. The resultant deformation visualized at the bottom of Figure 8.4b is close to zero.

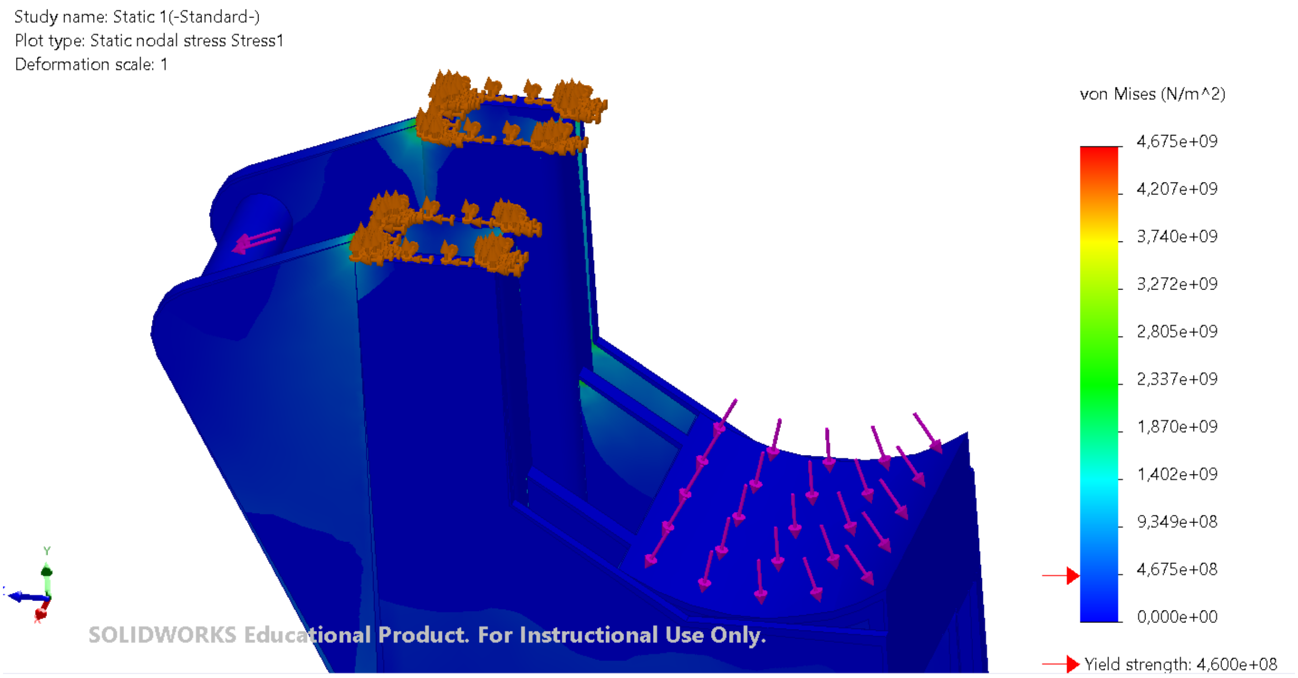


Figure 8.4a: Result FEM Analysis – Stress Plot - Rotation Clamp

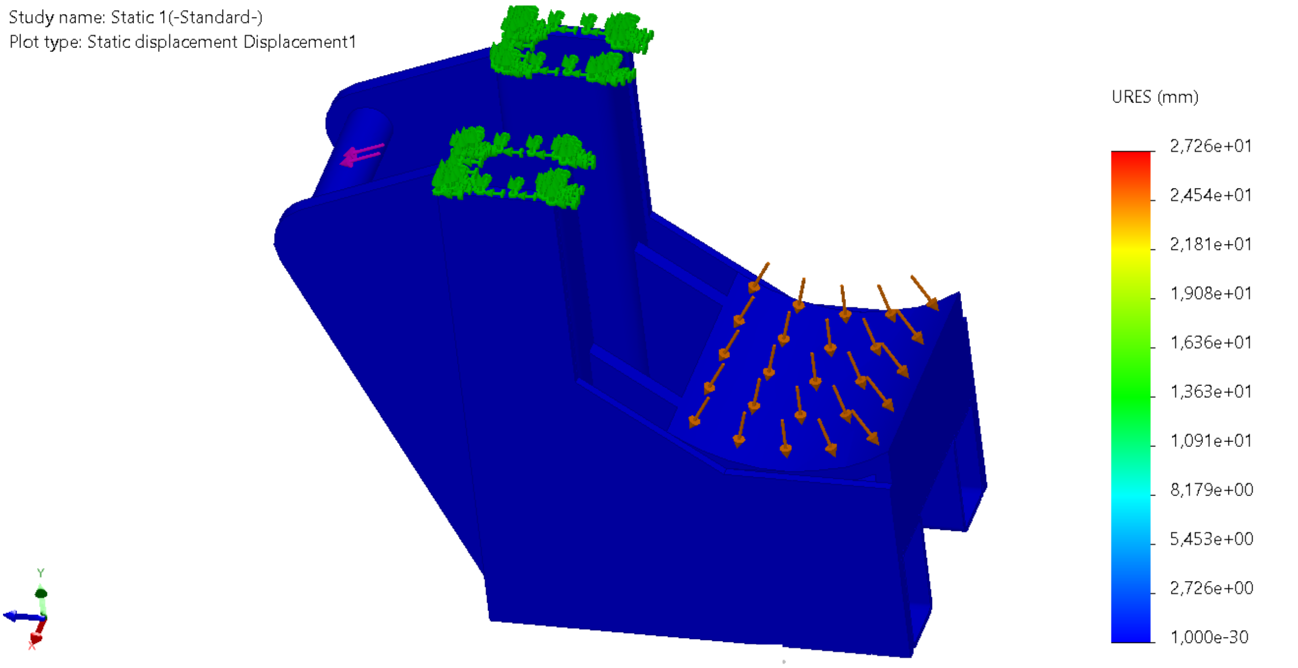


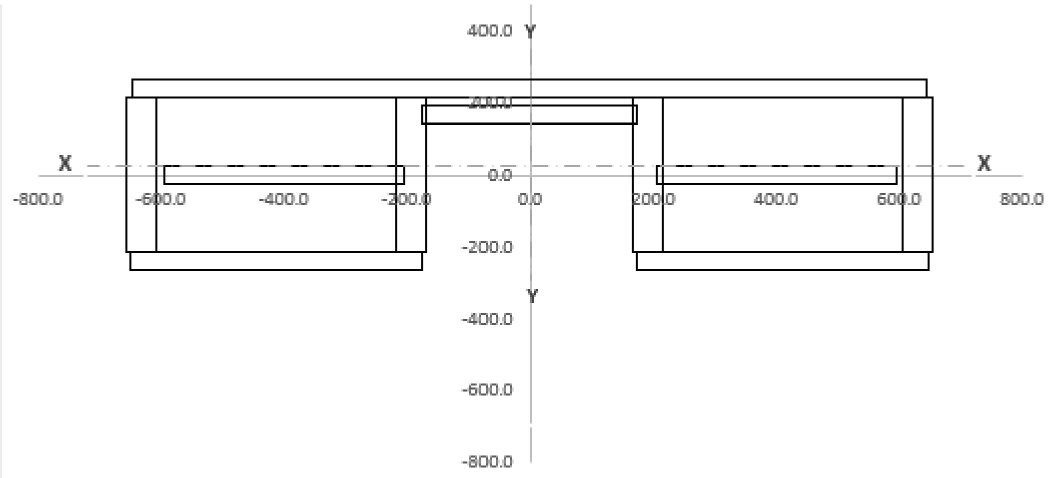
Figure 8.4b: Result FEM Analysis – Resultant Displacement - Rotation Clamp

The bend dies are the half shell shaped elements on either side, in a small horizontal distance, to the middle clamps. These can be seen in Figure 8.3, marked with number 2. They are the interface between the hydraulic cylinder and the pipe. As it is the case for the contact area of the clamp, the contact area to the pipe of the bend die is calculated according to the allowable 10MPa force on the PE coating. When bending the pipeline the bend dies should be able to tilt a little such that when the pipe bends the bend die can follow the movement and maintain contact over the total contact area. A solution can be tilt saddles, which are a part mounted on the top of the cylinder. These saddles can tilt up to 5° and are usually used to extend the lifetime of the hydraulic cylinders as they compensate the side load [91]. In general, hydraulic cylinders are sensitive to sideloads and can be damaged by them.

This concept can be adapted to different pipe sizes by mounting pads onto the clamp and bend die areas which are in contact with the pipe. This way the radius of the clamps and bend dies have the curvature equal to the new pipe diameter. Additionally, the rollers, 4, which are enabling the tool to move along the pipe, and the 300ton cylinders, 5, can be moved up or down to an initially optimized position such that the cylinder stroke is still sufficient. The mechanism is basically a pipe inside a pipe whereas the outer pipe is rigidly connected to the tool body. Bolts guided through holes in both pipes fix the inner pipe in position.

When designing any underwater tool the goal is to make the machine as light, small and simple as reasonably possible. At the same time the deformations of this tool should be minimized. One option to increase the stiffness of the body is to make the machine short. This reduces the leaver arm between the clamps in the middle and the reaction force of the hydraulic cylinder at the end and thus the deformation. It can be noted that the length of the machine still needs to be sufficient such that the bend dies are not directly over the clamp. This would cause the pipe being compressed and make it oval instead of bending it.

To get a first idea about the dimensions of the required cross-section of the body, 3, its bending resistance is calculated with a simple in-house spreadsheet developed and used by Allseas engineers. Different configurations and sheet thicknesses have been tested to find a suitable design which is used as basis for the 3D model. The cross-section which is used eventually for the first draft of the machine is presented in Figure 8.5. In this case 50mm thick steel plates are assumed which are welded to a box of the width of 1300mm and height of 430mm. The resulting bending stiffness of $2.7\text{x}10^7$ mm^3 is larger than $1.7\text{x}10^7$ mm^3 the bending stiffness of the largest pipeline to be bent which is calculated in equation 6.3.



	X-X	Y-Y	
I_{gl} [mm^4]	8,118,254,174	43,624,246,935	Moment of inertia
e_1 [mm]	292.9	656.5	Extreme fiber distance
e_2 [mm]	237.1	656.5	Secondary extreme fiber distance
$W_{be\ 1}$ [mm^3]	27,716,000	66,449,729	Bending resistance at extreme fiber
$W_{be\ 2}$ [mm^3]	34,241,029	66,449,729	Bending resistance at secondary extreme fib

Figure 8.5: Initial Cross-Section of the Tool Body and Cross-Sectional Properties

To increase the stiffness of the body, steel sheets are added in the hollow inside of the body, which distribute the loads like a roof. These sheets are designed such that they reach from the top middle to the lower outer part of the body. This feature can be seen in the top left drawing in Figure 8.3 which shows the hidden edges of the design as well. The height of the body is adjusted as well. It is increased to integrate the system with which the 300ton cylinders can be set to the required position. To verify its stiffness a similar FEM analysis as for the clamp, has been performed with SolidWorks. Again, as the deformations are assumed to be small a linear static analysis is used to estimate the von Mises stresses and the expected deformation during bending. The hinges which are the connection between the clamps and the body are modelled as fixed constraints. The reaction force of the 300ton bending cylinders, 2667kN, is applied on the projected area of the body of the hydraulic cylinder. In Figure 8.6 it can be seen that both, stress and deformation, are reasonably low and this design of the body can be considered as stiff enough. As simplification this analysis has been performed on the tool body which does not include the mechanism with which it is possible to adjust the cylinder height to the pipe diameter.

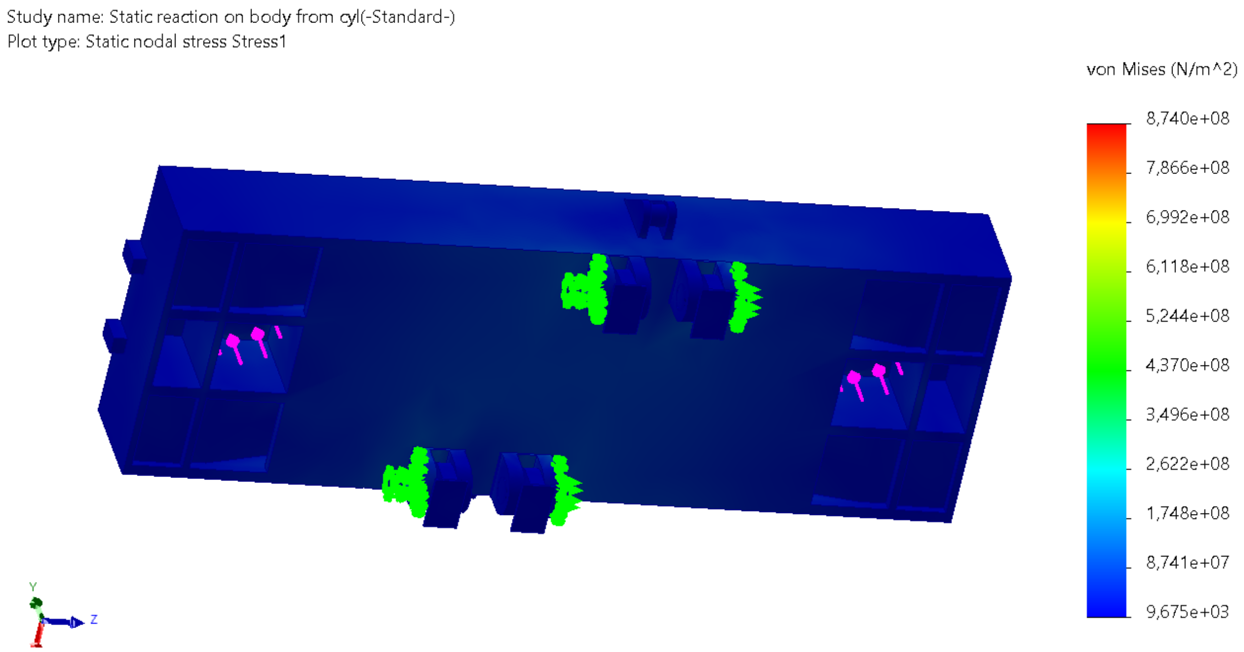


Figure 8.6a: Result FEM Analysis - Stress Plot - Body of Rotation Clamp Concept

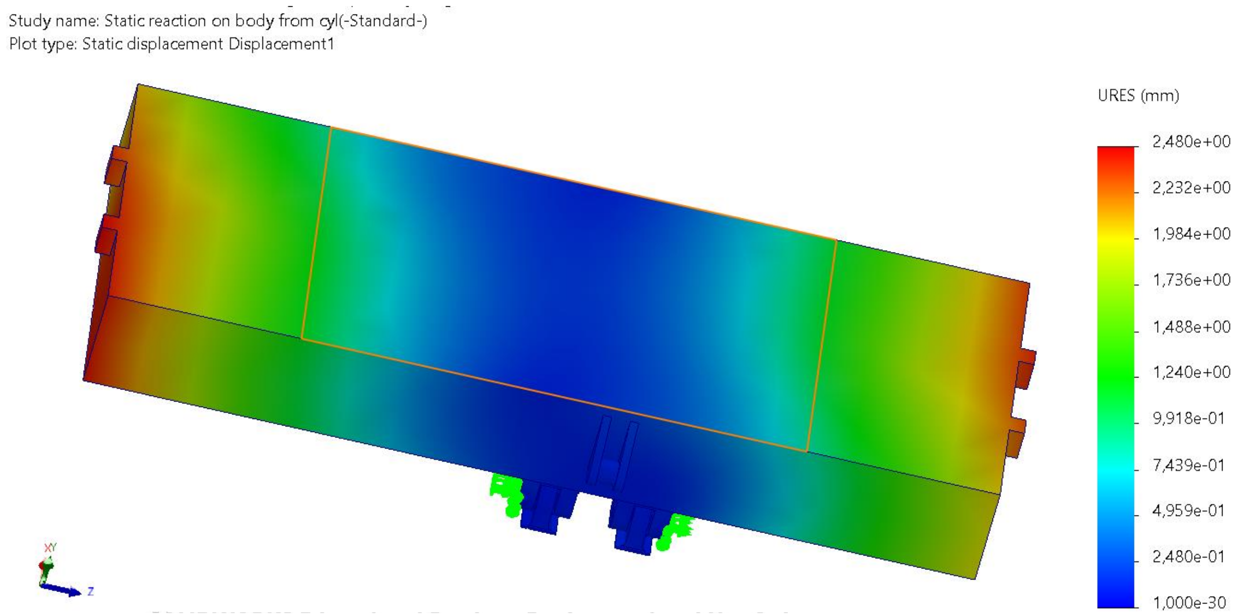


Figure 8.6b: Result FEM Analysis – Resultant Displacement - Body of Rotation Clamp Concept

This tool has now the overall dimensions of about 3800mm length, 1300m width, the body is about 700mm high and the height of the clamps is 1100mm. The total weight is about 12.7ton. The inner distance between the bend die and the clamp is about 500mm. These dimensions enable the machine to be transported in a standard 20ft container and its weight is within the lifting capacity of the AUXROV of 30ton [59].

To achieve the submerged weight of 5ton the required volume of buoyancy modules is 12.52m^3 for this concept [87]. As this is the first draft of the concept design it needs to be taken into consideration that the weight can be subject to system optimization aiming to reduce the weight but sustain the stiffness. Here the hollow body can be filled with buoyant material. Additionally, similar to the caterpillar concept buoyancy modules can be distributed over the assembly under the consideration that the assembly is still stable under water. This depends on the centre of gravity (CG) and centre of buoyancy (CB) as it is also the case for ROVs [92]. Figure 8.7 shows the righting

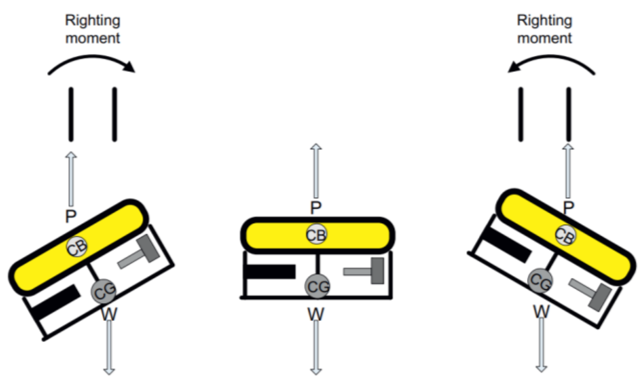


Figure 8.7: Floating stability – ROVs [92]

moment which is the result of the horizontal leaver arm between the CG and CB. The ROV returns to its initial position such that the CB and CG are in vertical alignment and is considered stable in the water. It might be required to design a frame structure on top or surrounding the tool in order to install the buoyancy modules. The manufacturer of the buoyancy modules advises to contact them for the detailed design and material selection [87].

8.3.2 The Caterpillar Concept

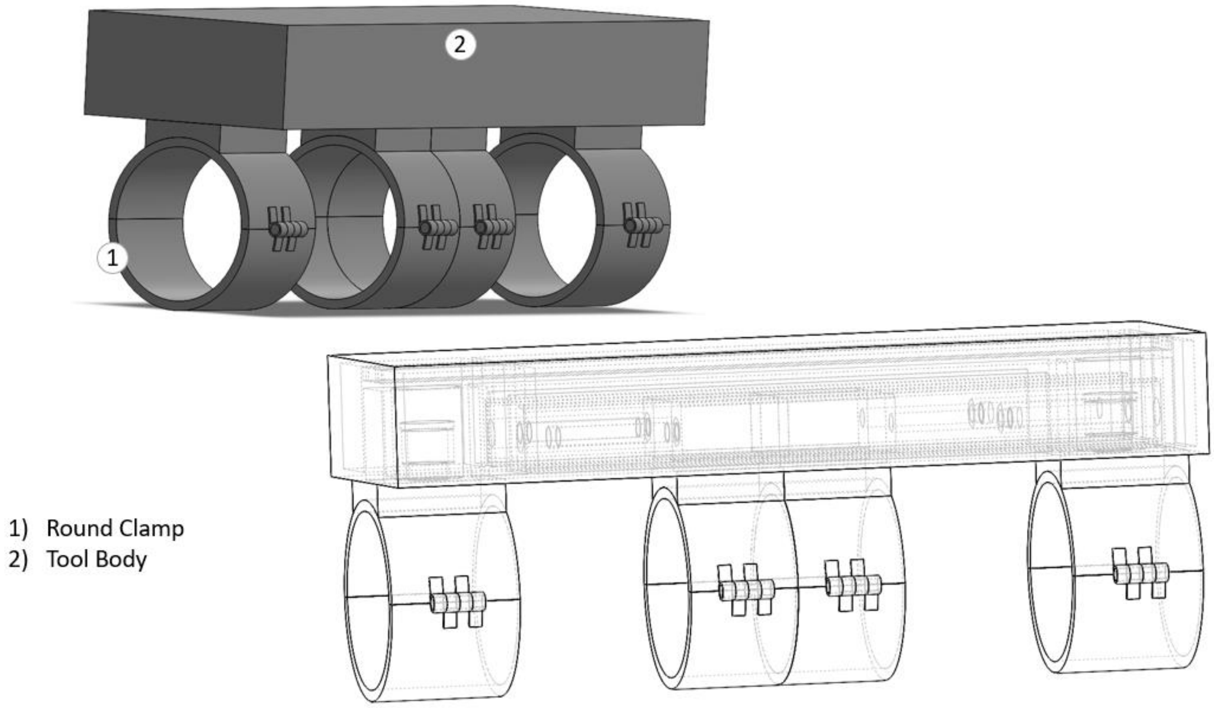


Figure 8.8: 3D Drawing - Caterpillar Concept

The machines' design is based on the required bending moment (calculation described in section 6.2), assuming a total leaver arm of 3m, 1.5m from the middle to each side. The pipe is bent when the clamps are closed around it and the two outer clamps are forced down by hydraulic cylinders. The middle clamps provide the reactive force and the pipe is bent downwards. Then 300ton cylinders with 15cm stroke are large enough to provide the required force and the machine is a little over 3m long. The dimensions of the hydraulic cylinders in the drawing are based on available onshore cylinders [64].

The 300ton hydraulic cylinders, 5, are built into a machine body, 2, which needs to be stiff enough to not deform because of the reaction forces of the cylinders. The goal is to bend the pipe, not the bending machine. At the 300ton cylinders attached are round clamps made out of two half shells from round pipe sections which inner diameter equals the outer diameter of the pipe which is supposed to be bent (Figure 8.10). To adapt to different pipe sizes for different projects there are two different options. A first idea is to exchange the clamps such that the inner diameter of the new clamps equals the outer diameter of the new pipe. An alternative can be to attach pads into the half shells when smaller pipes are planned to be bent. The two parts of the clamp are connected with a hinge which is controlled by an rotary actuator. This rotary actuator can be similar to those used for the Titan manipulator [93] which is a manipulator used for WROVs. The manipulator can handle loads up to 454kg which is larger than the weight of one half shell of the clamp which is about 317kg.

Due to the high forces it is assumed that the rotary actuators cannot hold the clamps closed during bending. Consequently, on the opposite side of the hinge of these clamps, there needs to be a locking mechanism, which is displayed in detail in Figure 8.9, such that when the forces are applied the clamp does not open. The idea displayed here is based on the working principle of a suitcase buckle. The length of the clamps is similar to the rotation clamp concept. There the length is a result of the minimum contact area which is a requirement to prevent the damage of the coating.

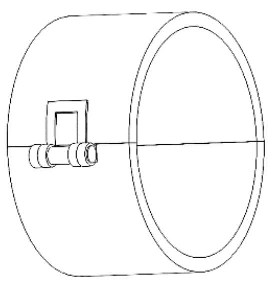


Figure 8.9: Round Clamp with Locking Mechanism

The same clamps are used in the middle of the machine to provide the reaction force to the bending force of the outer 300ton cylinders. The initial design was to only use one single clamp in the middle. When the pipe is bent it is assumed that it will deform on both sides of the middle clamp because the deformation of the pipe section clamped by the middle clamp is restricted. When the machine moves forward, the clamp should not clamp around any bend in the pipe. This means that the 1m long clamp needs to be moved by 1m. To avoid using cylinders with over 1m stroke to move it, the single middle clamp is therefore divided into two at this design stage, as it can be seen in Figure 8.8.

The two middle clamps are moved by two, horizontal installed, hydraulic cylinders each, which are inside the tools body as it can be seen in Figure 8.10. Using two cylinders simultaneously to push or pull something prevents rotating of the part and as a result that it might get stuck. One end of each of these cylinders is attached to the left and right side of a block, 3, and the other end is attached to the machine body. The middle clamps are each welded to this block which is attached to the horizontal cylinders, 4. This block is formed such that it reaches around the bottom sheet of the body and into UPE300 profiles which serve as guide rail for the blocks when they move. These U-profiles extend along the middle part of the machine to provide additional stiffness to the body.

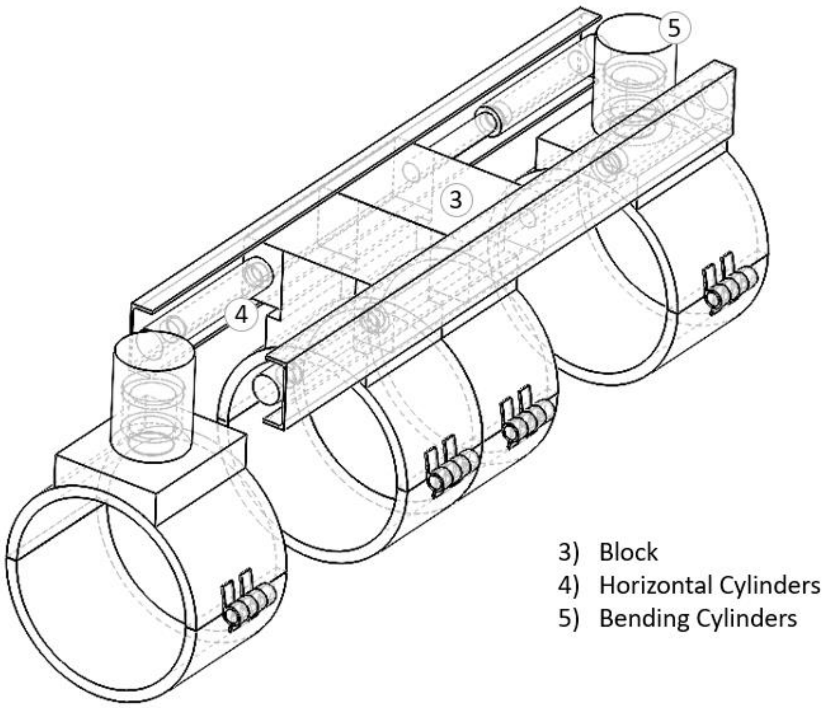


Figure 8.10: 3D Drawing of Internal Parts - Caterpillar Concept

In the middle, the initial position, the cylinders of the one middle clamp are extended and the cylinders of the other middle clamp are retracted (Figure 8.10). When moving the machine along the pipeline the middle clamp with the extended cylinders moves first by retracting the cylinders, then the other clamp is moved by extending its cylinders. When pulling the machine forward all horizontal cylinders can be activated at the same time while the middle clamps are closed around the pipe. The only reason to move the two middle clamps separately is that one of the deformations is located in the middle of the two clamps during the first pull to the next bending section. Then the distance which the clamps have to move is divided in half compared to the version with only 1 middle clamp. This means that also smaller horizontal cylinders can be used which fit better into the body. In this case 50ton cylinders with a stroke of 511mm are selected [89].

At this level of detail this concept has a total weight of 11ton. The body has the dimensions of 3800mm x 1300mm x 550mm (L x W x H) and is based on the initial cross-section as presented in Figure 8.5 like the rotation clamp concept. The steel sheets of which the body is made of, are surrounding the mechanism to move the middle clamps in horizontal direction, provide protection. The body structure is rigid enough to compensate the reaction forces of the 300ton hydraulic cylinders, which are used for bending the pipeline. This is verified by a FEM analysis with SolidWorks as it can be seen Figure 8.11. The reaction force is modelled as a pressure on the projected area in which the body of the cylinder is connected with the tool body. This force is shown as pink arrows in Figure 8.11. The middle clamp is considered fixed, indicated by green arrows, as it clamps around the pipe during bending. The results of the von Mises stress are shown in Figure 8.11a and the resultant displacement in Figure 8.11b. The AUXROV does not add much stiffness to the body when attached. It is equipped with a 30ton lift frame which means it is stiff enough to resist forces of 294.3kN. Compared to the 2667kN which is the reaction force of each of the 2 hydraulic cylinders the stiffness of the AUXROV can be neglected in the calculations. The same applies for the submerged weight of the machine which can be up to 5ton in order to ensure the integrity of the pipeline as it is shown in the OrcaFlex calculation in section 6.4.1.

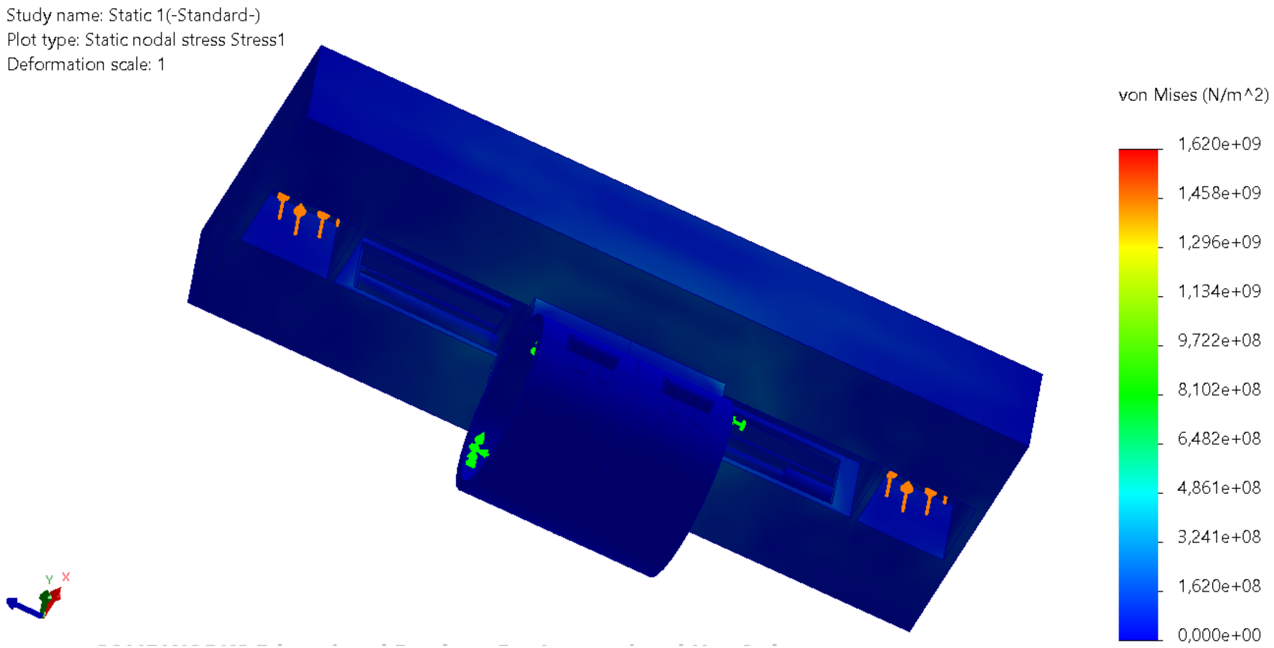


Figure 8.11a: FEM Result – Stress Plot - Body of Caterpillar Concept

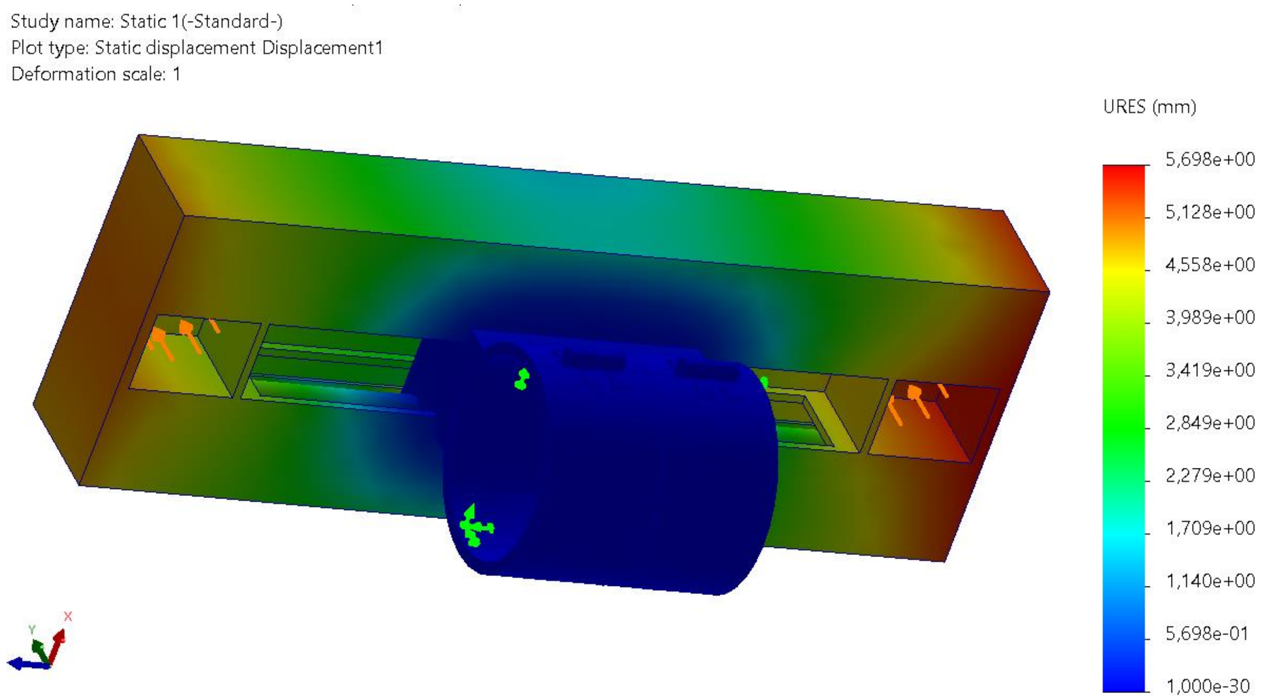


Figure 8.11b: FEM Result – Resultant Displacement - Body of Caterpillar Concept

Buoyancy modules are used to reduce the submerged weight to 5ton such that the suspended pipeline does not buckle. With a weight of 11ton of this concept, this means that 6ton of buoyancy should be attached to the assembly. When assuming the same modules as introduced in section 8.1 with the buoyancy per element of 11.85kg [87], then the stacked up modules on top of the tool would reach a height of about 2m. The 9.75m^3 buoyancy can be attached to the sides of the tool as well as the top, to the AUXROV, or as cylinder around the umbilical as well to optimize the distribution. It can be noted that the usual procedure as it is done for the cost calculation is to contact the supplier for buoyancy modules and they help with selecting the correct material and number of elements or design of one continuous block. That means that the presented calculation of buoyancy might change when the tool is designed in more detail and the input of the buoyancy supplier is included in the design.

8.4 Impact on Installation Procedure

In this section it is described how much time it takes to bend a 12m long pipe section and how these durations are calculated.

The calculation on the duration of the procedure for the rotation clamp concept is based on the extension and retraction time of the hydraulic cylinders which are used to bend the pipe, 300ton cylinder, and rotate the clamp, 10ton cylinder. As the cylinders are operating at a pressure of 700bar a IHUP unit is needed to additionally compress the hydraulic fluid because the output pressure of the AUXROV is 250bar. This capacity can be calculated from the information in the brochure [59] with the following equation 8.2 [94]:

$$hp = \frac{P[pound/inch^2] \cdot Q\ [gallons/min]}{1714} \qquad 8.2$$

The output flow rate of the IHPU unit with which the hydraulic cylinders are activated is then 30 l/min [95]. The time the cylinder extends or retract is then calculated as [96]

$$t[s] = \frac{A[cm^2] \cdot 60 \cdot L[cm]}{Q[l/min] \cdot 1000} \qquad 8.3$$

Additionally, the pipe joint is bent such that little bends in a sequence are introduced to achieve the desired total angle. The distance between the bends is 0.95m such that no bend is in the direct contact area of either the bend die or the clamp. If this would be the case, the pipe does not have contact along the whole contact area which can result in the damage of the coating and straightening of the bend. As it can be seen in Figure 8.12 the distance between the bend die and the clamp is about 50cm. Assuming a third bend is being produced, the distance of the first bend to the outer side of the bend die is then about 18cm, and the second bend is located 17cm away from the bend die towards the rotation clamp. The second bend is then located between the outer bend die and the rotation clamp.

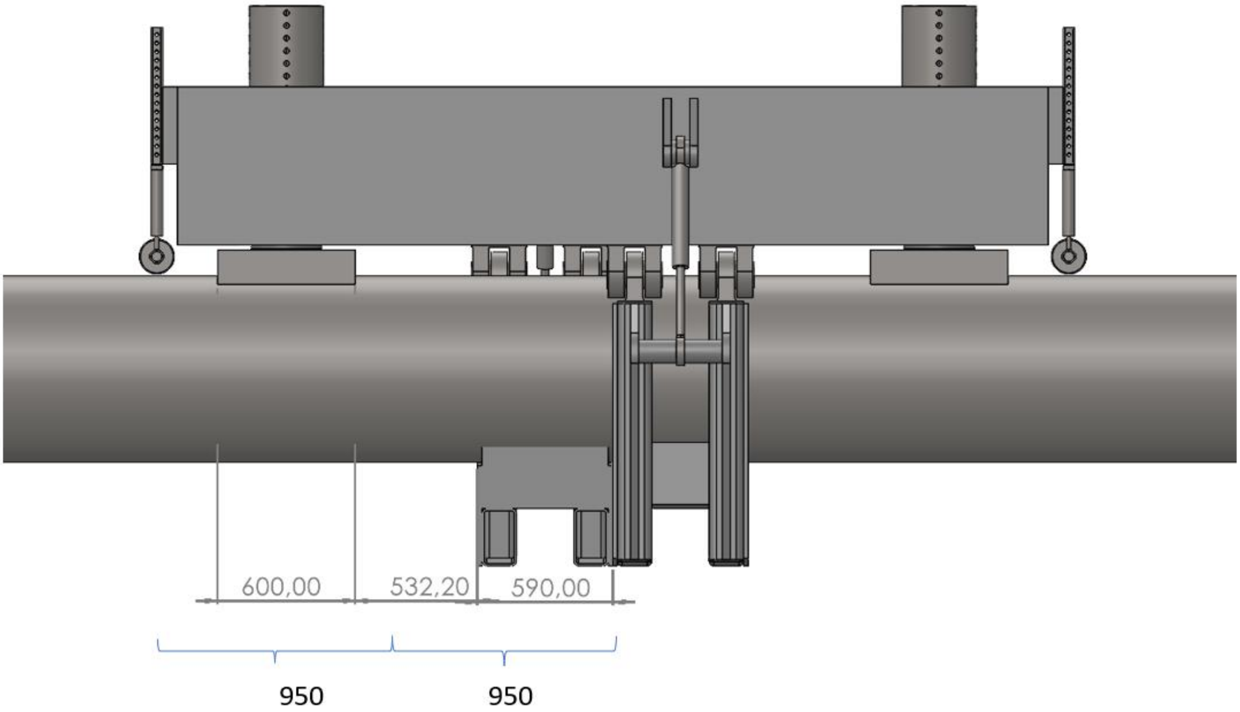


Figure 8.12: Distance between Bends in [mm] - Rotation Clamp Concept

With this distance between the bends and the regulated distance to the welds the number of bends which can be introduced in one 12m joint is 11. This results in a total achievable angle per 12m, 32" pipe section of $18.5°$. Assuming the tool rolls over the pipe which is powered by the thruster of the AUXROV the speed which the tool can roll along the pipe is 0.25knots. It can be noted that the maximum thruster speed is about 4 knots but the heavier the machine the slower it can roll along the pipe. The total time required for the tool to clamp the pipe and bend the maximum number of bends in a sequence in one pipe joint is approximately about 5 minutes.

For the caterpillar concept the required duration of bending one pipe joint of 12m length is dependent not only on the extension and retraction time of the hydraulic cylinders which can be calculated with equation 8.3, but also on the time which is needed to open the clamps with the rotary actuator. Equivalent to the previous concept 300ton cylinders are used to bend the pipe with the outer clamps. Instead of using the thrust of the AUXROV this concept uses the hydraulic clamping system which is similar to the linear winch. 50ton cylinders are used to move the middle clamps in the horizontal direction. To rotate the clamp the rotary actuators which are used for the Titan manipulator, a ROV arm, are used as reference [93]. The wrist rotating speed of the Titan manipulator is 6-35rpm. Choosing a quarter circle as rotation and the slowest speed of 6rpm means that opening the clamp takes 5sec. When reaching over the bend with the first middle clamp and the following the movement which is described in section 8.3.2 the clamps need to open 5 times, the 50ton cylinders extent 6 times and retract 3 times. With the distance between the bends of 1.5m the resultant number of bends per 12m joint is 7. When adding up the clamp opening and cylinder extension and retraction times, this machine needs about 10min to introduce these bends including the time needed to move between the bend sections.

The distance between the bends of 1.5m is selected because of the following assumptions: Using this tool the pipe is likely to bend on either side of the middle clamps which results in two bends which should not be in the area of a clamp when producing the next bend. As it can be seen in Figure 8.13 the bend on the right side of the middle clamps is located between the outer clamps and the middle clamp (marked in blue) and the bend on the left side of the middle clamps is located on the left side of the outer clamp (marked in green) when the tool is moved 1500mm forward.

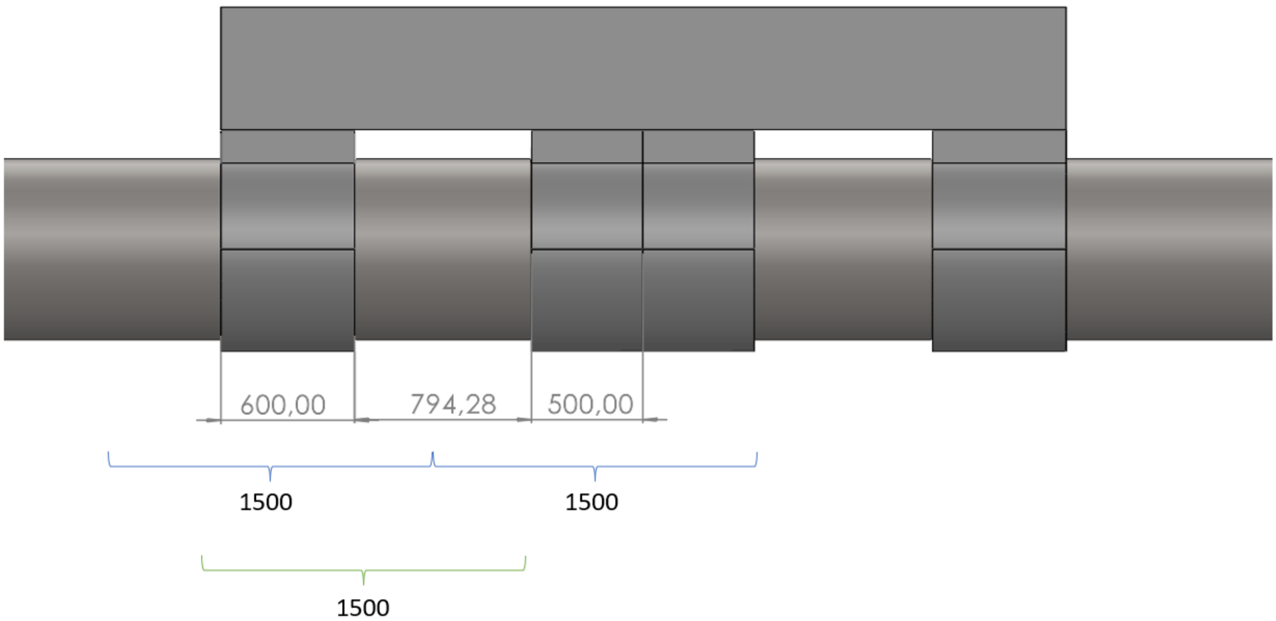


Figure 8.13: Distance between Bends in [mm] - Caterpillar Concept

Assuming 7 bends per pipe joint the achievable bending angle per 12m pipe section of the size 32” is 11.5°. With this locomotion system it is possible to move the machine within about 5min over the distance of one 12m long pipe joint. This is faster than moving between the bends as the middle clamps can be moved simultaneously where no bend sections are in the pipe. For a distance of 2000m it takes 13 hours to recover the machine when it crawls back along the pipe. The cylinders retraction and extension times can be found in Appendix G.

8.4.1 Functionality of the Concept

To test the functionality of the tool a previous project in which the pipeline crossed a steep slope is chosen to compare the free span length, and height, and bending moment of the pipeline. The pipeline of this project has an outer diameter of 22". The bending radius of 40D of this pipeline is therefore about 22.35m. The distance between the little bends for the rotation clamp concept is 0.95m. This concept is chosen for this analysis as the achievable bending angle is larger per 12m pipe joint compared to the handover clamp concept. With these dimensions the angle is then approximated as

$$tan^{-1}\left(\frac{0.95m}{22.35m}\right) = 0.0425rad. \qquad 8.4$$

This angle serves as input in a static OrcaFlex simulation in which the pipeline is modelled on the elevated seabed. The achievable angle per 12m pipe joint is then about $28°$. Within this simulation the pipe is modelled with an outer diameter of 562mm, and an internal diameter of 495.4mm, with two fixed ends. Without the bend sections the pipeline forms two free spans. The larger one can be seen on the right in Figure 8.14a.

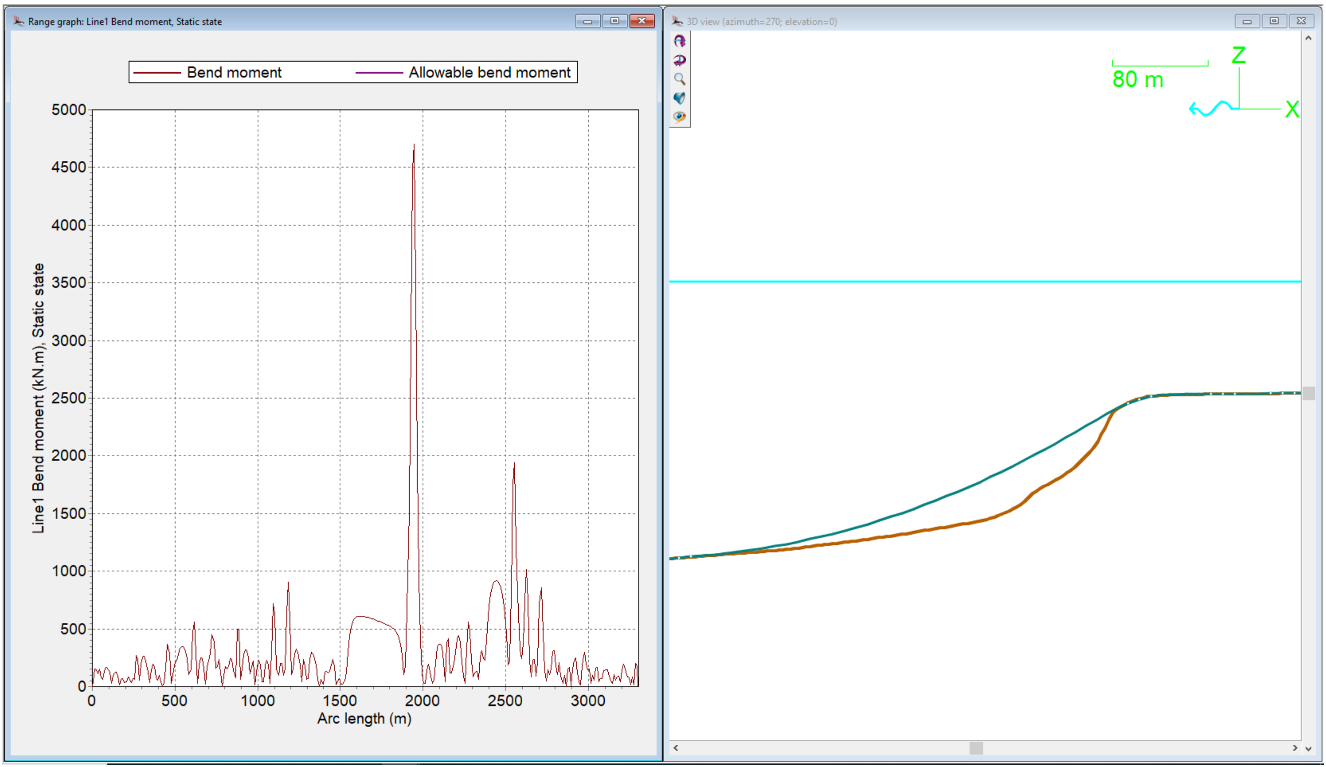


Figure 8.14a: Bending Moment and Pipe Elevation without Bends

In a second simulation the bends are included in the pipeline as "prebend" input with a curvature of 0.0424 rad/m at the shoulder of the longer slope in deeper water depth. It is assumed that two consecutive pipe joints of 12m are bent. To account for the required distance to the girth welds which is regulated by DNV the bend sections are about 10m long with a straight section between them of 1.6m length [8]. As it can be seen when comparing the drawing and the graphs in Figure 8.14a and Figure 8.14b the free span length and the moment at the free span shoulder can be reduced with the new system concept developed throughout this thesis, as well. The results are similar to the once published by Pigliapoco in his article about underwater cold bending [13].

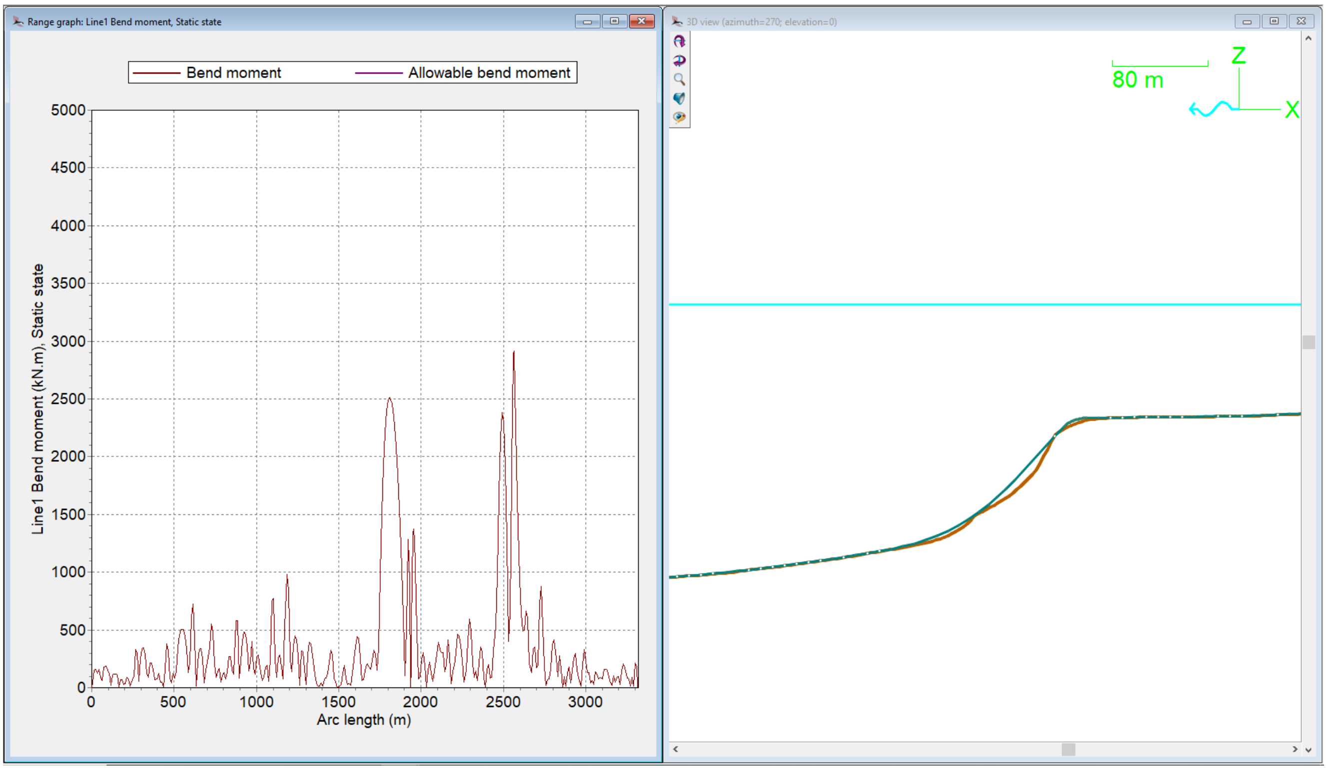


Figure 8.14b: Bending Moment and Pipe Elevation with Bends

8.4.2 Comparison of Required Forces of the Two Concepts

A simple model is used to compare the reaction forces of the two system concepts. In this model no pipe loads are considered. Only the clamps and bend dies of the concepts are modelled onto the pipe. A prescribed displacement of 40mm is applied to the outer clamps or bend dies, respectively. A linear static analysis is performed in SolidWorks with the assumption of large displacements. The pipe itself is modelled as a solid as well as the other parts, instead of the option to tread the pipe as a beam. When applying the same displacement the reaction forces are lower in the model with the rotation clamp system as it can be seen in Figure 8.15. This means that less force is required to achieve the same displacement. The difference of the reaction forces is 2.7×10^7 N. This result is reasonable as the pipe wants to move upwards in the middle of the two bend dies which are forcing the pipe downwards.

The same simulation is performed with a prescribed displacement of 100mm which leads to the same result that the rotation clamp concept requires less bending force than the caterpillar concept. It can be noted that due to the linear static simulation the resulting reaction forces are higher than in reality as deforming a steel pipe plastically cause the material to behave nonlinear. However this outcome, that one concept requires more force than the other, is assumed to be the same when using the same nonlinear material model for both concept analyses.

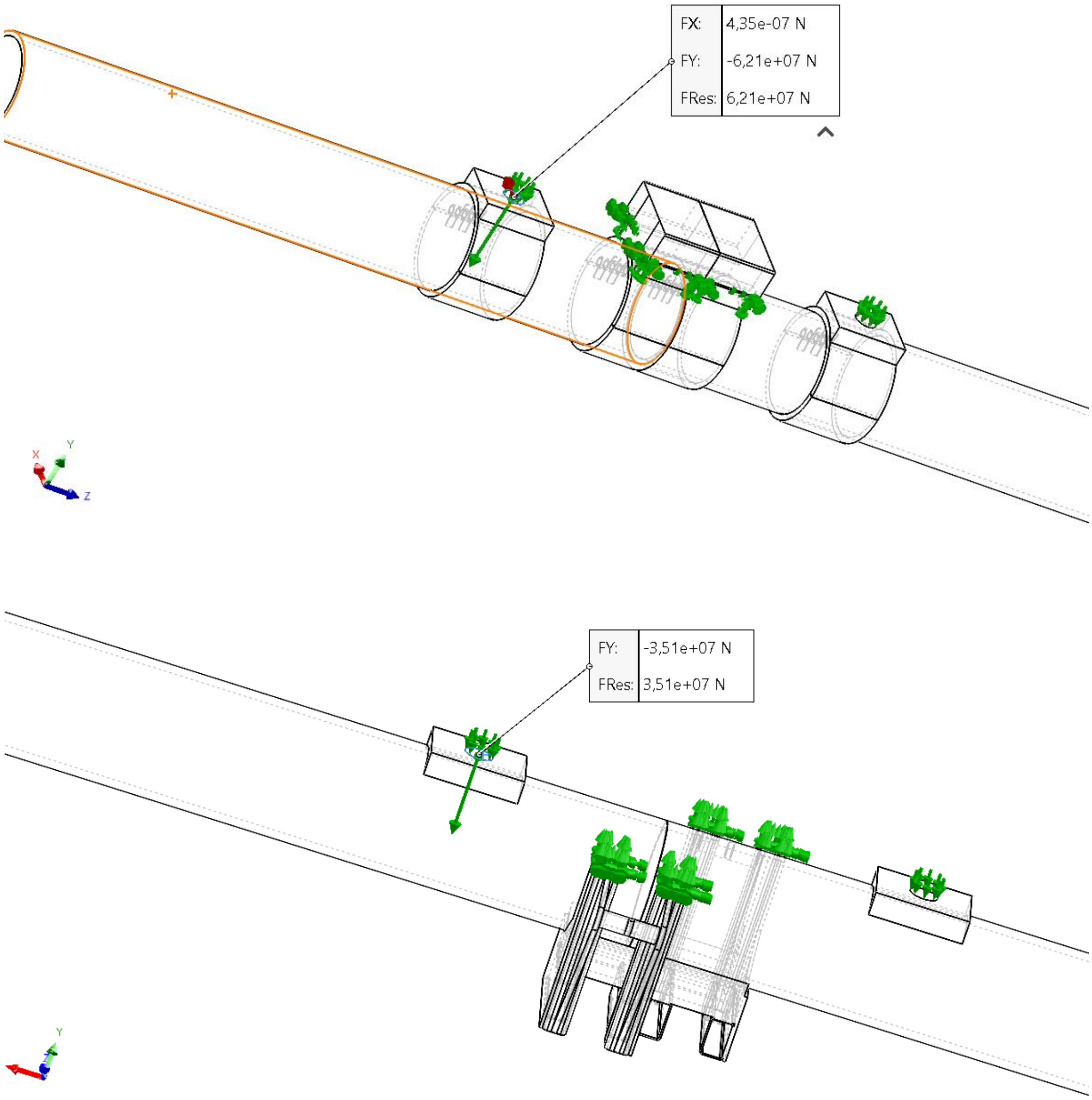


Figure 8.15: FEM Results - Reaction Forces due to Prescribed Displacement

8.4.3 Risks

In the following risks are presented together with possible mitigation measures.

- One remaining risk is the turning of the bend. The bend might land on the seabed in a wrong orientation as there is torsion in the pipeline which can lead to a rotation of the bend. A distance of at least 1.5m of the bend to the seabed leaves some space for the bend to rotate. This risk is minimized by bending the pipeline as close to the seabed as feasible but it is not eliminated.

- Operating a ROV or any other tool in deep water is a challenging task, with limited view and currents influencing the movement of the tool. Although ROV operators are highly trained people the risk of damaging the pipeline while approaching and clamping the tool around it remains.

- When using a hydraulic system there is always a risk of leakage or other system failure. This risk can be however significantly reduced by adequate maintenance.

- If the hydraulic system fails it might happen that the tool is lost as it remains clamped around the pipeline. Measures to reduce this risk is to design the tool with a fail-safe approach. It should be designed such that it can still be recovered when the system fails.

- The pipeline and coating can be damaged if the bending radius is too small and the strains get too large. This risk can be minimized by doing test bends onshore prior to the pipeline installation and using sensors to monitor the bending during the procedure.

8.5 Costs Analysis

For the remaining two concepts a cost analysis is done to get an idea about the initial investment required to build the machine as proposed in this thesis. It can be noted that these costs are a first estimate based on the level of detail of the concepts available at this design stage. Further, the costs for concept development such as electrical and hydraulic design, prototyping, testing, and certification are not included in this estimate.

For the tool itself the steel structure is considered as the material price and the construction work as 8€/kg. This price is used by an experienced senior engineer from the technical departed of Allseas to get a first idea about the price range of any steel structure. Any required wiring such as the power cables, hydraulic hoses, valves, as well as the rotatory actuators which are needed to move the clamps and hydraulic cylinders of the tool are summarized as one price for "small parts". This ballpark figure is an estimation from a ROV support engineer whos' daily task include requesting prices for ROV parts.

Prices for buoyancy and the AUXROV have been requested from suppliers by a ROV support engineer. Instead of the detailed pricing of the AUXROV the total price is included in the table below. The price for the umbilical as well as the buoyancy is based on prices of ROVs which Allseas purchased in the past.

As it can be seen from this cost analysis, the most expensive part is the AUXROV, followed by the buoyancy. Design optimization can lead to a change in costs, as the price of the steel structure, including the material and construction work, is based on the weight of the tool in this basic cost analysis. A reduced total weight leads to less required buoyancy and consequently in a further reduced cost. Assuming for example a weight reduction by 7ton for the tool structure of the rotation clamp concept then the required buoyancy reduces to 0.7ton. The resultant price difference is then about 300000€ as it can be seen from the adjusted pricing in Table 8.2 where the changes compared to Table 8.1 are written in italic font.

At this stage of concept development the cost of either of the concepts is about 2 million €, as the steel structures are of similar weight and for the small parts the same total value is assumed. However, it is assumed that the total investment in the end might be higher than this estimate. On the one hand it is possible to reduce the price due to weight and buoyancy reduction but on the other hand there are several uncertainties which might lead to a higher total investment. This may be caused by the development costs which are not included in this estimate and the small parts which are summarized as a ballpark figure. Additional design optimizations or errors might lead to unforeseen costs as well. Overall this estimate is however sufficient to get a first impression of the material costs and work costs to manufacture the tool.

Table 8.1: Cost Analysis - System Concepts

Part			Rotation clamp concept				Caterpillar Concept			
	Costs per part [EUR]		number		costs		number		costs	
300ton cyl HCG3006	13,489.65	€	2	stk	27,000.00	€	2	stk	27,000.00	€
50ton cyl RR5020	7,563.69	€	0	stk	0.00	€	4	stk	30,000.00	€
10ton cyl RR1012	2,133.42	€	2	stk	4,000.00	€	0	stk	0.00	€
Small parts	20,000.00	€	1	stk	20,000.00	€	1	stk	20,000.00	€
Buoyancy	43,043.49	€/ton	7.7	ton	331,000.00	€	6	ton	258,000.00	€
AUXROV	1,435,295.00	€	1	stk	1,435,000.00	€	1	stk	1,435,000.00	€
umbilical 3x185kW (2000m water depth)	57.00	€/m	2000	m	114,000.00	€	2000	m	114,000.00	€
steel structure: material plus construction work	8.00	€/kg	12.7	ton	102,000.00	€	11	ton	88,000.00	€
			total:		2,000,000.00	€			2,000,000.00	€

Table 8.2: Cost Analysis - Example reduced Weight of Rotation Clamp Concept

Part			Rotation clamp concept				Caterpillar Concept			
	Costs per part [EUR]		number		costs		number		costs	
300ton cyl HCG3006	13,489.65	€	2	stk	27,000.00	€	2	stk	27,000.00	€
50ton cyl RR5020	7,563.69	€	0	stk	0.00	€	4	stk	30,000.00	€
10ton cyl RR1012	2,133.42	€	2	stk	4,000.00	€	0	stk	0.00	€
Small parts	20,000.00	€	1	stk	20,000.00	€	1	stk	20,000.00	€
Buoyancy	43,043.49	€/ton	*0.7*	*ton*	*30,000.00*	*€*	6	ton	258,000.00	€
AUXROV	1,435,295.00	€	1	stk	1,435,000.00	€	1	stk	1,435,000.00	€
umbilical 3x185kW (2000m water depth)	57.00	€/m	2000	m	114,000.00	€	2000	m	114,000.00	€
steel structure: material plus construction work	8.00	€/kg	*5.7*	*ton*	*46,000.00*	*€*	11	ton	88,000.00	€
			total:		*1,700,000.00*	*€*			2,000,000.00	€

8.6 Advantages and Disadvantages

So eventually, one part of the research questions remains: Which of these two last possible concept designs found within this thesis is the most promising? To answer this question, which is part of research question d), the advantages and disadvantages of the concepts are summarized in this section.

The caterpillar concept introduced in section 8.3.2 seems like an elegant solution at first glance, using the clamping mechanism at the same time as locomotion system for the tool to travel along the pipe. The rotation clamp concept presented in section 8.3.1 leaves a more robust and simple impression.

The bending principle of these concepts is the same. It is based on a cold bending procedure for pipes as it is used onshore to prefabricate bends. Only the moving and fixed parts are interchanged compared to the press bending as introduced in section 3.1. Using this known principle makes the tool more likely to work properly than inventing a completely new mechanism.

Both concepts are technically assemblies of known elements. Hydraulic systems are used frequently not only for onshore bending machines such as the vertical bending machine introduced in section 3.2.1 but also for underwater applications like hydraulic manipulators of ROVs [58]. The clamping and locomotion system of the caterpillar concept is based on the linear winch design as shown in section 6.4.4. Similarly is the clamp design of the rotation clamp concept based on a tensioner which rotates open to release the pipeline on a vessel (section 6.4.3).

The following analyses presented in this thesis can be used to compare the two concepts in a more objective manner:

- As shown in section 8.4.2 the required force for bending is higher when using the clamping system of the caterpillar concept.
- The speed of the locomotion solution of the caterpillar concept is slower than the solution found for the rotation clamp concept, as presented in section 8.4.
- A larger distance between the small bends in the bending sequence of the caterpillar concept leads to a smaller achievable total angle per 12m joint which is shown in section 8.4 as well.

An additional risk identified, which is a disadvantage of the caterpillar concept, is a result of the clamp design. As the clamps of this concept reach around the total circumference of the pipe, this concept design is more sensitive to pipe cross-section deformations. It might happen that a clamp does not close properly if the cross-section is deformed. This risk does not exist for the rotation clamp concept as the clamps here do not reach around the total pipe.

The cost analysis in section 8.4.3 shows that the total costs of the two concepts are the same with a price of about 2 million €. As the costs are subject to change throughout more detailed development, this analysis is not assumed to be a reasonable argument for or against one of these concepts.

The advantages and disadvantages presented above can be summarised as the following selection criteria to find the most promising possible system concept.

- Simplicity and robustness
- Risk of new technology
- Required bending force
- Speed to complete one 12m pipe section
- Achievable bending angle

It is concluded from a comparison of these concepts that the rotation clamp concept is the most promising possible tool for bending marine pipelines as it is faster, requires less force due to the clamping mechanism and can produce larger angles. This result answers part of research question d).

9 REVIEW OF SYSTEMS ENGINEERING APPROACH

An alternative approach to follow when developing a new concept is the Technology Readiness Level. The Technology Readiness Level (TRL) describes the maturity of a new technology. There are in total nine TRLs defined in the literature, for example in DNV-RP-A203, which a new technology reaches one after another from "Basic principles observed an reported" to a proven concept which is used in its actual environment [97]. These TRL are summarized in Figure 9.1.

Readiness Level	Definition	Explanation
TRL 1	Basic principles observed and reported	Lowest level of technology readiness. Scientific research begins to be translated into applied research and development. (See Paragraph 4.2)
TRL 2	Technology concept and/or application formulated	Once basic principles are observed, practical applications can be invented and R&D started. Applications are speculative and may be unproven. (See Paragraph 4.3).
TRL 3	Analytical and experimental critical function and/or characteristic proof-of-concept	Active research and development is initiated, including analytical / laboratory studies to validate predictions regarding the technology. (See Paragraph 4.4)
TRL 4	Component and/or breadboard validation in laboratory environment	Basic technological components are integrated to establish that they will work together. (See Paragraph 4.5)
TRL 5	Component and/or breadboard validation in relevant environment	The basic technological components are integrated with reasonably realistic supporting elements so it can be tested in a simulated environment. (See Paragraph 4.6)
TRL 6	System/subsystem model or prototype demonstration in a relevant environment (ground or space)	A representative model or prototype system is tested in a relevant environment. (See Paragraph 4.7)
TRL 7	System prototype demonstration in a space environment	A prototype system that is near, or at, the planned operational system. (See Paragraph 4.8)
TRL 8	Actual system completed and "flight qualified" through test and demonstration (ground or space)	In an actual system, the technology has been proven to work in its final form and under expected conditions. (See Paragraph 4.9)
TRL 9	Actual system "flight proven" through successful mission operations	The system incorporating the new technology in its final form has been used under actual mission conditions. (See Paragraph 4.2.10)

Figure 9.1: The Basic Technology Readiness Levels [97, 98]

For the new concept to reach the next level an assessment is done which is summarized in Figure 9.2 below. It is based on the description of the details, definition of requirements, verification, and prospective future viability which is an assessment of risks and effort.

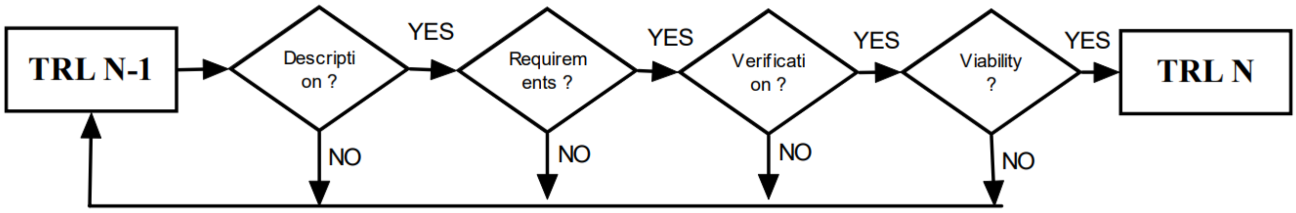


Figure 9.2: Generic Technology Readiness Assessment Steps [97]

A summary of the work which has been done during this thesis covers the assessment steps of TRL1 to TRL3. In the first stage of the system engineering approach, the Needs Analysis, it has been presented that bending large oil and gas pipelines are state of the art onshore and that different principles exists on bending pipes. As one of the operational requirements, it is defined that larger marine pipelines which have a size between 16” and 32” are to be bent to reduce free spans, as these are too stiff to follow the seabed topography. The new principle for which no functional tool exists up to this day, is to bend these offshore pipelines under water. The idea at this level is to use one of the proven pipe bending principles and develop it such that it has the capability to bend pipes in the new environment. Defining this new concept idea and its capabilities is the main idea of TRL1.

As a result of the Needs Analysis, a study which has been performed, the usefulness of the new concept is shown. Different studies presented in section 4.2.2 demonstrate that with bending offshore pipelines free span length and height can be reduced as well as bending moments at the free span shoulder. Operational and functional requirements have been defined in the Needs Analysis phase and Concept Exploration phase. Together with the identified principles on pipe bending TRL2 is concluded.

To reach TRL3 the following results have been produced. In the Concept Exploration phase of the systems engineering approach different subfunctions are defined which are summarized in the morphology matrix, Table 6.1. By combining these subfunctions twelve feasible concepts are found. In the Concept Definition phase the critical subfunctions have been analysed with a FEM analysis to verify that these parts can resists the required loads. These required forces are derived from the bending moment, which is determined with analytical calculations in the Concept Exploration phase.

In every phase of TRL it is possible to optimize the design according to the results of analyses, experiments or simulations. In general it can be said that following the systems engineering approach with the Needs Analysis, Concept Exploration and Concept Definition phase, in principle similar steps of assessment are performed as defined in the first three TRLs. The order in which specific requirements are defined to reach the next level or phase are just a matter of the approach that is followed when designing a new technology or concept.

According to the systems engineering approach the next step is the advanced development which is part of the Engineering Development stage. Within this phase the uncertainties and risks are reduced by development, simulation and prototyping which are similar to the requirements for TRL4. In the end of this engineering development stage the new system concept is qualified for production and operational use.

10
CONCLUSION

Using the systems engineering approach, the first steps of designing a new system concept have been taken. As guideline to focus the work within these phases research questions have been defined in the beginning which have all been answered and the results presented throughout the thesis.

At first a Needs Analysis has been done to analyse the state of the art onshore as well as offshore. Onshore, pipelines are being bent such they follow the topography, whereas offshore free span mitigations are necessary to ensure pipeline integrity. It has been found that there is a need for a new free span mitigation and that this mitigation can be bending pipelines under water. Different studies show that this approach can lead to a reduction in bending moments at free span shoulder and that a bend as free span mitigation is stable at steep slopes. Resulting from the Needs Analysis operational requirements of the new system concept have been formulated, and from these more specific functional requirements derived.

By engineering different solutions for the common subfunctions of the bending machine concept, twelve feasible variations of the concept design have been developed. These twelve concepts have been narrowed down to three remaining concepts in the Concept Exploration phase with a formal trade-off analysis.

While refining the design in the Concept Definition the three concepts merged into two different concepts. Sketches of these two concepts have been turned into more detailed 3D drawings to analyse the assembly of the different components. Subsequently, the impact of the pipe bending procedure with the designed concepts on normal installation procedure has been investigated as well. By comparison of the concepts according to specified criteria and results from performed analyses it has been found that the rotation clamp concept is the most promising tool to bend marine pipelines. It has been shown that with the given parameters of the new system concept the pipeline can be bent such that it follows the seabed topography. Within this analysis the same results of a reduced bending moment have been achieved as presented in previous studies.

During the Concept Exploration and Concept Definition the new system concept is designed such that most of the defined functional requirements are met. Some of these requirements are not applicable anymore due to the progressing development of the concept design. The deployment method using an AUXROV as described in section 8.1, for instance, leads to more freedom in dimensioning the machine as it is not limited by the available space on the pipelay vessel anymore these are specified in functional requirement 1.2 and 1.3. The summary of the functional requirements and where it is described how they are satisfied can be found in Appendix I.

The goal of developing a concept of an underwater cold bending system for marine pipelines has been achieved. This concept design is suitable as basis for the next phase of the systems engineering approach "Engineering Development" which includes advanced development, system optimization and testing.

10.1 Summary of Novel Aspects

Instead of making the onshore vertical bending machine waterproof a new concept design is developed which is based on the cold bending principle of press bending. The design of both final concepts as presented in section 8.3 is a new combination of known elements in a new environment:

- the linear winch principle for the caterpillar concept
- the rotating opening system of a tensioner for the rotation clamp concept

As shown in section 8.2 using two cylinders to apply the required bending force instead of one has the advantage of less deformation of the machine body. This makes it more reliable to bend large pipe diameters which require a large bending moment to be bent.

This tool is designed to be deployed from a support vessel right above the bend section and being connected to an AUXROV which provides power, cameras, sensors, and connection to the vessel. Compared to a proposed concept design of an underwater bending tool presented in section 4.2.3, where the tool is manoeuvred with a WROV, less buoyancy is required. The payload of a WROV is about 300kg, whereas the AUXROV has a 30ton lift frame. Buoyancy is a significant cost factor. Additionally, it is shown with a numerical simulation, which is presented in section 6.4.1, that it is not necessary to reduce the weight of the machine such that it does not add weight to the suspended pipeline during installation. It is found that a maximum submerged weight of the system concept of 5ton is acceptable for pipelines of sizes between 16″ and 32″.

New ideas on how to monitor the pipe and tool during the bending procedure are summarized in section 10.3. It is recommend to analyse and investigate this topic on sensors further in the detailed design stage of the systems engineering approach.

10.2 Summary of Limitations of Bending Pipes

There are some parameters to be considered during the free span assessment as presented in section 4 to decide whether this new tool as presented in this thesis is suitable to mitigate the specific free span. It can be noted that these limitations are resulting from the defined requirements and consequently the design of the tool.

As presented in section 6.3 the machine is designed such that PP and PE coating is not damaged. When considering to bend pipeline with other coating, the coating might be too brittle to remain intact due to bending.

In the operational requirements it is defined that pipelines of sizes 16″ to 32″ are to be bent. Pipelines of this diameter are typically resistant against collapse in water depth of up to 2000m but are too stiff to follow the seabed topography easily.

This machine is designed such that it can only bend the pipe in one direction such that the pipeline, being bend downward, can follow a rapid slope change from smooth to steep, for example at a continental shelf break. In the future it might be beneficial to bend the pipeline in both vertical directions when the change in slope is rapid at the end of the slope as well, from steep to smooth.

Depending on the concept design and thus the distance between the small bends, which sum up to the required bending angle, the maximum achievable angle per 12m pipe joint is calculated in section 8.4. This achievable angle is $18.5°$ for the rotation clamp concept and $11.5°$ for the caterpillar concept for a pipe diameter of 32″.

10.3 Recommendations

At last some recommendations are presented of aspects which require more attention, analysis and thought.

- **Design optimization**

As it is always the case for concepts in the early design stages optimization is done by further development and testing of the concept. This can be done with simulations or prototyping at sufficient level of detail of the design. Some details which still need to be developed for the preferred concept design are presented below.

- The wiring for the hydraulic cylinders (electrical and hydraulic engineering),
- Connection between bend dies and hydraulic cylinders,
- The docking plate between the AUXROV and the tool according to manufacturer requirement,
- Details on how to integrate the IHPU to the AUXROV or the tool such that 700bar cylinders can be used,
- Framework for buoyancy modules or block, if required

One option to reduce weight of the system concept is to change the design of the machine body. A possible optimized design is presented in Figure 10.1 as it is proposed by a senior technical inspector who is part of Allseas technical department. This design is based on the assumption that the frame is not made out of steel plates but rectangular steel tubes and it is 1m high instead of 70cm. A weight reduction by about 7ton can be achieved which lowers the cost of the system concept accordingly (section 8.5).

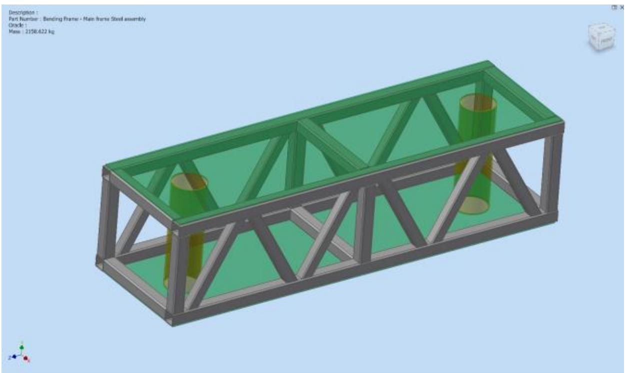


Figure 10.1: Optimized Machine Body

- **Advanced pipe analysis**

In this thesis it is focused on the development of the tool, instead of the behaviour and integrity of the submarine pipeline when bent during installation close to the seabed. As presented in the functional requirements field bends are permitted according to the DNV as long as the pipeline integrity can be maintained (functional requirement 3.1 to 3.5). This means that a minimum bending radius is allowed which is corresponding to a permanent strain of 1.25%. Buckling or wrinkling of the pipe wall needs to be prevented. To check if these requirements are met for the specific pipe in each project, it is recommended to analyse the stresses and strains, as well as local buckling of the pipe prior to project execution.

- *Stress and strain in the pipe*

The force of a hydraulic cylinder is constant as presented in equation 6.9 as pressure and area are constant during operation. As the displacement however changes it is recommended to perform a dynamic analysis of the machine modelled on a pipe section to verify the deformation and loads on the pipe, as well as the machine as total assembly. The pipe model shall include the loads, bending moment and tension, which are present in the sagbend region where the pipe is bent. Non-linear static analysis have been performed to find the strain and deformation of the pipe but these are assumed to not be correct. The details of this analysis can be found in Appendix H. To analyse the global behaviour of the suspended pipe when being bend, an OrcaFlex simulation similar to the one presented in section 6.4.1 is recommended. For a static analysis the forces on the pipe during bending can be modelled as point forces on the pipe in the sagbend with the distances to each other according to the tool design as presented in section 8.3. This analysis can be used to verify the results from the dynamic analysis of the pipe section. For the hydrodynamic analysis the machine on the pipeline shall be taken into account as it might influence VIV and drag forces during bending.

- *Ovalization collapse due to external pressure*

In the calculation of the maximum required bending moment in section 6.2 ovalization of the cross-section is neglected as the deformed cross-section leads to a reduction of moment. Ovalization, however, is an important aspect to consider, as it can lead to collapse of the pipeline under external overpressure, when it is installed in deep water.

In 2006 a finite element study was published on critical local buckling conditions for deepwater pipelines in water depth greater than 2000m. It is tested analytically how the critical collapse pressure is influenced by ovalization of the cross-section due to bending. The FEM model is hereby validated by comparing it with experimental studies which have been done for different deep water projects, Blue Stream gas pipeline and Oman-India gas pipeline [100]. It is concluded from the output data which is presented in Figure 10.2, that the collapse of the pipe is not sensitive to the prebend, even when the permanent strain is 1.5% [99]. The reason for this is the strengthening of the material due to strain hardening during bending [99].

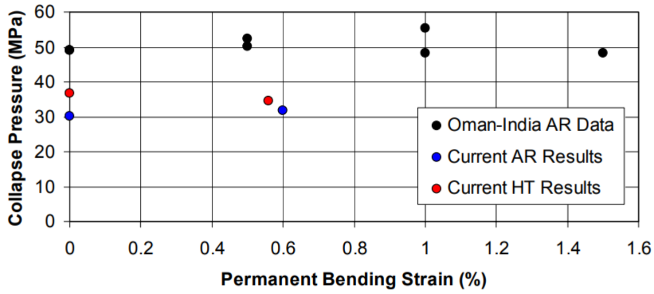


Figure 10.2: Pre-Bent Pipe - Collapse Results [99]

Although the results of these studies seem reliable, the "ovalization caused during the construction phase shall be included in the total ovality to be used in the design" of the pipe according to the DNV standard published in 2017 [8].

- *Spring back*

In the analytical static calculations the spring back of the pipe has been considered as a factor between the minimum bending radius and the residual radius. A cylinder with larger stroke is therefore selected. However to determine the correct stroke required to achieve the desired bend radius more detailed calculations should be performed for the specific pipe and project.

- **Influence of pipelay vessel thrusters**

The assembly of the tool and AUXROV are deployed from the support vessel right above the sagbend. Depending on the water depth the sagbend might be so close to the pipelay vessel that the thrust of the thrusters can influence the behaviour of the assembly when it is lowered through the splash zone and the first few meters below the sea surface. As the thrusters are needed to maintain the tension in the pipeline and move the vessel forward for the next pull they cannot be shut down for the deployment of the bending tool. Therefore it is recommended to investigate the influence of this flow on the assembly during deployment. A solution to avoid this problem might be to bend the pipeline after installation. As the bending is performed with a support vessel the bending procedure can be performed independently of the pipeline installation itself. The free span needs to be temporarily mitigated with buoyancy modules or supports to ensure pipeline integrity until the bending is completed.

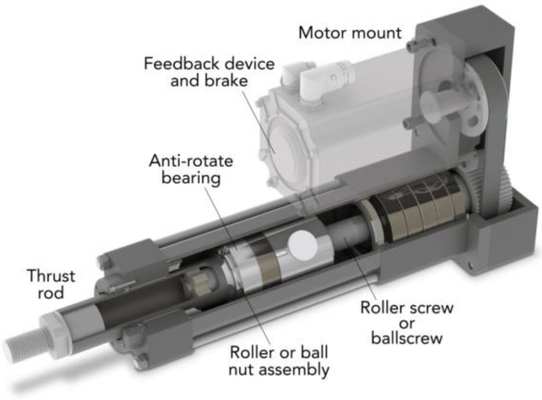
- **Test bends**

As it is the case for welds, it is recommended to produce a number of test bends onshore with the bending tool. Pipe sections which have the same properties and quality as being installed at a later stage in the new project shall be used. The test bends can be used to verify the required stroke of the hydraulic cylinders to achieve the desired bending radius per bend. This way the analytically obtained parameters are confirmed and can be included in the bending procedure for the offshore crew. An additional benefit is that the functionality of the machine can be tested, and possible errors be detected and repaired.

- **Electrical actuators**

There is an ongoing discussion about the use of a hydraulic system versus a full electric system for offshore applications. A full electric system requires less maintenance and there is no risk of leaking. The hydraulic cylinders could be replaced by electrical linear actuators for the new system concept which are displayed in Figure 10.3. The question remains however if there are electrical linear actuators available which can provide the required force. It is therefore recommended to get in contact with suppliers to investigate this possibility properly.

Electrical Linear Actuator



Hydraulic Cylinder

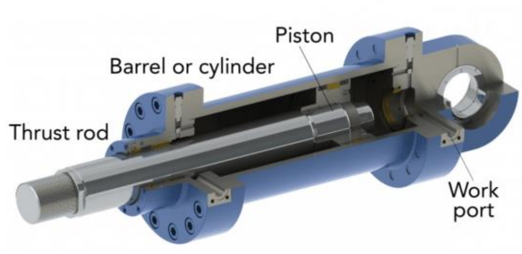


Figure 10.3: Electrical Linear Actuator vs. Hydraulic Cylinder [101]

- **A note on sensors**

To control the quality of the bend during the bending process, sensors can be used. Different ideas are presented in the following which resulted from discussions with different engineers, the unit head of the survey department and unit head of the ROV department. However, which sensors the most suitable are and how exactly they should be integrated in the new system concept is not defined yet.

A sensor which is usually used for strain measurements onshore is a strain gage. These are stripes glued to the object and measure strain either as change of electrical resistance or change in light transmission through optical fibres. As these need to be placed exactly at the bend section and the strain in the coating instead of pipe material is measured another option might be more feasible.

One possibility to monitor the bending procedure is to check the stroke of the hydraulic cylinders. A solution to do this is to use cameras which show the cylinder rod on which stripes indicate how far the cylinder is extruded. As the bending procedure is performed close to the seabed all measurements which are performed with cameras or echosounders are considered less accurate because of soil particles which are stirred up by the moving pipe in the sagbend or the thrusters of the AUXROV. These might be however suitable as redundancy, to verify the measurements of a second sensor.

Most accurate is direct measurement. A donator wheel is used for example to measure the travelled distance of a ROV when it travels along a pipeline. This measuring wheel, as shown in Figure 10.4, can be used to measure the cylinder stroke as well. When it is installed such that the wheel touches the rod, the distance the rod is extended is measured by counting the turns the wheel has made. The smaller the wheel the more accurate the measurement gets. One risk using this sensor is the slipping of the wheel and therefore not accurate measurements. This is why it should be considered to use two different measuring methods, to receive different sets of data to compare and get a better idea of the measurements.

Figure 10.4: Measuring Wheel [102]

- **Redundancy**

Another question which has not been sufficiently answered in this report is: What happens if the hydraulic system fails? A typical fail safe mechanism for single acting hydraulic cylinders is the spring return. In case of the rotation clamp concept these springs inside the cylinders which open and close the clamp might not be strong enough. As one clamp weights about 860kg an additional spring can be considered to compensate for this weight. When the clamps close the spring extends such that if the system fails it retracts and opens the clamp. The tool and AUXROV assembly can then be retrieved and repaired onboard the vessel.

In case of the caterpillar concept it might be beneficial to use electrical rotary actuators which open and close the clamps. In the event of a power outage these can be activated with the power provided by batteries. These can be installed on the tool or AUXROV as redundancy measure. The clamps can then be opened and the machine recovered in a similar manner as the rotation clamp concept.

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APPENDIX A

Phases of Onshore Pipeline Construction

The onshore pipeline construction can be categorized into the following phases [26]:

1) Route selection
   The most favourable pipeline route is selected with respect to other infrastructure systems (e.g. highways or railroads), populated areas and environment (e.g. wildlife, soil type).

2) Regulatory process
   The selected route has to be approved by the authorities.

3) Design
   During design phase all details about the pipeline system like material, coating and size of the pipe are specified as well as the details about the installation.

4) Pipe fabrication
   Pipe sections are fabricated in steel rolling mills and coating is applied. After testing the sections are transported to the building site.

5) Site preparation
   The pipeline route is cleared of trees, boulders and bush where necessary. The working area is prepared such that the working surface is accessible for the construction equipment.

6) Pipe stringing
   The pipe joints are laid out along the route according to the design plan.

7) Trenching
   Where the pipeline is buried into the ground a trench is dug along the pipeline route. The soil is stored next to the trench such it can be used for backfilling.

8) Bending
   Different options for pipe bending are presented in detail in section 3.

9) Welding and weld inspection
   The pipe joints are aligned and welded together following strict specifications.

10) Field coating
   The end of the pipe sections are not coated in the factory to allow for welding. To prevent corrosion the coating has to be applied over the completed weld at the building site.

11) Lowering and backfilling
   When it is required to bury the pipeline, it is lowered into the trench after welding and coating. The excavated soil is then returned into the trench.

12) Pressure testing
   The pipeline is filled with water and the internal pressure is raised to a specific level which is above the operating pressure. The aim is to find any defect in the pipeline.

13) Site restoration
   The pipeline route is restored as close as possible to its original condition.

APPENDIX B

Prefabricated Bends

Cold Bending Procedures

3) Compression bending

The schematic in Figure B1 shows the principle of compression bending. Here the pipe is clamped between the stationary roll and the clamping piece. The pipe is bent by rotating the sliding carriage around the stationary roll. Because of this setup the bending radius of the pipe is the same as the radius of the stationary roll. [28]

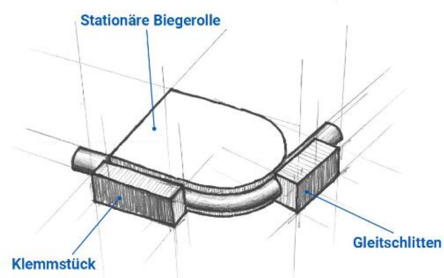


Figure B1: Schematic Compression Bending [28]

4) Rotational tension bending

The fourth cold bending principle introduced in this thesis is the rotational tension bending or rotary draw tube bending which can be seen in Figure B2. Here the pipe is clamped between the clamp die and the bend die. By rotating the clamp die and the bend die, the pipe is bent to the desired radius which is the same as the radius of the bend die. Additional elements like a wiper die and a mandrel can be used to improve the quality of the bend. [28]

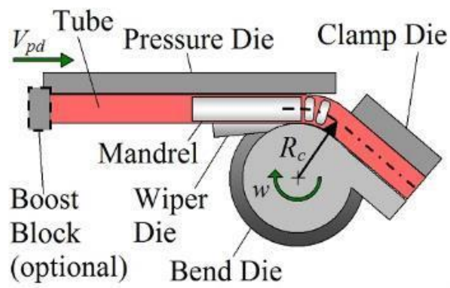


Figure B2: Schematic Rotational Tension Bending [103]

A mandrel is a piece of equipment which has a cylindrical shape as it can be seen in Figure 3.3b. Before the bending process it is inserted into the pipe and extended such that it engages the pipe walls. It is supporting the pipe wall from the inside and prevents the deformation of the cross-section during bending. [30]

A wiper die looks like a half shell which tapers into a wedge (Figure B3). The diameter of the half shell matches the outside diameter of the pipe. When engaged to the pipe the feathered edge positioned close to that area where the pipe starts to deform plasticly, prevents the forming of wrinkles as there is no space on the inside of the bend. The displacement of the pipe wall is controlled. [103]

Figure B3: Wiper Die [104]

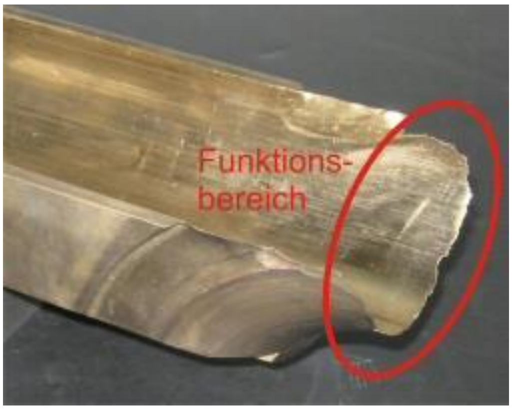


Figure B4: Used Wiper Die [104]

This bending procedure is used when a tight bending radius is required as well as when little tolerances are allowed. It is suitable for high quality alloys [28]. Tests performed at the University Siegen in Germany show that bending without wiper die is not possible because of wrinkling [104]. The wiper die has a mechanical impact on the pipe as it can be seen from the signs of wear on the wiper die presented in Figure B4. This leads to the conclusion that this procedure is not feasible for coated pipelines. It is however a good solution for prefabricating bends at a factory.

Hot Bending Procedure

As an example of a hot bending procedure, induction tube bending is explained as it is performed by the pipe bending plant Salzgitter Mannesmann Grobblech GmbH. After inspection, performed when the pipe arrives at the factory, the pipe section is guided into the bending machine. A schematic of the bending procedure is shown in Figure B5. It can be seen that the end of the pipe section is clamped in the bending arm of the machine which "describes a circular arc around its pivot point" [21]. The length of the arm determines the radius of the bend as it controls the direction of the bending process. A feed unit provides the bending force and pushes the pipe through an induction coil which heats the pipe. Cooling nozzles limit the heated area on the pipe.

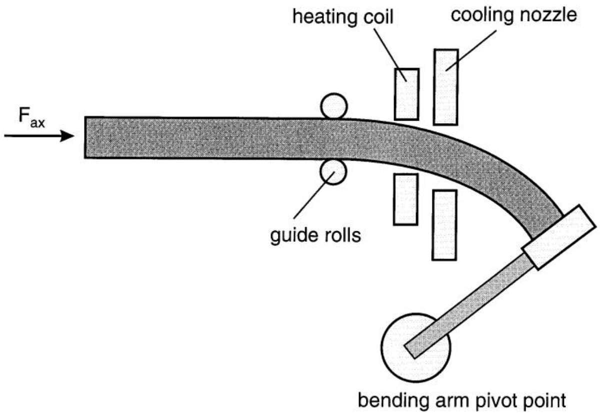


Figure B5: Schematic Induction Bending [27]

Figure B6: Heat Treatment [21]

To optimize the microstructure and mechanical-technological properties of the pipe the next step in the process is the heat treatment. The pipe joint is heated in a furnace to a predefined temperature and then cooled in a large water basin (Figure B6) or in air with the use of ventilators depending on the required properties. After testing and inspection procedures the pipe coating is applied and the pipe is shipped to the customer.

APPENDIX C

Bending Marine Pipelines with Sleeves

In a patent from 2012 it is proposed to install a sleeve as bending restrictor and utilize external force to produce controlled bending of a pipeline beyond elastic limits [105]. The procedure which is shown in the upper drawings of Figure C1 is explained as follows: The bending restrictor is installed on the pipeline such that inner and outer segments are connected and the sleeve may be welded to the pipeline. Then the pipe is lowered as to the usual pipelay procedure, S-lay or J-lay. From a separate vessel a weight, 107, is installed on the shoulder of the slope as counter weight. Another weight, 110, is lowered from the vessel onto the pipeline. Through the external applied force the pipe bends and the curvature is restricted by the bend restrictors. The weights as well as the sleeves, 102 and 104, are left in place to enhance on-bottom stability of the pipeline. [105]

The lower part of Figure C1 shows the composition of the bending restrictor. The sleeve consists of inner and outer segments which are connected through pins which go through keyways. Thus, when the sleeve extends the pin moves to one side of the keyway as it can be seen in "fig 4" and in compression it moves to the other side of the keyway as seen in "fig 5". The bending of the pipe is restricted by the ability of the sleeve to extend and compress. This is limited by the diameter of the pins and the length of the keyways. [105]

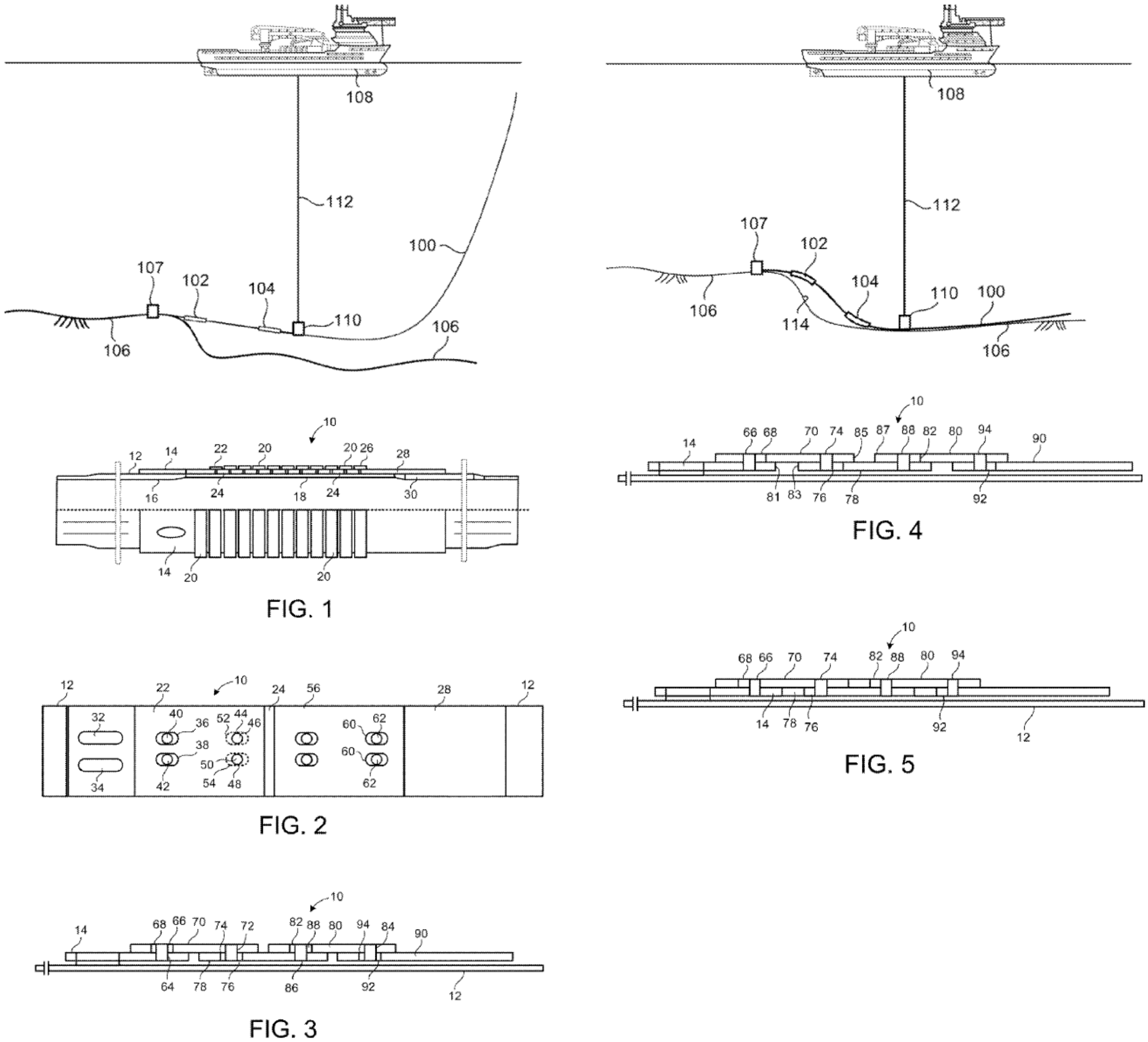


Figure C1: Underwater Pipe Bending Utilizing External Force with Bending Restrictors [105]

In 1992 a, now expired, patent was filed which describes a similar idea of using a sleeve as controlling\stopping means for bending [106]. The sleeve can be installed on pipelines without concrete coating and with concrete coating once the coating is removed in the section in which the sleeve is welded to the pipeline. This sleeve is made out of tubular sections ("Fig. 1." of Figure C2) which are interconnected by a connecter which is depicted on the right of Figure C2. The weight of the sleeve replaces the weight of the concrete to ensure on-bottom stability.

In contrast to the more recent invention the cold bend is achieved by weight loading the pipeline on the inside at the bending zone. As it can be seen in Figure C2 "Fig. 2." this internal weight can be a chain, 17, which is connected to a winch, 20, on the pipelay vessel such that the bending process can be performed during the pipe installation. It is stated that it is necessary to perform the bending during pipelay as longer length is required when the pipeline follows the seabed. [106]

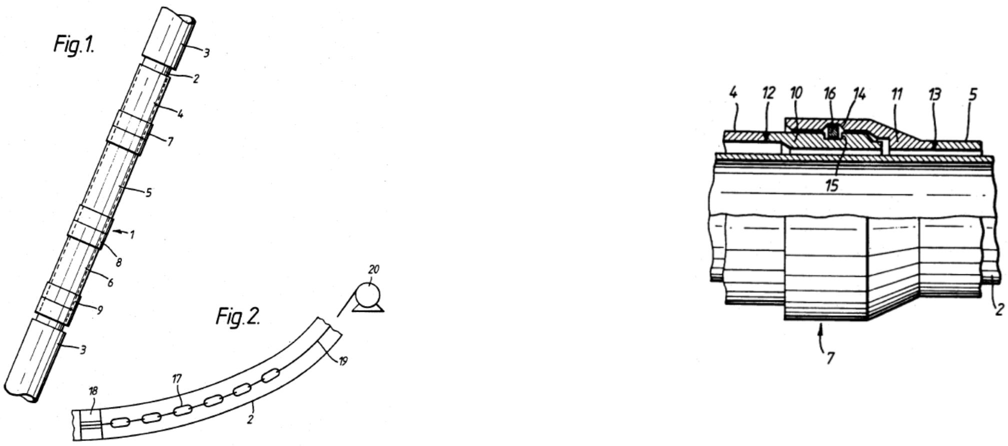


Figure C2: Pipe Bending with Internal Weight [106]

APPENDIX D

Moment Calculation: BendPipe Output

Loadcase: Moment-Curvature
Run Date and Time: 31/05/2023 - 11:00:28

Analysis type: Moment-Curvature calculation

Main Pipe:

Yield moment	[kNm]	7842.2
Yield radius	[m]	187.8
Maximum bending moment	[kNm]	10336.0
Maximum stress (top)	[MPa]	458.58
Maximum strain (top)	[%]	0.741
Residual strain (top)	[%]	0.454
Residual stress (top)	[MPa]	-136.95
Residual radius	[m]	89.4

Moment-Curvature listing for bare pipe:

k/ky	M/My	k	M
[-]	[-]	[1/m]	[kNm]
0.015	0.015	0.000080	117.5
0.225	0.225	0.001200	1762.6
0.436	0.435	0.002320	3407.8
0.646	0.644	0.003440	5052.9
0.856	0.852	0.004560	6683.5
1.067	1.010	0.005680	7922.6
1.277	1.100	0.006800	8624.9
1.487	1.156	0.007920	9065.9
1.698	1.195	0.009040	9369.7
1.908	1.223	0.010160	9592.9
2.118	1.245	0.011280	9765.6
2.328	1.263	0.012400	9902.9
2.539	1.277	0.013520	10015.0
2.749	1.289	0.014640	10110.7
2.959	1.300	0.015760	10192.7
3.170	1.308	0.016880	10261.3
3.380	1.316	0.018000	10323.3

Ramberg and Osgood curvefit:

Coefficient A		0.0420
Exponent B		14.0099
Largest error	[%(k/ky)]	2.908
Average error	[k/ky]	0.0183

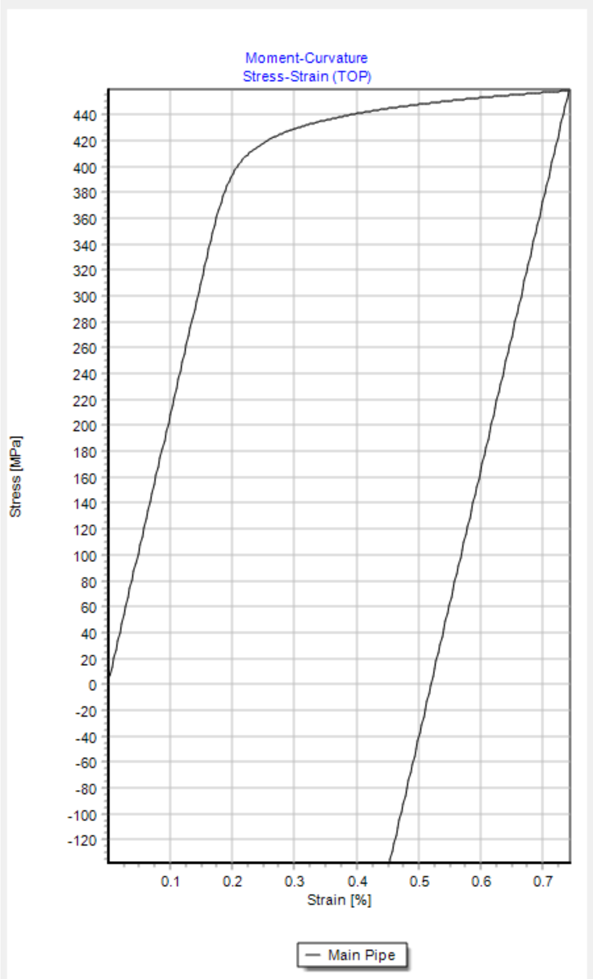


Figure D: Output BendPipe - Required Bending Moment for 32'' Pipe (X65) and 55m Minimum Radius

APPENDIX E

Calculations to Prove the Integrity of the Pipeline for Internal Bending Concept

As the internal machine adds weight to the pipeline calculations have been done to ensure the integrity of the pipeline during installation when the machine is inside the pipeline. The pipeline which is modelled in this simulation has the size of 16″ which is the smallest pipeline diameter of the range of pipes sizes that shall be bent according to the operational requirements. The wall thickness is chosen as 2cm which leads to a D/t ratio of about 20. This value is within the range of 15≤d/t≤45 for which the buckling check according to DNVGL-ST-F101 is valid. The buckling check in OrcaFlex is in compliance with this standard.

The machine is modelled as an attachment in form of a stiffener which is inside the pipe as the outer diameter of the stiffener is smaller than the internal diameter of the line pipe. The machine is modelled in the sagbend where the bending shall take place. The outer diameter is set to 30cm such that there is 3cm space surrounding the machine, leaving room for part of rollers or tracks. The inner diameter is set to 0 such that the machine is a solid cylinder with the length of 6m. The bending stiffness of the machine is then

$$EI = 83497.64\ kNm^2$$

with
Youngs modulus E = 210 000 000kPa
Inertia $I = \frac{\pi}{4} r^4$

The simulations are performed for different water depth which are 100m, 900m and 2000m and the buckling check and the strains are analysed. The limiting strain for the overbend is typically 0.35%. The allowable strain is higher in the overbend compared to the sagbend which is typically 0.15% as the pipeline is supported by the stinger in the overbend. In the figures below two different results for the buckling check are shown. Both follow the DNVGL-ST-F101 standard but one graph shows the displacement controlled results which are applicable for the pipeline section in the overbend on the stinger. The second graph shows the buckling check for load controlled which is decisive for the sagbend [8]. As it can be seen in Figure E1 to Figure E3 below all the buckling checks are acceptable as well as the strains in the line pipe.

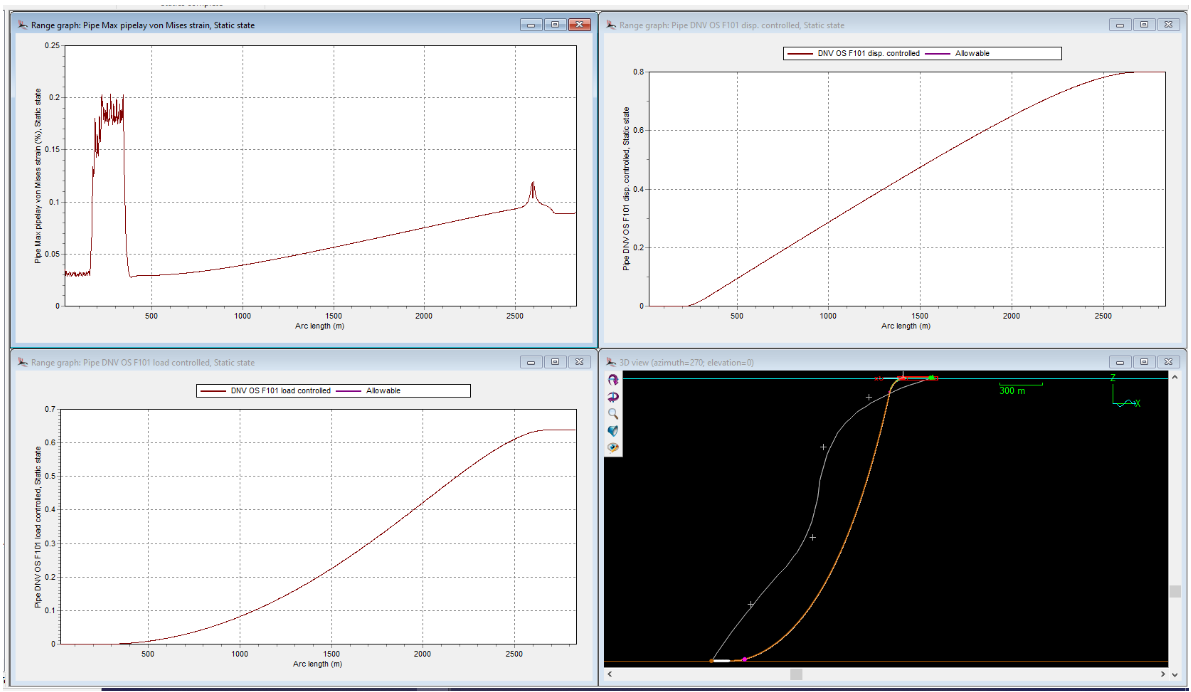


Figure E1: Internal Bending Concept – Water Depth 2000m

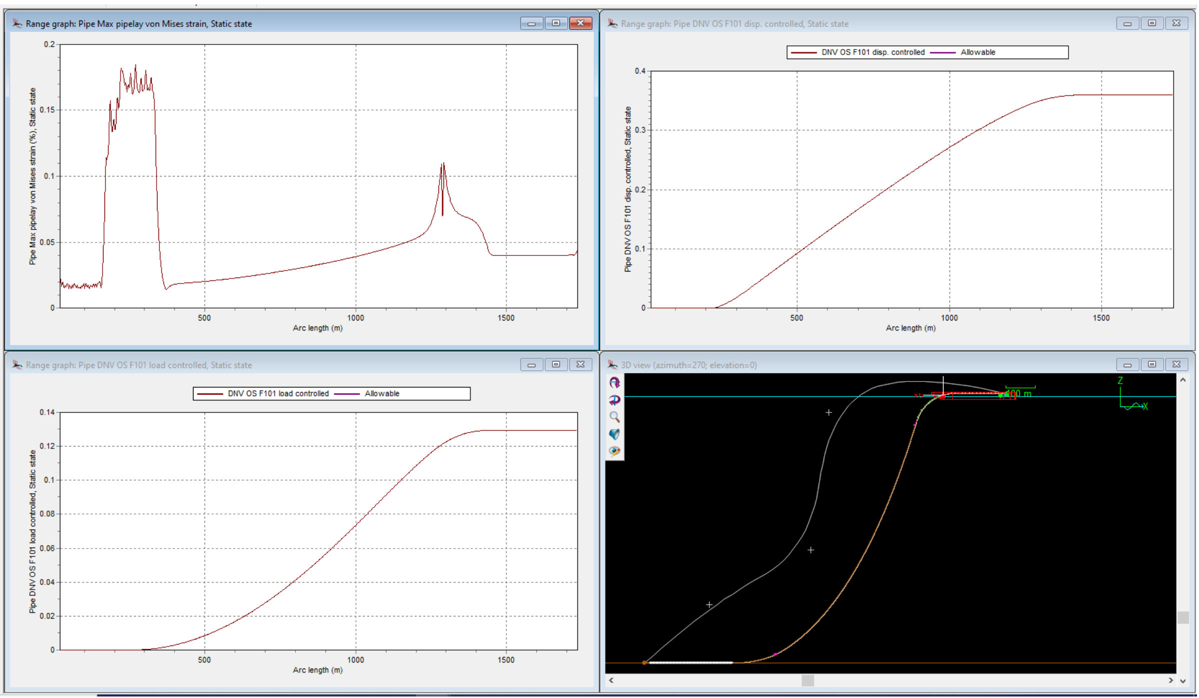


Figure E2: Internal Bending Concept – Water Depth 900m

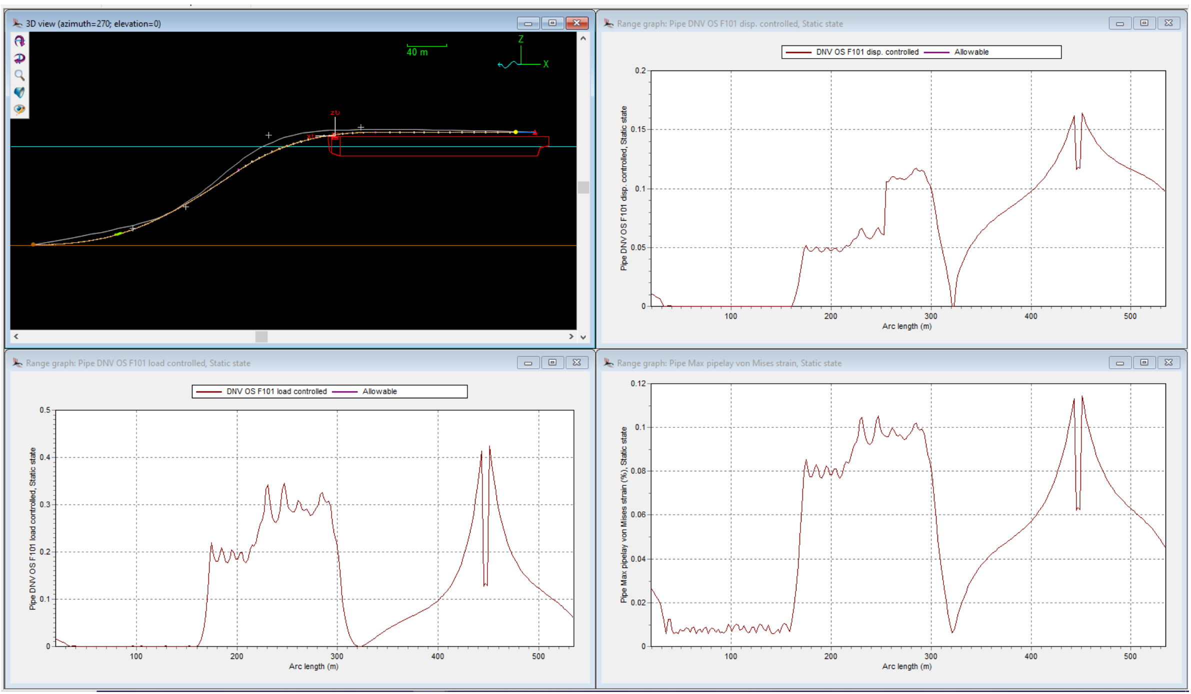


Figure E3: Internal Bending Concept – Water Depth 100m

In Figure E4 below the machine is modelled on the stinger which has the radius of 120m it can be seen that the pipe before and after the machine has no contact to the roller box. Roller boxes which have pipe contact are green, whereas those which have no pipe contact are purple. However, the strain and buckling check are still in an acceptable range which leads to the conclusion that the length of 6m of the rigid machine is still within the limitations such that the pipeline is not damaged.

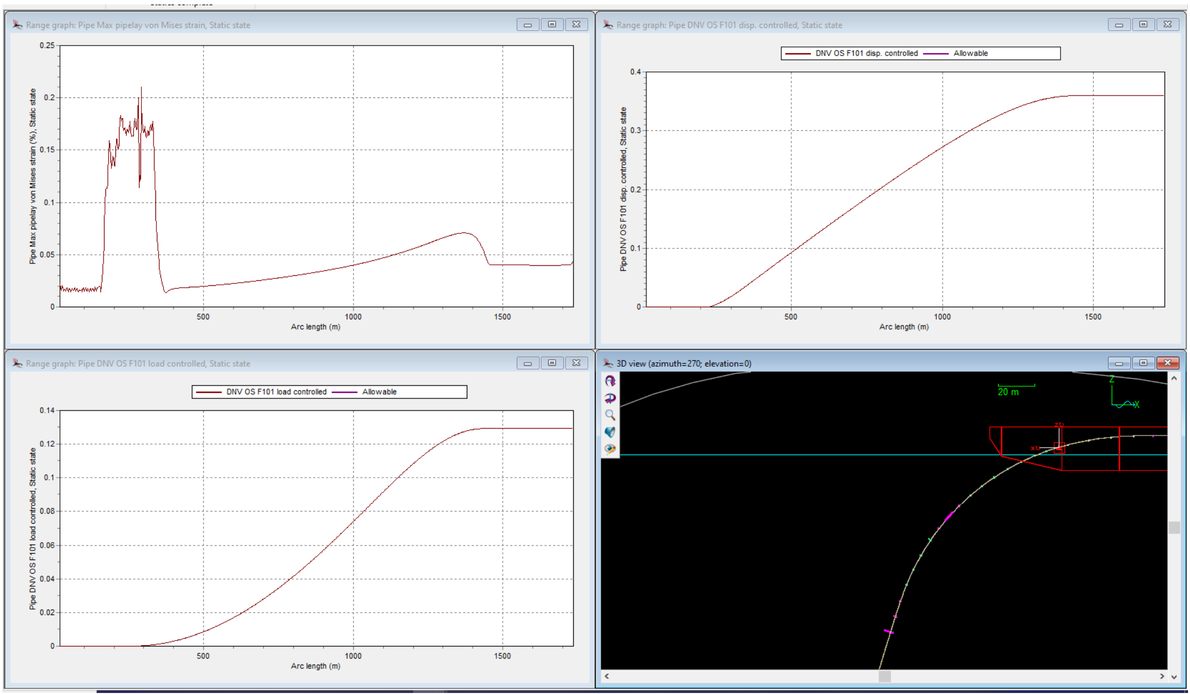
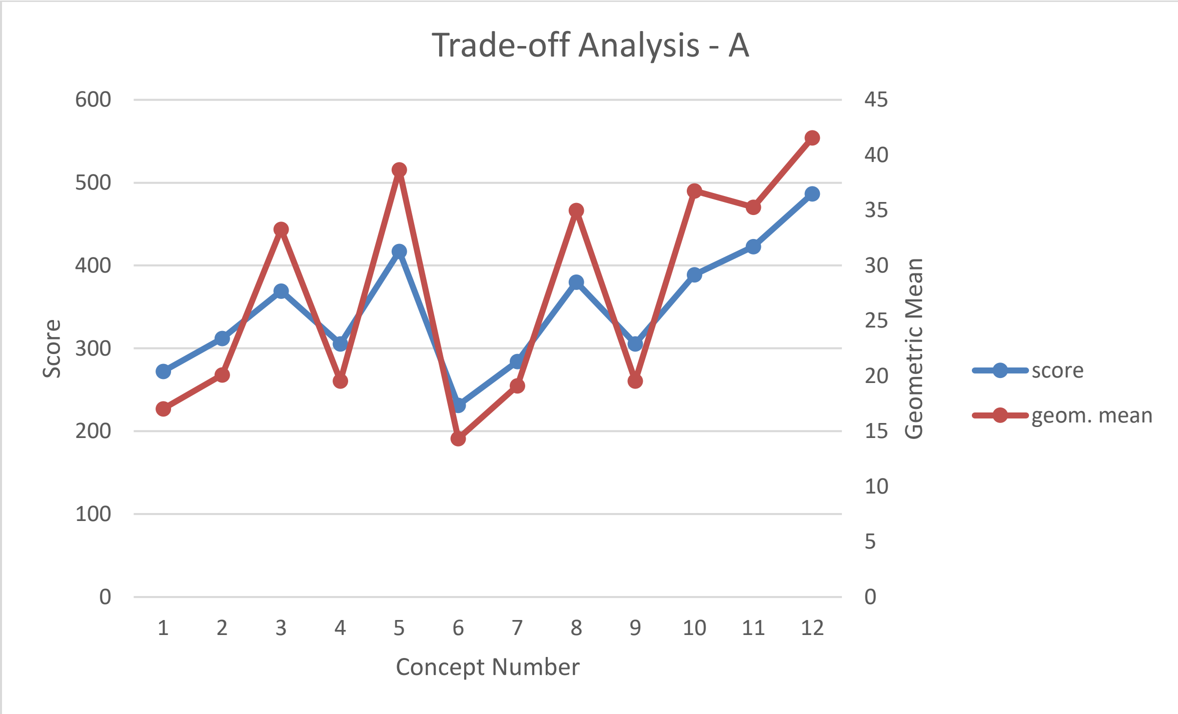
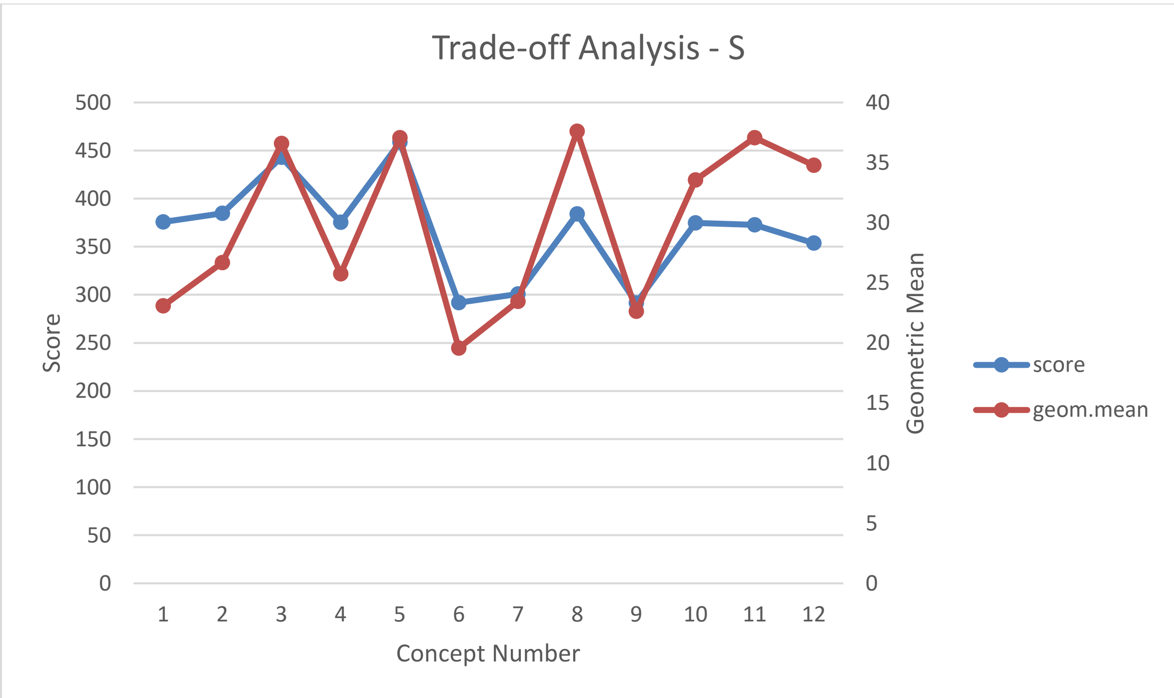


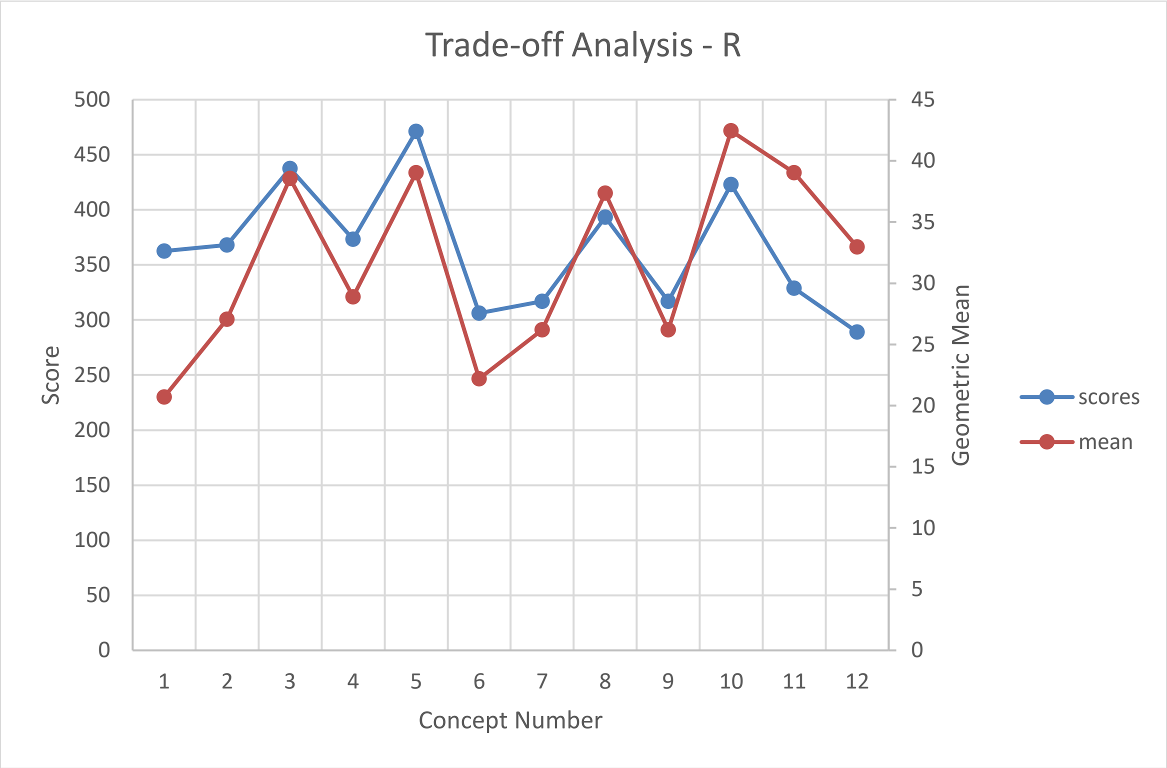
Figure E4: Internal Bending Concept – Water Depth 900m, Machine on Stinger

APPENDIX F

Results of Different Trade-off Analyses







Trade-off Analysis - R
Score
Geometric Mean
Concept Number
500
450
400
350
300
250
200
150
100
50
0
45
40
35
30
25
20
15
10
5
0
1
2
3
4
5
6
7
8
9
10
11
12
scores
mean

APPENDIX G

Calculation of Time Estimate for Bending

Table G1: Extension and Retraction Times Hydraulic Cylinders

	300ton [64]	**50ton** [89]	**10ton** [89]
piston area push [cm^2]	456.2	71.2	14.5
annual area pull [cm^2]	151.4	21.5	4.8
Stroke [mm]	150	511	305
Stroke [cm]	15	51.1	30.5
extension time [sec]	*13.686*	*7.27664*	*0.8845*
retraction time [sec]	*4.542*	*2.1973*	*0.2928*

Table G2: Calculation of Time to Bend One 12m Joint

	handover system			**rotating clamp system**		
			notes			*notes*
clamp around pipe	5.00	sec	from titan gripper 6rpm 1/4 rev needed open and close	*11.18*	*sec*	open and close the clamp
introducing bend	*18.23*	*sec*	extend and retract 300ton cyl.	*18.23*	*sec*	
travel to next bend	*75.25*	*sec*		0.25	knots	if use thrusters
				0.13	m/s	max is 4 knots for ROV
				7.39	*sec*	
distance between bends	1.50	m		0.95	m	
number of bends per joint	7			11		
required time bending 1 12m joint	614.11	sec		287.22	sec	
	10.24	min		4.79	min	

APPENDIX H

Numerical Pipeline Analysis

In SolidWorks a non-linear static analysis is conducted as plastic deformation of the pipeline is the goal and the material behaves nonlinear when it is deformed in the plastic region as it can be seen from the stress strain curve.

The model to simulate the strain and the pipeline deformation is set up as follows: The pipeline used for this simulation is the 32″ pipeline with the wall thickness of 39mm. This pipe is under tension along its total length, which resembles the bottom tension during installation, resulting from the friction between pipe and soil, the weight of the already installed pipeline, and the vessel pulling on the pipeline during installation to prevent buckling. As the pipeline is bend close to the sagbend there is a bending moment in the pipeline as well (Figure 6.1). This moment is applied to the joint at one end of the pipe while it is fixed in translational directions on the other end as shown in Figure H1. This fixture means that the pipe is free to rotate at both ends. The pipe section length is assumed such that the fixtures in the simulation have a sufficient distance to not influence the deformation and loads in the pipe section. To be able to apply the moment as well, the pipe is treated as a beam in the software.

To simulate the tool, bending the pipe, the bend dies and the clamps of the rotation clamp concept are modeled onto the pipe section. The clamps are, as in the linear static simulation to find the stresses in the clamp, fixed at the upper end of the vertical beams (section 8.3.1). On top of the bend dies the contact area of the hydraulic cylinder is projected. This area is used to apply a prescribed displacement of 40mm. These 40mm are analytically estimated according to equation 6.10, neglecting the springback of the material.

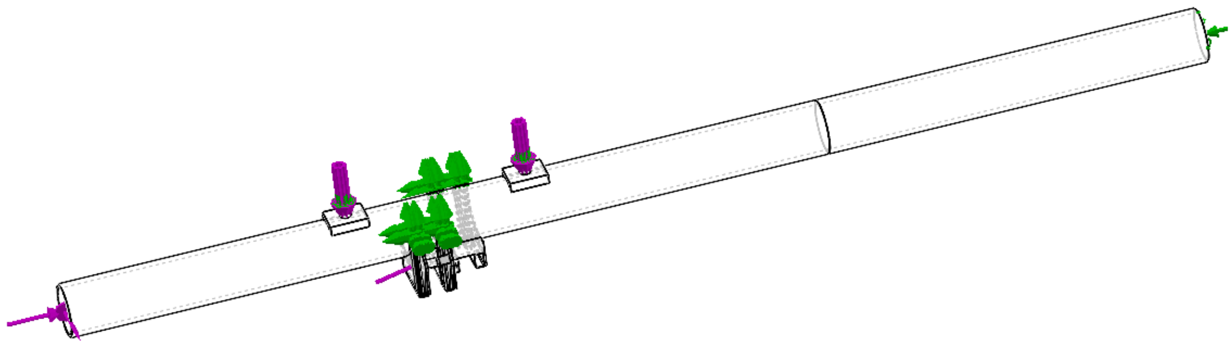


Figure H1: FEM Analysis Pipe - Applied Fixtures and Forces

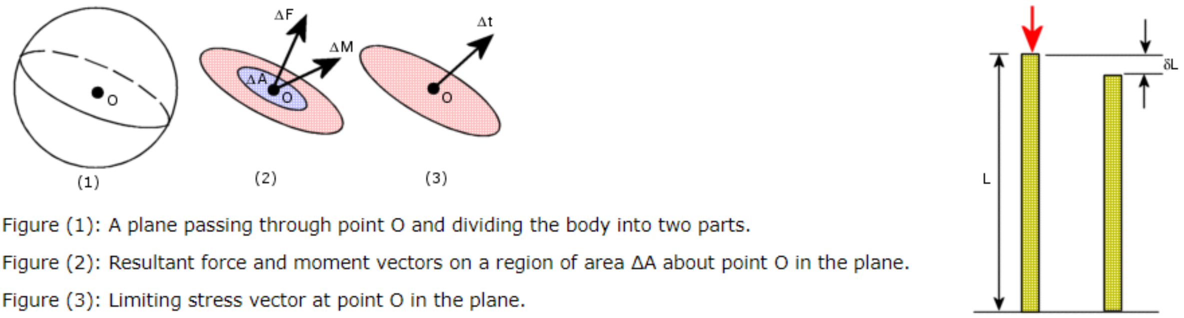
For the applied material of the pipe, it has been tested to use the exact stress-strain curve of X65 as it can be extracted from BendPipe (Appendix D). The problem is that SolidWorks uses only the plastic part of the stress-strain curve starting with the yield stress to get the material properties for the calculation. In case of the stress-strain curve from BendPipe this means that only 3 points of this curve are available and the calculation does not converge properly as it can be seen from the resulting deformation. Consequently, material of the SolidWorks library with pre-defined properties are selected instead in order to converge the simulation. The two materials for which the analysis has been performed have a yield strength of 325MPa and 620MPa.

Additionally to different materials, different tensions and their corresponding sagbend moments are applied for different water depth. These parameters are output of the same OrcaFlex calculation used before to estimate the impact of the machine on the pipe in section 6.4.1. The values used in the FEM analysis are listed in Table K1 below.

Table K1: Moments and Tensions for FEM Analysis

Water depth [m]	Bending moment in sagbend [kNm]	Bottom tension [kN]
2000	2454.75	1095.68
900	3518.00	665.00
60	2793.03	421.109

From the output of the nonlinear static analysis only the displacement and strain could show reliable values. This is the case because the forces the program assumes are not realistic. In SolidWorks the force increases when the displacement becomes larger. In reality hydraulic cylinders are used to apply the force and the displacement. Thus assuming that the forces accumulate with larger displacement is not correct as the dynamic extension of the cylinder is not taken into account. To calculate the correct stresses a dynamic analysis has to be performed. The wrong forces are the reason for the excessive stresses in the pipeline when comparing them with the strain results. The reason is that SolidWorks uses the forces to calculate the stresses, but displacement to calculate the strain, as presented in Figure H2.



$Stress = \Delta F/\Delta A$ as ΔA approaches zero

$Strain = \delta L/L$

Figure H2: Stress and Strain Calculation SolidWorks [107]

In Figure H3 to Figure H6 the results of the simulation of the pipeline installed in 2000m water depth with the material yield strength of 620MPa are shown. As described previously the stresses are unrealistically high as expected and the deformation in y-direction seems reasonable. However, the strains are close to zero which is assumed to be wrong. Although the pipe is restrained in movement it deforms in such a way that the resultant displacement is zero which consequently means no strain.

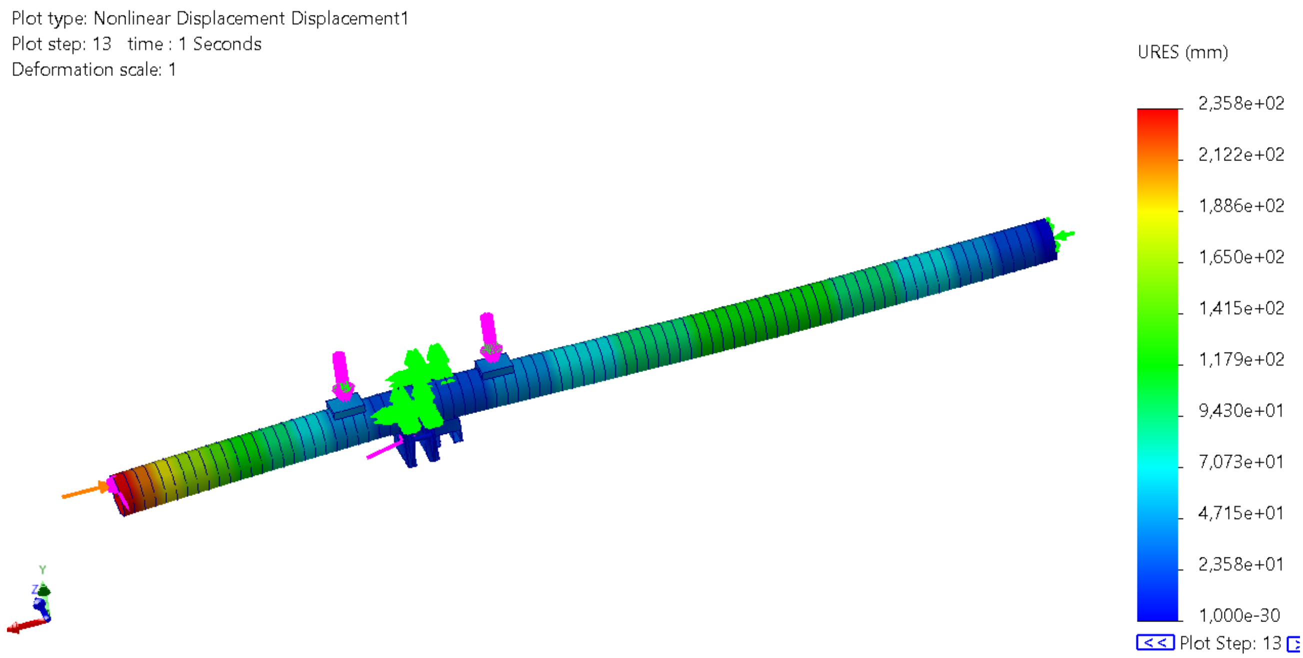


Figure H3: FEM Analysis Pipe - Resultant Displacement

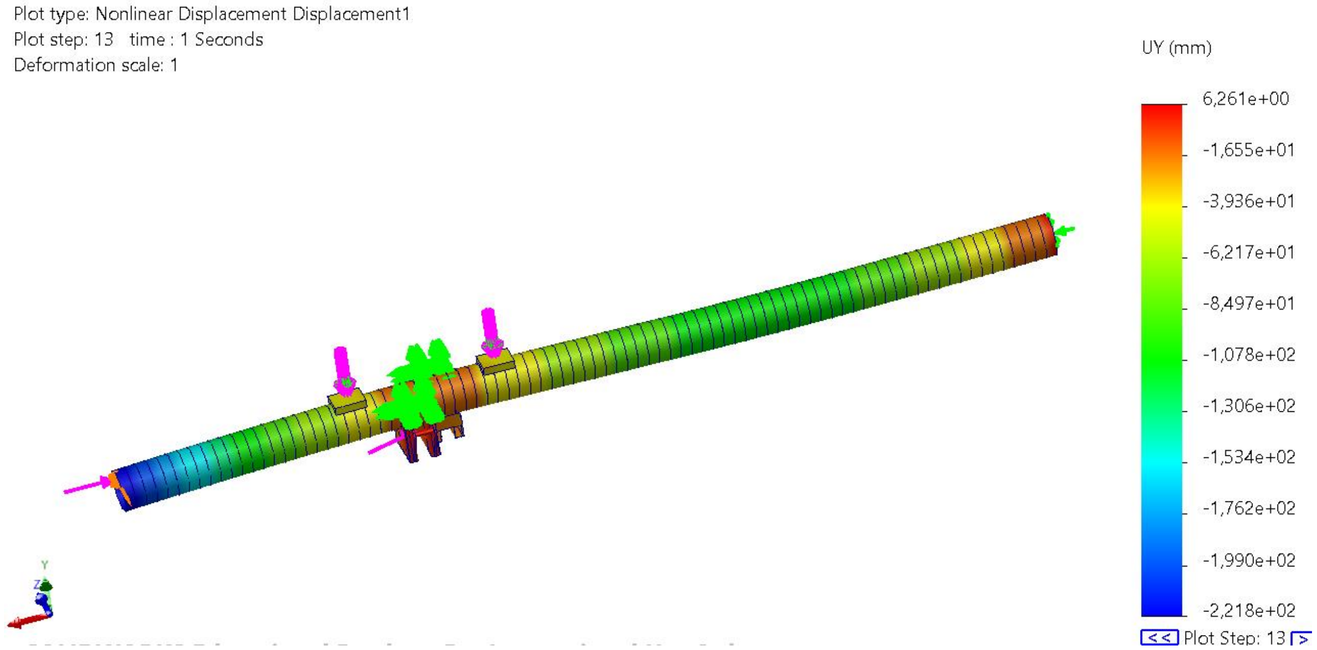


Figure H4: FEM Analysis Pipe - Displacement Y-Direction

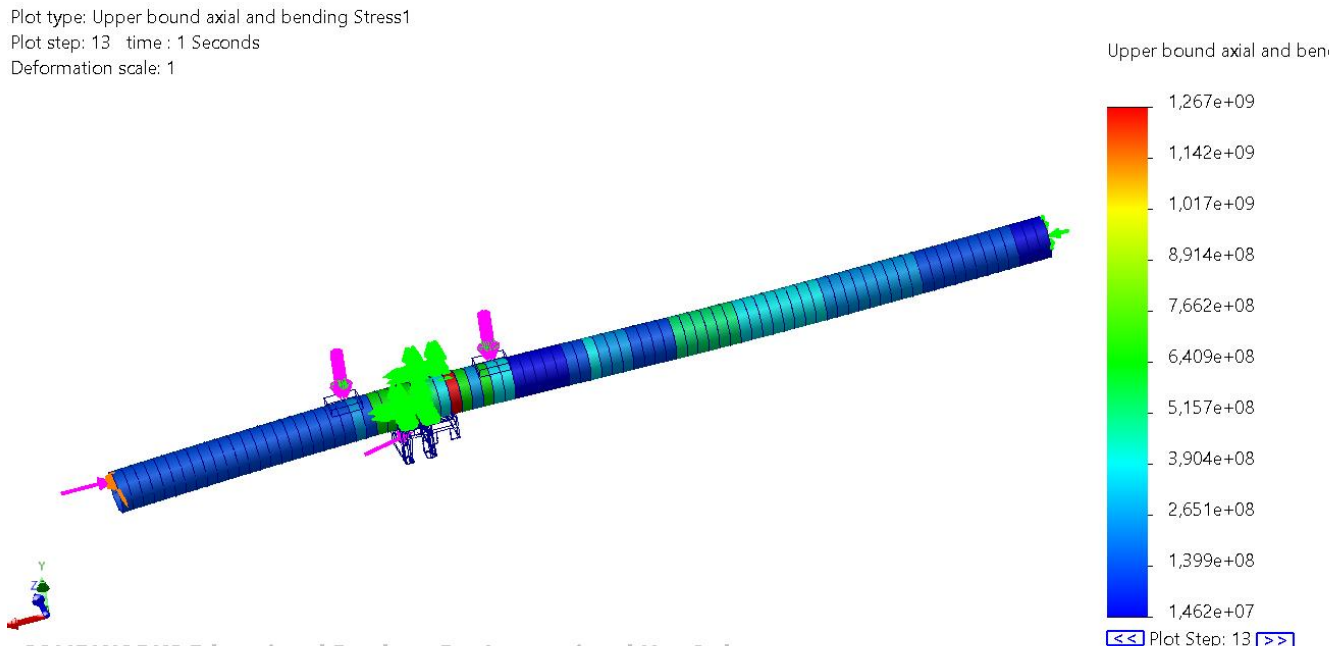


Figure H5: FEM Analysis Pipe - Combined Stress

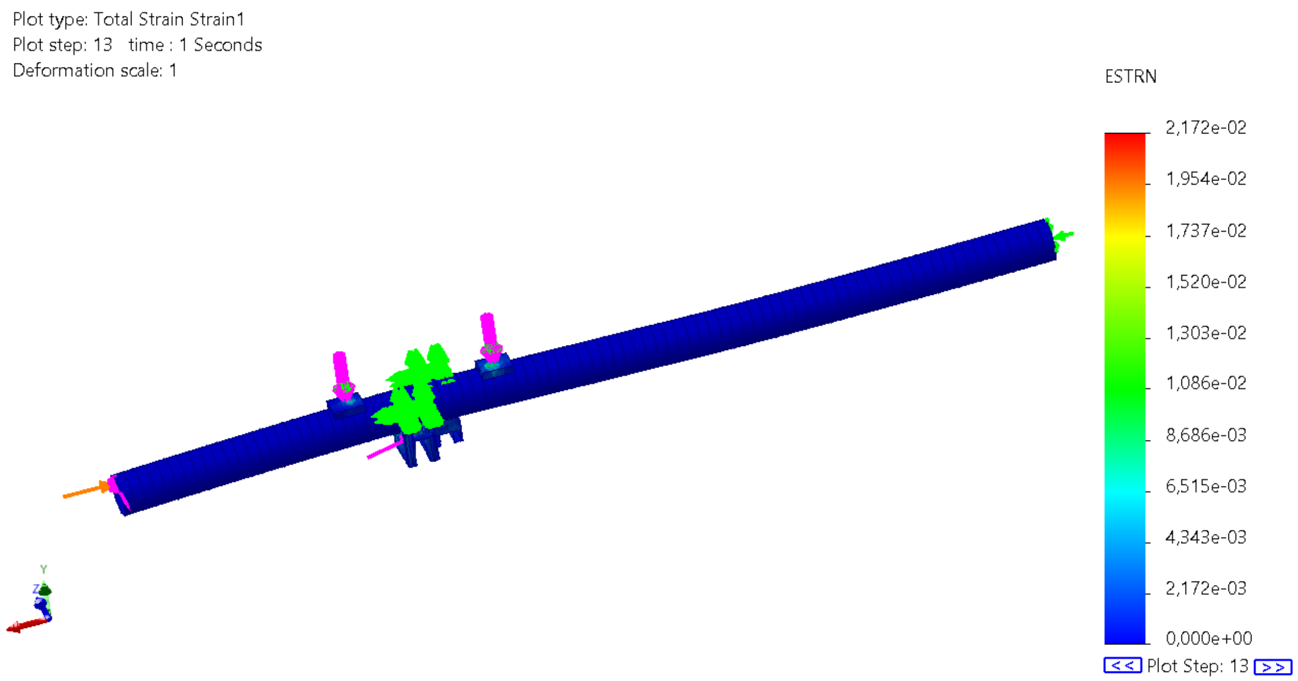


Figure H6: FEM Analysis Pipe - Equivalent Strain

The setup of the model with the applied forces, prescribed displacement, and constrains has been discussed with a senior structural engineer. The resulting deformation of the pipeline seems reasonable to him as well although modelling only a pipe section compared to the total suspended pipe lead to slight differences according to his experience.

APPENDIX I

Summary of Compliance of System Concept with Functional Requirements

Vessel requirements	**1.1**	The bending machine shall be deployed from the pipelay vessel or support vessel with the available cranes and deck space.	AUXROV (section 8.1)
	1.2	The limiting capacities of the pipelay vessel are the ones from *Lorelay* as she is the smallest pipelay vessel which can install pipelines of the size 2" to 34". She can operate in water depth ranging from 18m to about 1600m. [52]	N.A. because of AUXROV (section 8.1)
	1.3	If the machine is deployed from onboard the pipelay vessel in the way that it is clamped to the pipeline in the firing line the available space for the machine is 573mm in height and 1100mm in width. If the weight exceeds 12.5ton then the length is required to have a minimum value of 6300mm. The maximum limiting weight is 20ton. If the machine is attached to the pipeline after the last station in the firing line the width is limited to 1700mm.	N.A. because of AUXROV (section 8.1)
	1.4	If the machine is deployed by crane the limiting weight in air is 300t which is the capacity of *Lorelays* special purpose crane SPC at a radius of 14m. If it is deployed with the AHC winch the limiting weight is 150t. [52]	Detailed design (section 8.3.1 and 8.3.2)
	1.5	If the machine is deployed through a moonpool in the middle of the vessel the machine size shall be sufficiently small. The dimension of the smallest moonpool from the support vessels, *Oceanic* and *Fortitude*, is 4.8m x 4.9m. The main moonpool on *Oceanic* is 7.2m x 7.2m and 8.4m x 8.4m on *Fortitude*. [53] [54]	Detailed design (section 8.3.1 and 8.3.2)
Environmental requirements	**2.1**	The machine shall be manoeuvrable at steep slopes such as shelf breaks.	Detailed design (section 8.3.1 and 8.3.2)
	2.2	The machine shall be clamped securely to the pipe such that it follows its movements during bending. Further it shall be built such that it is not damaged when deploying it through the splash zone.	Detailed design (section 8.3.1 and 8.3.2)
	2.3	The machine shall be suitable to operate in sea state of Hs 3m and Tz 7s which is the limiting sea state for pipeline installation for *Lorelay*, *Audacia*, and *Solitaire*. If the machine is deployed by crane this needs to have heave compensation (AHC) such that the machine does not damage the pipeline while approaching it.	AUXROV (section 8.1)
	2.4	The machine shall be operational in a current of 1.5knot. This is the current speed at which the maximum bottom tension for the pipeline installation is determined. These parameters depend on the vessels thruster system capacity. It can be noted that if less bottom tension is required the thrust capacity can be used to navigate in stronger currents.	AUXROV thruster capacity plus leaver of umbilical (section 8.1)
	2.5	The ROV requirement is that interfaces should be elevated minimum 1.5m above seabed to avoid interference due to seabed disturbance. Additionally, there needs to be sufficient space between the pipeline and the seabed to allow for clamping.	Placement of the tool during operation

	2.6	The mechanism which provides the force for bending as well as all other components shall be suitable for shallow as well as deep water up to 2000m. If a hydraulic system is used, the needed pressure must also be available in deep water (the hydraulic power unit HPU has to be on the machine).	AUXROV HPU unit (section 8.1)
Pipeline requirements DNG-GL-F101	**3.1**	The bending system shall to provide a bending moment of **16006.00 kNm** (pipe size 32”, wall thickness 39mm, water depth 45m, 40D minimum curvature). The calculation of this bending moment is presented below in section 6.2.	Moment calculation (section 6.2)
	3.2	The bending radius shall not exceed 40D to ensure that the maximum allowable strain of 1.25% as stated in DNG-GL-F101 is not exceeded [13] [30].	Input moment calculation (section 6.2)
	3.3	The pipe shall be bend such that it follows the topography. The achievable angle per pipe joint is described below in section 6.6.	Concept functionality (section 8.4.1)
	3.4	The pipe shall not buckle because of the weight of the machine. 1.25 ton/m for 4m long machine is acceptable according to OrcaFlex calculations. That is a total submerged weight of about 5ton.	Detailed design (section 8.3.1 and 8.3.2)
	3.5	500mm or 1.5D whichever is larger is the minimum distance of the bend to the weld.	Input angle calculation (section 8.4)
Coating requirements ISO 21809-1	**4.1**	The maximum angle per bend shall not exceed “an angle of 2.0° per pipe diameter length” as this is the flexibility requirement for polyolefin coating systems according to ISO 21809-1 [55].	Angle calculation (section 6.3)
	4.2	The pressure per area at the contact elements shall not exceed 10MPa as this is the stress at yield for PE top layers (class A) [55]. PP coating allows for 18MPa.	Detailed design (section 8.3.1 and 8.3.2)
Additional requirements	**5.1**	The bending machine shall have a communication system, or data transfer system, such that it is included in the installation process like an additional stage in the firing line.	Not met yet: part of recommendations (section 10.3)
	5.2	If the machine shall be handled by a ROV in water, the submerged weight is limited to an upper range of 100kg to 200kg.	N.A. because of AUXROV (section 8.1)
	5.3	The machine shall have sensors with which the bending radius can be determined as well as cameras to visually monitor the pipe to detect any cracks or wrinkles.	Not met yet: part of recommendations (section 10.3)

Research Questions and Where to Find the Answers Within the Report

a) What is the state of the art in pipeline bending onshore (section 3) as well as offshore (sections 4 and 4.2) and what is needed to address current limitations (section 4.1)? (Needs Analysis)
b) What are the operational requirements of the new system concept (section 5.1)? (Needs Analysis)
c) What are alternative possible concepts for an underwater cold bending system for marine pipelines (section 6) and what are their common functions and parameters (section 6.4) which are described as functional requirements (section 6.1) in the systems engineering approach? (Concept Exploration)
d) Which of these alternative concepts is the most promising one based on predefined criteria and how does the systems architecture look like, for example preliminary sizing of components of the subsea bending system (section 8.3)? (Concept Definition)
e) How does the new system concept impact normal pipeline installation (section 8.4)? (Concept Definition)