The onset of pipeline twist during reel-lay operations

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Challenge the future

The onset of pipeline twist during reel-lay operations

by

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ABSTRACT

Reel-lay operations in deep water with Heerema Marine Contractors (HMC) DCV Aegir showed the onset of axial twist during pipeline lowering and lay operations. For both phases of the pipelay, models have been made to approximate the twist observed during operations. The suspected instigators of pipeline twist that have been researched are: residual curvature in the pipeline after straightening operations, the plastic bending history of reeled pipelines, variable wall thickness along the pipeline due to fabrication and the effect of current. The main focus of this research however concerns the effect of residual curvature in the pipeline on pipeline twist.

The twist development during pipeline lowering has been modeled using analytically derived equations and by means of Finite Element (FE) analysis. Linear and non-linear analytical approximations for a vertical suspended beam with residual curvature and loads representing the end terminal and current have been compared. Given the pipeline's straightness during lowering, the linear equations were sufficient and were used for further analysis. In the analytical models, the residual curvature in the pipeline is made dependent on the twist. Via the principle of total potential energy minimization, an energetically advantageous twist angle is calculated. The results of the analytical models are compared with the FE models via a sensitivity study. Here, the amount of residual curvature, the current direction and its magnitude are varied. The results of the analytical models showed significant pipeline twist, whereas the FE models showed almost no twist. Upon inspection, it is concluded that the lack of the out-of-plane contribution for the analytical models results in inaccurate twist approximation. Therefore, the potential energy minimization method as used by Endal ([1]) is deemed unfit for the twist approximation during pipeline lowering.

Nevertheless, the correctly modeled FE models also gave twist results incomparable with the actual observed data: They gave much lower twist values than what was observed in reality, which led to further research. The effect of a spiral wise wall thickness variation along the pipeline, which is known to occur in seamless pipes, is investigated. Using a small scale FE model, axial strain during lowering of a pipeline with the aforementioned imperfection is modeled. The results showed that the imperfection gives a negligible twist contribution during lowering. Furthermore, the effect of current on the Pipeline End Terminal (PLET) is investigated. Qualitative static analyses show that a relatively low current speed on large pipeline lengths could give significant twist due to torsion: the lengthy pipelines offer less resistance to torsion. Also in house fluid body interaction research done on an Inline Structure (ILT) modeled by a flat plate subjected to an oscillating flow ([2]) shows that current can potentially give a large amount of torsion resulting in a significant twist.

For the investigation of the twist development during lay operations again an analytical approach and a FE approach are used for modeling. The results of the models are compared with one another in a sensitivity analysis. Here, the residual curvature, the pipeline diameter, buoyancy, axial tension and wall thickness are varied. The results show that residual curvature is the primary reason for twist during laying and that the direction of the residual curvature with respect to the lay direction is very important. Given Aegir's reel configuration, under-straightened pipelines can give significant twist, whereas over-straightening leads to negligible twist.

The other parameters tested seem to have little influence. From model result comparison, it is concluded that the analytical approach gives a decreasing accuracy of twist approximation when increasing the amount of residual curvature. Research showed that the out-of-plane bending which is disregarded in the analytical models, becomes more dominant with increasing residual curvature and hence gives a larger discrepancy with the FE models. Still, the twist results are in the range of observed twist data from a previous pipelay project.

Additionally, the effect of plastic deformation to obtain residual curvature in the FE models is investigated. Here, a simplified beam model is made where plastic residual curvature is induced prior to either pipeline lowering or laying. It should be noted however that the induced plastic strain in the pipeline model is relatively low compared to the strains induced during the actual reeling process of the Aegir. Also, in the model the beam is only bent plastically once instead of four times. The results were compared with the elastically pre-curved FE models and during lowering as well as laying no significant difference in twist was identified. Given the model simplifications and the indifference in results between plastically and elastically pre-curved beams, the tests were deemed inconclusive. More representative (and computer intensive) FE models will have to be made to accurately model the effect of plastic deformations that occur during reeling.

Although twist development during lowering is still unclear, the cause of twist during laying is understood and can be preventively decreased. This can be done by either reducing the amount of residual curvature in the pipeline after straightening as much as possible or by keeping the residual curvature in the over straightening domain.

PREFACE

Hereby, I proudly present my graduation thesis entitled: 'The onset of pipeline twist during reel-lay operations'. I was motivated by the eagerness to better understand the twist development during reel-lay operations occurring with Heerema Marine Contractors (HMC) DCV Aegir. It has been a challenging research project given the limited knowledge beforehand, but highly rewarding due to the better understanding of the phenomena afterwards. The thesis has been written to fulfill the graduation requirements of the master's program Offshore and Dredging Engineering at the Delft University of Technology (DUT).

Firstly, I would like to thank Harm at HMC. Your profound interest in my research, your dedication and your patience have not gone unnoticed. And not to forget, you have given me a quote that I will use as an excuse for myself from time to time: "It should work...". Secondly, I would like to thank Erwan, for your constructive feedback and for asking the sharp questions which really excelled the research to a higher level. Kees, thank you for your heart in the subject matter, for sharing your vast knowledge of the offshore industry and for being just as stubborn as me during the research: it led to a great result.

From the university I would first of all like to thank Jeroen, my daily supervisor. Thank you for always being available for help, for your dedication to my Thesis, and for your creativity in jokes regarding my Antillean background. Furthermore I would like to thank Andrei, for his academic expertise and his sharp feedback during meetings: it has been truly constructive.

Special thanks to my parents and my sister, for giving support from overseas. Especially during the challenging moments I was reminded of the importance of family, and how distance is only relative.

Lastly I would like to thank all the students at HMC, for making the time enjoyable. I will miss the coffee breaks and the daily 'where to go to lunch' debate starting way too early in the morning. For those still busy at the moment, I wish you good luck.

V.J. Taams Leiden, July 2016

COMPANY PROFILE

Heerema Marine Contractors is an internationally active marine contractor with more than 50 years of specialized experience in the offshore and gas industry. Together with the other division Heerema Fabrication Group, which focuses more on the engineering and fabrication of structures for the offshore industry, they form the Heerema Group. The Heerema Group has an international workforce of approximately 2000 persons of which the majority is employed at HMC.

Ever since the start of operations in Venezuela in 1948, HMC has been committed to solving complex offshore related challenges. HMC's scope of work is the transportation, installation and removal of offshore facilities. These facilities comprise of fixed structures, floating structures, subsea pipelines and subsea infrastructures in shallow, deep or ultradeep waters. To achieve this, the fleet of Heerema's vessels are able to facilitate heavy lift operations, float-overs and pipelay operations. In total HMC owns four large specialized vessels, and a series of barges and smaller support vessels. The deep water construction vessel Balder (Figure 3) is a combination of a semi-submersible crane vessel and a pipelay vessel, capable of J-lay and multiple subsea installation operations. HMC's two semi-submersible crane vessels the Hermod (Figure 2) and the Thialf (Figure 1) are capable of heavy lift operations, where the Thialf is at present the largest crane vessel in the world. The latest specialized vessel is the Aegir (Figure 4), a deep-water construction vessel which is capable of executing complex subsea and pipeline projects in ultra-deep water and with a sufficient lifting capacity for fixed platform installation in relatively shallow water. The reel/J-lay process of this vessel will be the focal point of further research in this thesis, for it is the onset of pipe rotation during the reel-lay of the Aegir that is to be investigated.



Figure 1: The SSCV Thialf, the largest crane vessel in the world, executing a tandem heavy lift operation.



Figure 3: The DCV Balder with J-lay tower.



Figure 2: The SSCV Hermod in stormy weather.



Figure 4: The DCV Aegir during transit.

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GLOSSARY

NOMENCLATURE

GREEK SYMBOLS

Symbol	Property	[Unit]
α	Tower or top angle	[rad]
α_s	Non-dimensional quantity which measures the effect of the bending stiffness compared to the non-dimensional tension contribution <i>h</i>	[-]
β	Angle of attack of the current	[rad]
γ	Load factor relating the bending stiffness with the axial tension	[m]
δ_a	Axial displacement of Abaqus shell model	[<i>m</i>]
δ_r	Axial displacement in 'real life' pipeline as used during Lucius Project	[<i>m</i>]
$\Delta \epsilon$	Change in strain	[-]
ϵ	Strain	[-]
ϵ_A	Strain in Abaqus shell model	[-]
ϵ_i	Strain at a certain step number	[-]
ϵ_{nom}	Nominal strain	[%]
ϵ_p	Plastic strain	[-]
ϵ_r	Stain in real life pipeline as used during Lucius Project	[-]
ϵ_{T_b}	Strain in pipeline caused by the bottom tension	[-]
e_{W_s}	Strain in pipeline caused by submerged weight	[-] [rad]
0 A(1)	Bending angle of pipeline	[rad]
0(1) A(2)	First derivative of the bending angle with respect to distance along	[<i>rad</i>]
0(2)	the pipeline <i>s</i>	$\left[\frac{1}{s}\right]$
θ(3)	the pipeline <i>s</i>	$\left[\frac{r}{s^2}\right]$
θ_a	Bending angle of pipeline at clamping point	[rad]
θ_b	Bending angle of pipeline at end point	[rad]
θ_s	Rotation of reel's center point during spooling on	[rad]
θ_u	Rotation of reel's center point during un spooling	[rad]
ao	Change in the bending angle	$\begin{bmatrix} r a a \end{bmatrix}$
ĸ	Cross-sectional average of the curvature in the pipeline	$\begin{bmatrix} 1 \\ m \end{bmatrix}$
K _C	catenary theory	$\lfloor \frac{1}{m} \rfloor$
κ _{esc}	Curvature development along the pipeline as described by with en- hanced stiffened catenary theory	$\left[\frac{1}{m}\right]$
κ_i	Curvature at a certain step number	$\left[\frac{1}{m}\right]$
κ _{nl}	Curvature development along the pipeline as given by non-linear equations	$\left[\frac{1}{m}\right]$
K _{SC}	Curvature development along the pipeline as described by stiffened catenary theory	$\left[\frac{1}{m}\right]$
κ _r	Residual curvature in the pipeline after straightening operations	$\left[\frac{1}{m}\right]$
λ	Load factor relating the bending stiffness with the horizontal tension	[m]
П	Total potential energy of the system	[/]
ρ	Density	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	Density of coating	$\left[\frac{kg}{m^3}\right]$
ρ_{ex}	Density external medium pipeline	$\left[\frac{\kappa g}{m^3}\right]$
$ ho_i$	Density internal medium pipeline	$\left[\frac{\kappa g}{m^3}\right]$
$ ho_{st}$	Density of X-65 steel	$\left[\frac{kg}{m^3}\right]$
ρ_w	Density of seawater	$\left[\frac{kg}{m^3}\right]$
$\sigma_{1,2}$	Principle stresses	[Pa]
σ	Stress	[Pa]
σ_i	Stress at a certain step number	[Pa]

Σ	Summation	[-]
$\Delta \sigma$	Change in stress	[Pa]
σ_y	Yield strength	[Pa]
ω_1	First frequency of excitation force	$\left[\frac{rad}{s}\right]$
ω_2	Second frequency of excitation force	$\left[\frac{rad}{s}\right]$
ϕ_0	Pipeline twist, torsion or roll angle at the seabed	[rad]
ϕ	Pipeline twist, torsion or roll angle	[rad]
ϕ_A	Twist in Abaqus shell model	[rad]
ϕ_L	Pipeline twist during lowering operations	[rad]
ϕ_{max}	Maximum occurring pipeline twist	[rad]
$\phi_{max,e}$	Maximum occurring pipeline twist for elastic model	[rad]
$\phi_{max,p}$	Maximum occurring pipeline twist for plastic model	[rad]
ϕ_{PF-01}	Pipeline twist of the PF-01 pipeline that is analyzed	[rad]
ϕ_r	Twist in 'real life' pipeline as used during Lucius Project	[-]
ϕ_S	Pipeline twist of suspended pipeline during lay operations	[rad]

LATIN SYMBOLS

Symbol	Property	[Unit]
\angle_{sp}	Angle of the spiral wise wall thickness variation	[°]
a	Constant equal to $\frac{T_H}{W_{e}}$	[m]
a_i	Strain at the axis of symmetry for a given step number i	[%]
A	Cross-sectional area pipeline	$[m^2]$
A_{ex}	External cross-sectional area pipeline	$[m^2]$
$A_{H,PLET}$	Surface are PLET upon horizontal loading	$[m^2]$
A_i	Internal cross-sectional area pipeline	$[m^2]$
A_r	Pipeline cross-sectional area of 'real life' pipeline as used during Lu-	$[m^2]$
	cius Project	
A_{st}	cross-sectional area of steel pipeline	$[m^2]$
b_x	Displacement of bottom point in x-direction	[<i>m</i>]
b_y	Displacement of bottom point in y-direction	[<i>m</i>]
dA	Segment of cross-sectional area pipeline	$[m^2]$
C_d	Drag coefficient	[-]
$C_{d,PLET}$	Drag coefficient for PLET	[-]
C_m	Inertia coefficient	[-]
C_D	Drag coefficient for flat plate	[-]
C_L	Lift coefficient for flat plate	[-]
C_N	Normal coefficient	[-]
COG	Center of Gravity	[<i>m</i>]
COP	Center of Pressure	[<i>m</i>]
d_i	Location of the neutral axis d for a given step number i	[<i>m</i>]
ds	Beam segment	[<i>m</i>]
D	Diameter of the pipeline	[<i>m</i>]
D_i	Inner diameter of the pipeline	[<i>m</i>]
D_o	Outer diameter of the pipeline	[<i>m</i>]
Ε	Young's Modulus	[Pa]
EP	Potential energy	[<i>J</i>]
E_r	Young's Modulus of 'real life' pipeline as used during Lucius Project	[Pa]
E_{st}	Young's Modulus X-65 steel	[Pa]
f	Exiting force	[N]
f_D	Drag force as described in Morison's equation	[N]
f_I	Inertia force as described in Morison's equation	[N]
F_b	Buoyancy force per m of length	$\left[\frac{N}{m}\right]$
F_c	Current load on pipeline	[N]
F_e	Current load and weight of end terminal	[N]
$F_{H,PLET}$	Horizontal force acting on PLET	[N]
F_i	Instigating force	[N]
F_N	Normal force	[N]
F_x	Forces in the x-direction	[N]
F_z	Forces in the z-direction	[N]
g	Gravitational acceleration	$\left[\frac{m}{s^2}\right]$
g eq	Equivalent gravitational acceleration	$\left[\frac{\tilde{m}}{s^2}\right]$

g_s	Small gravitational acceleration	$\left[\frac{m}{s^2}\right]$
G	Shear modulus	[Pa]
G_A	Shear modulus of material shell Abaqus model	[Pa]
h	Non dimensional tension contribution	[-]
H	Out-of-straightness	[<i>m</i>]
H_{FLET}	Height of the FLET	[<i>m</i>]
Ι	Second moment of Area	$[m^{4}]$
I _{st}	Second moment of Area of steel pipeline	$[m^{4}]$
J	Torsional constant	$[m^{4}]$
J_A	Torsional constant of material in Abaqus shell model	$[m^{4}]$
J _r	Torsional constant of 'real life' pipeline as used during Lucius Project	$[m^{4}]$
L	Length of the pipeline	[m]
Lapprox	Approximate pipeline length	[m]
L_A	Length of Abaqus shell model	[m]
L _c	Bundled residual curvature length	[m]
L_{FLET}	Length of the FLET	[m]
L_{κ_r}	Length of pipeline with residual curvature	[m]
L_{PF-01}	Length of PF-01 pipeline that is analyzed	[<i>m</i>]
L_r	Length of 'real life' pipeline as used during Lucius Project	[m]
m_a	added mass of the pipeline per m of length	$\left[\frac{\kappa g}{m}\right]$
m_c	mass of coating pipeline per m of length	$\left[\frac{kg}{m}\right]$
m_{ct}	mass of pipeline content per m of length	$\left[\frac{kg}{m}\right]$
m_n	mass of pipeline per m of length	$\left[\frac{kg}{kg}\right]$
m	mass of steel nineline per m of length	$\begin{bmatrix} m \\ kg \end{bmatrix}$
mst	Tatal mass of steel pipeline per in of length	$\begin{bmatrix} m \\ m \end{bmatrix}$
m_T	Noment	$\left[\frac{m}{m}\right]$
IVI M	Mollielli Overturning moment due to effect COC of DLET from ningling	[Nm]
M	Diastic moment	[Nm]
M _p M _m	Total model	[1\11]
M	Vaw moment	[⁻]
Myaw M_:	Bending moment around the z-axis for step number i	[Nm]
$N_{z,i}$	Axial force in z-direction for step number i	[N]
p	Pressure	[MPa]
P P	Axial tension	[N]
P_h	Bottom point of pipeline	[-]
$P_{h}^{\nu}*$	New bottom point of pipeline	[-]
P_{cl}	Clamping point on the pipeline	[-]
P_{e}	End terminal COG point	[-]
P_s	Surface point of pipeline	[-]
r _{COP}	Moment arm from COP to COG	[m]
R_{κ}	Radius of curvature	$\left[\frac{1}{m}\right]$
R_r	Radius of residual curvature	[m]
R_{sp}	Radius of the rigid spool in Abaqus model	[m]
s	Distance along the pipeline	[m]
S _{el}	Element size	[m]
t	time	[<i>s</i>]
t_c	Thickness coating	[m]
t _{st}	Wall thickness steel	[m]
Т	Axial tension in the pipeline	[N]
T_0	Top tension in the pipeline	[N]
T_A	Torsion in shell model Abaqus	[Nm]
T_b	Bottom tension in the pipeline	[N]
T _e	Effective tension in the pipeline	[N]
T_H	Horizontal tension in the pipeline	[N]
$T_{max,r}$	Maximum torsion as seen in Rodermans research	[Nm]
T_P	Torsion in the pipeline	[Nm]
T_r	Torsion in 'real life' pipeline as used during Lucius Project	[Nm]
T_V	Vertical tension in the pipeline	[N]
U II	velocity exciting force Kodermans' research	$\left[\frac{m}{s}\right]$
U_a	Axiai strain energy	[<i>J</i>]
O_{bip}	m-plane bending strain energy	[J]

U_{bop}	Out-of-plane bending strain energy	[]]
U_B	Bending strain energy	[<i>J</i>]
U_{t1}	Second order strain energy regarding tension and twist	[J]
U_{t2}	Second order strain energy regarding tension and large bending an-	[J]
	gles	
U_T	Total potential strain energy	[<i>J</i>]
U_R	Torsional strain energy	[<i>J</i>]
ν	Velocity	$\left[\frac{m}{s}\right]$
vc	Current velocity	$\left[\frac{\ddot{m}}{s}\right]$
v_H	Horizontal current velocity	$\left[\frac{\ddot{m}}{s}\right]$
V	Shear force	[N]
wd	Water depth	[m]
W	Half of the arcs' horizontal length	[m]
W_s	submerged weight of the pipeline	$\left[\frac{N}{m}\right]$
$W_{s,FLET}$	submerged weight of the FLET	[N]
W_{PLET}	Submerged weight of PLET	[mT]
Wwings,down	Width of FLET with wings down	[m]
Wwings,up	Width of FLET with wings up	[m]
<i>x</i> (1)	Horizontal displacement of vertical suspended beam	[m]
<i>x</i> (2)	Linearized bending angle along the vertical suspended beam	[rad]
<i>x</i> (3)	Curvature along the vertical suspended beam	$\left[\frac{rad}{s}\right]$
<i>x</i> (4)	Third derivative of the horizontal displacement of the vertical sus-	$\left[\frac{rad}{s^2}\right]$
	pended beam along the beam	3
x_i	x-coordinate of incremental length <i>i</i>	[m]
<i>Yi</i>	y-coordinate of incremental length <i>i</i>	[<i>m</i>]
z_i	Distance from symmetry axis to location on cross-section pipe for	[<i>m</i>]
	step number i	

ACRONYMS

ACRONYM	S
3mE	Mechanical, Maritime and Materials Engineering
AR	Abandonment and Recovery
AUT	Automatic Ultrasonic Testing
AWL	Auxiliary Welding Level
COG	Center of Gravity
СОР	Center of Pressure
DCV	Deep water Construction Vessel
DNV	Det Norske Veritas
E.o.M	Equation of Motion
EXGR	Gas Export Riser
EXOR	Gas Export Riser
FBE	Fusion-bonded Epoxy
FE	Finite Element
FEA	Finite Element Analysis
FLET	Flowline End Termination
GSPU	Glass Syntactic Polyurethane
НМС	Heerema Marine Contractors
ILT	Inline Structure
mT	Metric Tonnes
PF	Production Flowline

PI	Production Infield Flowline
PLEM	Pipeline End Manifold
PLET	Pipeline End Terminal
PR	Production Riser
RAO	Response Amplitude Operator
RB3D2	2-node-3D rigid beam element
ROV	Remotely Operated Vehicle
S4R	Standard Abaqus quad node shell elements with reduced integration
SPAR	Single Point Anchor Reservoir
SSCV	Semi Submersible Crane Vessel
DUT	Delft University of Technology

1

INTRODUCTION TO PIPELINE TWIST DURING REEL-LAY OPERATIONS

At the end of 2013, HMC's DCV vessel Aegir commenced with its first series of reel-lay operations[3]. The reel-lay operations entailed the installation of flowlines, export- and infield steel catenary risers for the Lucius SPAR of the Anadarko Petroleum corporation. The operations also entailed the lowering of the pipelines with end modules, such as pipeline end terminals (PLETs) or pipeline end manifolds (PLEMs). The reel-lay activities took place 390 km offshore at a water depth of approx. 2100 m. The subsea infrastructure that was laid by HMC, with the exception of the export risers, is visible in Figure 1.1. In the course of the lay operations it became apparent that the pipes were subject to rotation around their own axis. During lowering of the first part of a pipe section, the PLET had to be adjusted upon arrival at the target box on the sea floor. Using the remotely operated vehicles, changing the heading of the ship or by adding tension during the lay startup, the end module was rotated to the initial and anticipated angle at the sea bottom before the touchdown. This lead to the build-up of torque, which was relieved by freely rotating the pipeline at its end. In the final stage of the lay, a pipe head would be fastened to the end of the pipe. This head contained a rotational swivel attached to the Abandonment and Recovery (A & R) system, which upon easing the grip of the tensioner on the pipe, relieved the torque by free rotation. Depending on the length and the type of pipeline, different amounts of rotation were measured, with a maximum of 200 degrees[4]. The twisting of the pipe creates a threat to the levelness of the inline- and pipeline end structures, for the structures may land in an angle or topple over entirely, possibly damaging the integrity of the module and decreasing its functionality. Furthermore, for tie-in operations, any unexpected large angles will result in increased duration of subsea operations and will require modifications to be made to tie-in equipment. Structural damage to the pipeline or its modules, or lengthy mitigation procedures can dramatically increase the offshore lay expenditures and the project time. It is therefore of vital importance for future deep water reel-lay operations to understand the pipeline twist phenomena better.



Figure 1.1: Subsea infrastructure of laid pipelines at the Lucius SPAR project. Note: EXOR-01 and EXGR-01 were not laid by HMC [3].

The twisting of pipelines has been known to occur during S-lay, J-lay and Reel-lay operations (Appendix H). Therefore HMC had accounted for torsional mitigation procedures to be implemented during the Lucius Project. In an effort to isolate the instigation of pipeline twist, several tests were done and the observed twist during the project was carefully documented. The results of the tests are discussed in Appendix A. The Lucius project along with its tests during the whole process has given some insight as to what could contribute to the onset of twist, and more importantly, what isn't of influence. In Appendix B, the complete Aegir reel lay process is discussed, giving us insight in the stresses and strains that the pipeline endures. Lastly, an extensive literature study has been done, which provides information concerning previous research on offshore pipeline twist related topics. The literary review can be read in Appendix C. Based on all this information, together with the eagerness of Heerema Marine Contractors to better understand the twist phenomena that occurred during the Lucius Project, I have formulated my thesis scope of work.

1.1. RESEARCH QUESTION

The build-up of torque during the reel-lay process of long pipe sections in deep water leads to the involuntary rotation of the pipe. The pipeline twist can lead to increased difficulty of tie-in operations and can cause inline- or pipe end structures to land in an angle or even topple over. As of yet no definitive cause has been found for the onset op pipe rotation during reel- lay operations. One has not been able to estimate the amount of pipeline twist, resulting in time-consuming and therefore costly mitigation measures or project uncertainties. The research question is therefore formulated as follows:

What causes twist in reeled pipelines during lowering and lay operations?

1.2. RESEARCH AIM

The aim of the research is to find the most likely physical source or combination of sources which leads to torque buildup and the subsequent pipe twisting during Reel/J-lay operations of HMC's vessel Aegir. The strive is to be able to accurately predict the pipe twisting using analytically derived equations and/or numerical modeling by means of Finite Element (FE) methods.

1.3. NARROWING DOWN THE RESEARCH SCOPE

In order to isolate possible causes for pipe twist efficiently, it is necessary to look at the different stages of the Reel-lay process independently for issues that could contribute to the undesired phenomena. Together with the results of the tests that have already been done concerning the topic, it will be possible to formulate a plan of attack to get a better insight into the twist phenomena within the time frame of the Thesis Research. A schematic overview has been made to identify the key areas that may be subject to further investigation and to help formulate the thesis scope of work. The overview can be seen in Figure 1.2. The schematic overview implies that in the different stages of the reel-lay process there are multiple factors that can possibly contribute to pipeline twist, either dependent or independently of one another. Some of the factors have already been subject to research, such as the material and geometric properties of fabricated seamless pipes or the spooling on and -off of pipelines. This helps with the narrowing down of the possible source of the twist, and how the research time can be most efficiently utilized.



Figure 1.2: Schematic overview identifying key research topics

1.3.1. SEAMLESS PIPE FABRICATION

As mentioned in the appendix concerning pipeline fabrication (Appendix D), seamless pipes are known to have relatively high variations in wall thickness, out-of-straightness and out-of-roundness in comparison. However, it is statistically likely that the individual effects of for example the out-of-straightness of a pipe, are evened out due to the large amount of pipes that are welded together to form the pipe stalks. The one property that sticks out is the spiral wise distribution of the wall thickness, and possible a difference in yield strength along this distribution. Connecting two pipelines made in the same factory during the same continuous process will always lead to the longitudinal alignment of the spiral distribution, independent on which ends are connected with each other. It is therefore the spiral wise distribution of the wall thickness that is most appealing for possible research for the fabrication stage.

1.3.2. ONSHORE WELDING

It is likely that individual faults, such as in this case improper pipefitting, will be evened out over the whole stalk of the pipeline during welding procedures. Eye witnesses have stated that the mere lifting or adjusting of the stalks was enough to introduce twist to the pipelines. This arbitrary twist does not explain the increasing values of twist found during lowering and laying of the pipeline and thus it is unlikely that onshore welding and transportation is the root cause of the phenomena. Therefore, this stage is considered of less importance for further research.

1.3.3. Spooling on and -OFF

The effect that the double bend cycle has on the pipeline is of interest during the study of the twist phenomena. In the cycle, the pipeline is plastically deformed four times, before leaving the vessel via the moonpool. This leads to an asymmetric stress distribution along the cross-sectional area, which might contribute to twist. Furthermore, residual curvature after the pipeline leaves the straightener is of possible concern. For this thesis research, ovality is not taken into account and is assumed zero throughout the whole research.

1.3.4. REEL/J-LAY

There are multiple factors in the final stage of reel-lay that are of interest for further research. The effect of residual curvature, axial tension, current on the pipeline and end terminals are some of the parameters that are most promising concerning the research of pipeline twist. Additionally, this part of the pipelay process is highly suitable for further research via modeling.

1.4. OVERVIEW OF THESIS RESEARCH

A compact schematic overview of my Thesis research can be seen in Figure 1.3. After internal research and a thorough literary review, it was decided that four possible instigators of pipeline twist were subject to further investigation. Here residual curvature is expected to be the main culprit. During the pipeline operations, one must separate two distinct stages of the operation: The vertical lowering of the pipeline in a free suspended state, and the pipeline during lay operations where it is constrained at the seabed and the pipeline has a catenary-like shape. These stages have been modeled separately in order to investigate their individual contribution to the final amount of pipeline twist in the system. For both stages an array of models have been made, which can roughly be divided into the following: a set of 4DOF/element models with analytical derivations for finding solutions and a set of 6DOF/element models that use finite element analysis. These models have been compared with each other and provide an expected approximation of the twist development. The obtained twist values are validated to the twist data from the Lucius Project, and have given a better understanding of the root causes of pipeline twist and equally important, which suspected instigators are not of influence. The findings will be discussed throughout the rest of the report.



Figure 1.3: Schematic overview of plan of attack

 ϵ_P = Plastic deformations

 κ_r = Residual curvature

 t_{sp} = Spiral-wise wall thickness distribution along pipeline

 M_L = Modeling of pipeline twist during vertical lowering $M_{L,A}$ = Vertical lowering models using FEA in Abaqus $M_{L,M}$ = Vertical lowering models using Matlab

 M_S = Modeling of pipeline twist of suspended pipeline in lay configuration $M_{S,A}$ = Suspended pipeline during laying models using FEA in Abaqus $M_{S,M}$ = Suspended pipeline during laying models using Matlab

 ϕ_L = Pipeline twist during vertical lowering operations ϕ_S = Pipeline twist of suspended pipeline during lay operations

2

ANALYSIS TWIST DEVELOPMENT OF SUSPENDED PIPELINE DURING LAYING

During pipeline lay operations the pipeline leaves the moonpool vertically and is gradually bent to a horizontal configuration at the seabed, see Figure 2.1. The shape of a pipeline can be approximated via several methods, both analytical, numerical or a combination of the two. The simplest method is the natural catenary approximation, where the bending stiffness of the pipeline is not taken into account[5][6][7]. Lacking sufficient advanced mathematical theory in the past, the pipelines were approximated using this theory. Although the bending stiffness was neglected, it was believed that by making the upper end conditions of the pipeline equal to those required for natural catenary calculations, a conservative approximation is met for the pipeline during laying. One aspect of the natural catenary which makes it very appealing in the approximation of the pipeline during laying is that the horizontal components of the force at the lay barge T_H is independent of the water depth, as given in the catenary equation [5]. This signifies that the capacity required by the barge in terms of propulsion or mooring will not increase with increasing water depth. At increasing water depth, the tension T will increase, but the remaining horizontal force T_H will remain constant. According to Dixon et al. [5] the pipe stiffness is of significant importance in the analysis of short and stiff pipeline sections, whereas long sections of unsupported pipeline are analogous to a string or cable where the shear is negligible and the curvature exists without any noticeable moment. When looking at the shape of the pipeline when using the stiffened catenary equations, it is seen that the pipeline approximates the shape of a natural catenary over most of its length. Near either end the shape diverts from the natural one due to the effect of the bending stiffness of the pipeline and the boundary conditions which are not compatible with those of the natural catenary. If the pipeline is assumed to take the shape of a natural catenary, the maximum curvature will be reached at the point of contact with the seabed (with the assumption that the seabed is horizontal). It should be noted that the natural catenary method as well as the stiffened catenary method both neglect the dynamic stresses in the pipeline due to motions of the lay barge and due to current and wave forces acting on the pipeline.



Figure 2.1: Suspended pipeline during lay operations with the DCV Aegir

The goal is to approximate the shape of the pipeline during lay operations in shallow water (comparable to the Ichthys Project) and in deep water (comparable to the Lucius Project). Once the correct parameters have been obtained, the development of pipeline twist can be researched. This is done by total potential energy minimization and by finite element analysis. The goal here is to obtain results which are comparable to the pipeline twist data as measured during the Lucius Project (see Figure 2.2). Please note that the analysis in this chapter only concerns the pipelay operations and not the lowering of the pipeline. Ergo, an analysis is done to approximate the twist data right from the red dotted line in Figure 2.2.



Figure 2.2: Twist data of pipeline installation operations with the DCV Aegir. Left from dotted line: twist data during pipeline lowering (not to be considered in this chapter), Right from dotted line: twist data during laying.

2.1. APPROXIMATION PIPELINE GEOMETRY USING NATURAL CATENARY

A simple sketch is given of the geometry of a pipeline during lay operations in Figure 2.3. When solving the system, one obtains the classical natural catenary equation as given in Equation 2.1. Another useful equation is the one given in Equation 2.2, which upon differentiation gives us the curvature along the pipeline (Equation 2.3)[4].



Figure 2.3: Geometry of the pipeline during laying

$$z(x) = \frac{T_H}{W_s} \cdot \left[\cosh\left(\frac{x \cdot W_s}{T_H}\right) - 1 \right]$$
(2.1)

$$\theta(s) = \arctan\left(\frac{sW_s}{T_H}\right) \tag{2.2}$$

$$\frac{d\theta}{ds} = \kappa_n(s) = \frac{a}{a^2 + s^2}$$
(2.3)

 $T_{H} = \text{Horizontal tension in pipeline } [N]$ $W_{s} = \text{Submerged weight per m} \left[\frac{N}{m}\right]$ $\kappa_{n} = \text{Curvature along the pipeline } \left[\frac{1}{m}\right]$ $a = \text{constant equal to } \frac{T_{H}}{W_{s}} [m]$ As can be derived from the curvature equation $k_n(s)$, the maximum curvature is achieved at the seabed. This is in conflict with the boundary condition that the curvature at the seabed is equal to zero, making the natural catenary equation a conservative approach. Also, as mentioned before, the bending stiffness of the body is not taken into account, making it unsuitable for shallow pipelay and stiff pipelines and more appropriate for deep water operations.

2.2. APPROXIMATION PIPELINE GEOMETRY USING STIFFENED CATENARY

To improve the accuracy of the pipeline end points, an analytical approximation of the pipeline configuration was suggested by Plunkett [8], who introduced asymptotic expansion to the natural catenary equation. In this stiffened catenary equation, the bending stiffness is taken into account[8][5]. Here, a non-dimensional quantity which will be expressed as α_s will measure the effect of the bending stiffness, in comparison to the non-dimensional tension contribution *h*. The ratio $\frac{\alpha_s^2}{h^3}$ gives us the relative influence of the bending stiffness on deflection in comparison to the influence of axial tension. The value of this ratio must be between 0 and 1, thus h^3 is equal or greater than α_s^2 .

$$\alpha_s^2 = \frac{EI}{W_s L^3}$$
$$h = \frac{T_H}{W_s L}$$
$$0 \le \frac{\alpha_s^2}{h^3} \le 1$$

The boundary conditions at the lay-barge are considerably more complicated than those for the natural catenary. The equations for the minimum tension and top angle (the angle the tower makes with respect to the horizontal) are given as follows:

$$T = W_s L \sqrt{h^2 + 1} \tag{2.4}$$

$$\theta = \frac{\pi}{2} - \arctan(h) - \frac{\alpha_s h}{(h^2 + 1)^{\frac{1}{4}}}$$
(2.5)

The water depth corresponding to the values of α , *h* and *L* is given in Equation 2.6

$$wd = L\left[\sqrt{h^2 + 1} - \sqrt{h^2 + \frac{\alpha_s^2}{h}} + \alpha_s^2 \left[\frac{1}{\sqrt{h}\left(h^2 + \frac{\alpha_s^2}{h}\right)^{\frac{3}{4}}} - \frac{h^2}{\left(h^2 + 1\right)^2}\right]\right]$$
(2.6)

In order to solve the catenary equations, one must simultaneously solve a set of equations. This is done using a numerical, iterative process in Mathcad [9], where Equation 2.6 and Equation 2.5 are set to zero and for the range of α_s and hspecified previously. The values for α_s and h are then found that best approximate the solution of the equations given in 2.7 for the ranges as mentioned in 2.8.

$$\begin{cases} 0 = \theta - \frac{\pi}{2} + \arctan(h) + \frac{\alpha_s h}{(h^2 + 1)^{\frac{1}{4}}} \\ 0 = \sqrt{h^2 + 1} - \sqrt{h^2 + \frac{\alpha_s^2}{h}} + \alpha_s^2 \left[\frac{1}{\sqrt{h} \left(h^2 + \frac{\alpha_s^2}{h}\right)^{\frac{3}{4}}} - \frac{h^2}{(h^2 + 1)^2} \right] - \frac{wd}{\left(\frac{EI}{W_s \alpha_s^2}\right)^{\frac{1}{3}}} \end{cases}$$
(2.7)

For

$$0 \leq \alpha_s^2 \leq 1$$

$$h^3 > \alpha_s^2$$
(2.8)

The curvature $\kappa_n(s)$ of the stiffened catenary is given by the following in Equation 2.9, in which one can see that asymptotic expansion of the natural catenary equation has taken place. The expansion is dependent on the load factor γ , which is given in Equation 2.10.

$$\kappa_n(s) = \frac{a}{a^2 + s^2} - \frac{a}{(a + wd)^2} e^{\left(\frac{s}{\gamma} - \frac{L}{\gamma}\right)}$$
(2.9)

$$\gamma = \sqrt{\frac{EI}{T}} \tag{2.10}$$

As mentioned before, curvature based on natural catenary theory gets its maximum at the seabed. This is not correct however, for the curvature at the seabed and at the surface should be equal to zero if one assumes a simply supported pipe at the surface. In the stiffened catenary curvature Equation 2.9, the extra term allows the boundary conditions at the surface to be met. For the seabed boundary conditions, another term has to be added, which was done by Geir Endal in Endal et al. [10].

2.3. APPROXIMATION PIPELINE GEOMETRY USING ENHANCED STIFFENED CATENARY

The next step is to implement improvements to the curvature equation, so that also the approximation of the boundary conditions at the seabed is accurate. Additional terms are added to the stiffened catenary equation, making it more compliant. This is done by Endal et al. [10], and gives the following estimation of the curvature for the suspended pipeline during laying.

$$\kappa_n(s) = \frac{a}{a^2 + s^2} - \frac{a}{(a + wd)^2} e^{\left(\frac{s}{\gamma} - \frac{L}{\gamma}\right)} - \frac{a}{(a + s)^2} \frac{e^{\left(\frac{L}{\lambda} - \frac{s}{\lambda}\right)}}{e^{\left(\frac{L}{\lambda}\right)}}$$
(2.11)

$$\lambda = \sqrt{\frac{EI}{T_H}}$$
(2.12)

Where λ is another load factor, given by the ratio between the flexural rigidity and the horizontal tension. For the sake of further reference, Equation 2.11 is named the enhanced stiffened catenary equation.

2.4. APPROXIMATION PIPELINE GEOMETRY USING NON-LINEAR BEAM THEORY

In this section a non-linear mathematical model will be used to obtain the differential equations for a suspended pipeline during lay operations[11]. Using this method, it will be possible to compute the internal moment, shear and normal forces. The pipeline is assumed to be uniform along its arc length *s* and has isotropic material properties.

Beam segment under vertical loading

A beam segment *ds* is taken to obtain the force equilibriums and to deduce the differential equations that analytically describe the suspended pipeline shape during laying. See Figure 2.4.



Figure 2.4: Force analysis for a small pipeline segment

The pipeline is regarded as a tensioned beam with self weight. In the model no linearization will take place: the partial derivatives are taken along the arc length *s* of the beam segment, and not along the projected length of the arc. A subtle distinction, but an important difference with linearized models. The equilibrium in the vertical direction of this non-linear approximated beam segment is given in the following:

 $\Sigma F_z = 0$

Summing the force in the vertical direction gives,

$$(T+dT)\cdot \sin(\theta+d\theta) - T\sin\theta - (V+dV)\cdot \cos(\theta+d\theta) + V\cdot \cos\theta - W_{s}\cdot ds = 0$$
(2.13)

The same equilibrium of forces is sought after in the horizontal plane,

$$\Sigma F_x = 0$$

$$(T+dT)\cos(\theta+d\theta) - T\cos\theta + (V+dV) \cdot \sin(\theta+d\theta) - V\sin\theta + f(s) \cdot ds = 0$$
(2.14)

Solving these two equilibrium equations will give the governing differential equations for the system. Since a small segment of the pipeline in sagbend formation is being analyzed, small angle approximations ($d\theta \ll 1$) are used to rewrite the sine and cosine terms of the equation. The derivatives of the sine and cosine terms over a small angle change of $d\theta$ are given by:

$$\frac{\frac{\sin(\theta + d\theta) - \sin\theta}{d\theta}}{\frac{\cos\theta - \cos(\theta + d\theta)}{d\theta}} \cong \sin\theta$$

Therefore the terms $sin(\theta + d\theta)$ and $cos(\theta + d\theta)$ in Equation 2.13 and Equation 2.14 are approximated by:

$$sin(\theta + d\theta) = cos\theta \cdot d\theta + sin\theta$$
$$cos(\theta + d\theta) = cos\theta - sin\theta \cdot d\theta$$

With the new found approximated terms the equilibrium equations Equation 2.13 and Equation 2.14 are used to come up with the governing differential equations for the beam segment. A series of algebraic operations gives us the following:

$$T\frac{d\theta}{ds} - \frac{dV}{ds} - W_s \cos\theta - f(s)\sin\theta = 0$$
(2.15)

Governing equations

The shear force *V* is the partial derivative of the moment *M* to *s* (see Equation 2.17). Furthermore, the moment *M* is the partial derivative of the angular displacement against *s* (Equation 2.16). By rewriting the shear component to a derivative of the angular displacement θ , the equation of motion is restructured to a 3^{*rd*} order differential equation.

$$M = EI\kappa = EI\frac{d\theta}{ds}$$
(2.16)

$$V = \frac{dM}{ds} \tag{2.17}$$

$$V = EI \frac{d^2\theta}{ds^2}$$
(2.18)

Therefore the first governing differential equation of the system (G_1) is deduced to be equal to Equation 2.19.

$$\underline{G_1:} \qquad EI\frac{d^3\theta}{ds^3} - T\frac{d\theta}{ds} + W_s \cos\theta + f(s)\sin\theta = 0 \qquad (2.19)$$

To obtain the second governing differential equation a closer look it taken on the equilibrium along the centerline of the pipe segment. Balancing the normal forces along the centerline one obtains the following equilibrium:

$$\Sigma F_N = 0$$

$$0 = (T + dT) \cdot \cos(d\theta) - T - W_s ds \cdot \sin\theta + f(s) ds \cdot \cos\theta \qquad (2.20)$$

According to the small angles approximation, $d\theta \ll 1$. Therefore one assumes that $cos(d\theta) \cong 1$. This approximation gives us the simple second governing differential equation (*G*₂) for the system after minor algebraic operations:

$$\underline{G_2:} \qquad \qquad \frac{dT}{ds} = W_s \sin\theta - f(s)\cos\theta \qquad (2.21)$$

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The tension *T* can be decomposed into a horizontal and a vertical contribution, which is customary in offshore related pipelay projects, for the horizontal tension can be controlled by the forward motion of the vessel. Therefore the governing differential Equation 2.19 is also written in terms of horizontal lay tension T_H and vertical tension *V*.

$$T_H = T\cos\theta + V\sin\theta \tag{2.22}$$

By rewriting Equation 2.22 to terms expressed in tension *T* and filling in the tension component in Equation 2.19 one obtains a formulation for the 3^{rd} order differential equation dependent on horizontal lay tension T_H :

$$T = \frac{T_H}{\cos\theta} - V \tan\theta$$

$$EI\frac{d^{3}\theta}{ds^{3}} - \left(\frac{T_{H}}{\cos\theta} - Vtan\theta\right) \cdot \frac{d\theta}{ds} + W_{s}\cos\theta + f(s)\sin\theta = 0$$

$$EI\frac{d^{3}\theta}{ds^{3}} - \left(\frac{T_{H}}{\cos\theta} - EI\frac{d^{2}\theta}{ds^{2}}tan\theta\right) \cdot \frac{d\theta}{ds} + W_{s}\cos\theta + f(s)\sin\theta = 0$$

$$EI\frac{d^{3}\theta}{ds^{3}} - \frac{T_{H}}{\cos^{2}\theta}\frac{d\theta}{ds} + EI\frac{d^{2}\theta}{ds^{2}}\frac{tan\theta}{\cos\theta} \cdot \frac{d\theta}{ds} + W_{s} + f(s)tan\theta = 0$$

$$EI\frac{d^{3}\theta}{ds^{3}} - T_{H}sec^{2}\theta\frac{d\theta}{ds} + EI\frac{d^{2}\theta}{ds^{2}}tan\thetasec\theta\frac{d\theta}{ds} + W_{s} + f(s)tan\theta = 0$$
(2.23)

Given a trigonometric identity for differentation,

$$tan\theta sec\theta = \frac{d}{d\theta}sec\theta$$

The differential equation becomes:

$$EI\frac{d^{3}\theta}{ds^{3}} + EI\frac{d}{ds}\left(sec\theta\frac{d^{2}\theta}{ds^{2}}\right) - T_{H}sec^{2}\theta\frac{d\theta}{ds} + W_{s} + f(s)tan\theta = 0$$
(2.24)

Note that Equation 2.24 is merely the first governing differential equation 2.19, but written in terms of the horizontal tension. The obtained governing differential equations 2.19 and 2.21 form the non-linear bending equations for the suspended pipeline in the sagbend configuration. These equations can be used for all offshore related pipelay operations: S-lay, J-lay, abandonment and recovery operations. The main difference between the variety of offshore pipelay operations shall be in the definition of the boundary conditions. Furthermore, they are applicable from shallow to ultradeep water and for different initial angles θ_0 . When looking at the Aegir, the initial angle is dependent on the top angle of the Reel-J-lay tower.

Boundary conditions

For the reference system, one assumes that s=0 is located at the touchdown point of the pipeline with the sea bottom, and that s = L is located at the surface where the connection of the pipeline with the vessel starts. A schematic sketch of the pipeline during laying is given in Figure 2.5. The seabed is considered to be completely rigid, and therefore the change in bending angle, which is equal to the moment M will be equal to zero. Also the angle at the touchdown point is considered to be equal to zero. When looking at the tension components of the pipeline, it is clear that at the bottom the tension T of the pipeline is equal to the horizontal tension T_H . The given conditions give the boundary conditions at s = 0:


Figure 2.5: Mechanical parameters of suspended pipeline during laying

$$\left. \frac{d\theta}{ds} \right|_{s=0} = M|_{s=0} = 0 \tag{2.25}$$

$$\theta(0) = 0 \tag{2.26}$$

$$T(0) = T_H = T_0 \cos\theta_0 \tag{2.27}$$

At the surface, the angle of departure of the pipeline is equal to the top angle of the Reel-J-lay tower. Furthermore, due to the rigidity of the tensioner, it is assumed that the change of the rotation angle, and therefore the moment, is equal to zero. These relations give the boundary conditions for the pipeline at the surface (s = L):

$$\left. \frac{d\theta}{ds} \right|_{s=L} = M|_{s=L} = 0 \tag{2.28}$$

$$\theta(L) = \theta_0 \tag{2.29}$$

Solving the system

It is challenging to get analytical solutions for the system defined by the governing equations and the mentioned boundary conditions. Therefore to solve the equations, a solution is sought after using a fourth order numerical solver. But before one attempts to solve the system, a closer look must be taken at the boundary conditions. An important fact, which is not instantly obvious, is that the boundary conditions of the model are moving: the total length of the pipeline is undetermined before numerical computation, making the differential equation difficult to solve. The parameter *L* is needed as input for a possible numerical model to obtain solutions. In order to provide input for the parameter, the method of variable substitution is used. Here the variable *s* is taken to be equal to ϵL , leading to a rewriting of the governing differential equations 2.19 and 2.21:

$$\frac{EI}{L^3}\frac{d^3\theta}{d\epsilon^3} - \frac{T}{L}\frac{d\theta}{d\epsilon} + W_s \cos\theta + f(\epsilon L)\sin\theta = 0$$
(2.30)

$$\frac{dT}{d\epsilon} - W_s L \cdot \sin\theta + f(\epsilon L) \cdot L\cos\theta = 0$$
(2.31)

The boundary conditions also change and are now given at 0 and 1:

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$$\left. \frac{d\theta}{d\epsilon} \right|_0 = 0 \tag{2.32}$$

$$\theta(0) = 0 \tag{2.33}$$

$$T(0) = T_H = T_0 \cos\theta_0 \tag{2.34}$$

$$\left. \frac{d\theta}{d\epsilon} \right|_1 = 0 \tag{2.35}$$

$$\theta(1) = \theta_0 \tag{2.36}$$

However, since the unknown parameter *L* still goes into the differential equations, the system can still not be solved. To seek a solution of a boundary value problem involving an unknown parameter (in this case *L*), there are additional arguments that have to be specified. By investigating the equilibrium of forces in the direction of the centerline of the pipeline, an additional boundary condition is added:

$$T(L) = T_0 cos(\theta_0 - \theta(L))$$
(2.37)

The next step is to give an initial guess for the parameter *L*. The boundary conditions at the starting point of the pipeline on the bottom have no dependency regarding the actual value of *L*. The newest boundary condition at the surface however is only met for a certain pre-defined tension *T* and a total pipeline length *L* which is still undetermined. Here is were the beauty of the BVP4C solver comes in. The solver will vary the parameter until a value for it is found for which the latest boundary condition, which is dependent on the total length *L*, is qualitatively satisfied. Thus, the system is solved for a predefined tension *T* and top angle θ_0 , giving as output the angle θ , $\frac{d\theta}{d\epsilon}$, $\frac{d^2\theta}{d\epsilon^2}$ and the total line length *L*.

Geometric properties and physical quantities

With the obtained output, the geometric properties of the pipeline along with physical quantities such as moment and shear can be computed. Here accuracy of the results is dependent of the incremental length *i* that is used in the system. To obtain the pipeline's configuration, the coordinates along the pipeline are calculated as follows:

$$x_i = x_{i-1} + L(\epsilon_i - \epsilon_{i-1})\cos\theta_{i-1}$$
(2.38)

$$y_i = y_{i-1} + L(\epsilon_i - \epsilon_{i-1}) \sin \theta_{i-1}$$
(2.39)

The coordinates of the pipeline can also be approximated using empirical equations as described in Appendix F. This can be very useful, for a rough idea is obtained about the pipeline length that is obtained for a certain axial tension T, bending stiffness EI, selfweight W_s and top angle θ_0 . The curvature κ_i , bending moment M_i and shear force V_i can be calculated by the following:

$$\kappa_i = \frac{1}{L} \frac{\theta_i - \theta_{i-1}}{\epsilon_i - \epsilon_{i-1}}$$
(2.40)

$$M_i = \frac{EI}{L} \frac{\theta_i - \theta_{i-1}}{\epsilon_i - \epsilon_{i-1}}$$
(2.41)

$$V_i = \frac{EI}{L} \frac{d\theta_i - d\theta_{i-1}}{\epsilon_i - \epsilon_{i-1}}$$
(2.42)

Backwards substitution of *s* into ϵL gives us the curvature as function of the pipeline length *s*. So finally one has obtained the sought after curvature equation $\kappa(s)$ of the vertical suspended pipeline during laying by using non-linear beam theory.

2.5. APPROXIMATION PIPELINE GEOMETRY USING FINITE ELEMENT ANALYSIS

For the comparison of the accuracy of the different approaches to approximate the curvature of a pipeline during lay operations, a 6DOF/element Finite Element model is made using the software program Abaqus[12]. A brief overview of the properties is given below.

Elements

The pipeline is modeled using Abaqus PIPE31 beam elements [12]. These are linear, 2-node elements that model beams with pipe cross-sections that are subject to internal stress due to internal and/or external pressure loading. Having the choice of different pipe elements, the elements with the thin-walled formulation is chosen. Here the hoop stress is assumed to be constant and the radial stress is neglected for the material constitutive calculations.

Mesh

The pipeline is meshed along its length, with a mesh density of 1 element per m.

Material properties

The Elastic Modulus *E*, and material density ρ_{st} , are given the same values as used in previous models. Material is modeled as an elastic perfectly plastic material for the yield strength of X-65 steel.

Interactions

In the Abaqus model, the pipeline will interact with the seabed. Here the seabed is modeled as an 3D analytical rigid plate. This element is completely rigid and therefore not subject to deformations during interactions. The interaction during contact with the pipeline is modeled as frictionless: the interaction has solely the purpose of bringing the

Boundary conditions

At the surface, the pipe end is constrained in all translational and rotational degrees of freedom. Before simulation, the pipe end is rotated to a fixed angle, representative to the top angle. For the pipe end at the sea bottom, the pipeline is constrained in z-direction (see Figure 2.6).

Loads

The model is subjected to a gravitational field. Here, the buoyancy of the pipeline is taken into account, leading to an equivalent gravitational constant:



$$m_c = \frac{\pi}{4} \left[(D_o + 2t_c)^2 - D_o^2 \right] \rho_c \qquad (2.44)$$

$$m_T = m_{st} + m_c \tag{2.45}$$

$$F_b = \frac{\pi}{4} (D_o + 2t_c)^2 \rho_w g \qquad (2.46)$$

$$W_s = m_T g - F_b \tag{2.47}$$

$$g_{eq} = \frac{W_s}{m_{st}} \tag{2.48}$$

Figure 2.6: Global coordinate system that is used for the modeling of the pipeline in Abaqus using FE analysis.

X

$$\begin{split} m_{st} &= \text{mass of steel pipeline per m of length} \left[\frac{kg}{m} \right] \\ m_c &= \text{mass of pipeline coating per m of length} \left[\frac{kg}{m} \right] \\ m_T &= \text{total mass pipeline per m of length} \left[\frac{kg}{m} \right] \\ F_b &= \text{buoyancy force per m of length} \left[\frac{N}{m} \right] \\ W_s &= \text{submerged weight per m of length} \left[\frac{N}{m} \right] \\ g_{eq} &= \text{equivalent gravitational constant} \left[\frac{N}{kg} \right] \end{split}$$

The pipe end is subjected to a load in x-direction (see Figure 2.6), signifying the horizontal tension in the pipeline at seabed. The magnitude of this concentrated load is derived from the enhanced stiffened catenary theory [10]: for a certain pipeline configuration, along with all the material and geometric properties, the horizontal tension is calculated and implemented in the Finite Element model.

Steps

In a series of steps, the pipeline is brought into its final configuration of a pipeline during lay operations. In the 1st step, solely gravity is applied to the pipeline. In the next step, the horizontal load at the pipe end near the seabed is applied via a ramp function. In the last step, the seabed is brought up from underneath and interacts with the pipeline to create the characteristic shape.

2.6. DISCUSSION OF CURVATURE APPROXIMATIONS DURING LAY OPERATIONS

For the pipeline during lay operations (chapter 2), several models for the approximation of its curvature have been introduced. Here, the models can roughly be divided into two groups: models based on analytical equations modeled with MATLAB[13], and models based on finite element analysis executed in Abaqus. It is expected that the Finite Element model will approximate the shape of the pipeline most accurately. Finite element analysis however can be prone to large computational times, and elaborate sensitivity studies are usually more challenging to execute. The analytical models described earlier are of a more rudimentary nature, describing the pipeline behavior with 4DOF/element models using differential equations. These models are highly versatile and can solve the system in a matter of seconds. Therefore, one is looking for a model which has its origins in analytical equations, but is still able to accurately approximate the curvature during laying. The most accurate analytical models will then be verified against the corresponding finite element models. In this section, the different approximations for the curvature during lay operations will be compared with one another via a small sensitivity analysis. The sensitivity analysis is based on the project specifications of two pipe lay projects that have been executed with the Aegir in the past. During the Lucius project in the Gulf of Mexico, pipeline was laid at a water depth of 2100m with a high top angle necessary for deep water operations. High curvatures occurred in the sagbend, and due to its submerged weight, a high axial tension was present. The second analysis represents the conditions at the Ichthys pipe lay project off the west coast of Australia. Relatively shallow water up to 275 m lead to the installation of pipeline with a relatively low top angle and with relatively low axial tension. These to pipe lay scenarios will be used to identify the best approximations of the pipe lay shape.

Ichthys project

Below in Figure 2.7 the curvature graphs of the different approximations are plotted for an environment similar to the one at Ichthys. Here, the pipeline contains a thick GSPU coating, which significantly increases the buoyancy, but due to its low flexural rigidity, it bending stiffness is neglected for the calculations. The pipeline with coating is similar to pipelines that have been laid at the Lucius project. The same pipe is used for the analysis for Ichthys conditions to limit the amount of variables between the comparison of the two projects.



Figure 2.7: Left: table with properties of pipeline during Ichthys Project. Right: curvature graphs for different approximations

The following observations can be made from Figure 2.7. As predicted, the natural catenary κ_c does not take into account the boundary conditions at the clamping point at the surface and the point of interaction with the seabed. Therefore, along with the stiffened catenary κ_{sc} its maximum is found at the seabed, which in reality is not case. Introducing an additional term to the stiffened catenary equation gives us the enhanced stiffened catenary κ_{esc} , which is complaint with the curvature boundary condition at the seabed. Lastly one sees the graph of the curvature of the non-linear equations. Looking at the defined boundary conditions Equation 2.32 till Equation 2.36 of the non-linear solver in section 2.4, one sees that for the curvature at the surface end no boundary condition is specified. The curvature therefore is nonzero at the surface end, when using the non-linear solver.

Lucius project

A similar analysis is done for the pipelay operations during the Lucius project where the waterdepth was significantly deeper. This leads to an increase in the tension of the pipeline, along with a steeper top angle. The graphs for the different curvatures are given in Figure 2.8.



Figure 2.8: Left: table with properties of pipeline during Lucius Project. Right: curvature graphs for different approximations From Figure 2.8 the following can be mentioned. Due to the high top angle, the boundary condition for the curvature at the surface point gives less difference in results for the different models. All models more or less have the same curvature progression along the pipeline until the last 50m. Here one sees slight deviations in the curvature between the models, until the seabed interaction point is approached. Once again the natural catenary κ_c and the stiffened catenary κ_{sc} obtain their maximum curvature at the seabed interaction point. The non-linear curvature model κ_{nl} and the enhanced stiffened catenary equation κ_{esc} both achieve their maximum curvature at the same level at the same distance along the pipeline, and go to zero at the seabed interaction point.

2.7. CONCLUSION APPROXIMATION COMPARISONS

From the small sensitivity analysis the following can be concluded. At shallow water depths, the curvature has a more gradual development, compared to laying at deep water depths, where a profound peak in the curvature is present close to the seabed interaction point. For the shallow Ichthys comparison study, one sees that the the natural catenary approximation is inaccurate near the clamped surface point and the end point near the seabed. The stiffened catenary is able to approximate the curvature at the surface point more accurate, but still falls short in the region near the seabed. When comparing the non-linear model and the enhanced stiffened catenary equation, besides from the curvature at the surface point, their development is quite similar. For the deep water pipelay, the curvature remains close to zero for a significant part of the pipeline, for all approximations. The biggest difference is seen in the first 500m. Since for this Thesis one will be predominantly looking at the behaviour of pipelay in deep water, the best approximations would be the enhanced stiffened catenary and the non-linear set of equations. Due to the greater flexibility of sensitivity analyses and the shorter computation time when using the enhanced stiffened catenary, it is chosen to continue with this solver for the approximation of the curvature during lay operations. This approximation which finds its origins in analytical calculations, will be used to make a 4DOF/element model in MATLAB for the estimation of the pipe twist during lay operations.

2.8. ESTIMATION OF THE TWIST DEVELOPMENT ALONG A PIPELINE DURING LAY OPERATIONS

During the pipelay operations at the Lucius project torsion was build up in the pipeline and relieved by letting the pipeline twist freely during torque relieval. Here, the twist ranged from 70 ° to 200 ° during the pipelay operations, depending on the type of pipeline and its total length. Under the assumption that residual curvature in the pipeline leads to pipeline twist, MATLAB models using analytical equations together with total potential energy minimization are used to estimate pipeline twist. For results comparison, finite element models in Abaqus are made with the same pipeline properties as used in the models using analytical equations. Both methods, analytical and FE, will be further elaborated in this chapter.

2.8.1. POTENTIAL ENERGY MINIMIZATION OF A PIPELINE DURING LAY OPERATIONS

In the literature study given in Appendix C, methods of approximating the pipeline twist during S-lay operations or during intermittent application of residual curvature during Reel-lay are discussed. The reoccurring theory to the twist approximations during laying, is an approach towards the minimization of the system's energy with respect to its amount of twist. The same approach will be applied for the analysis of the amount of twist found during the lay operations with the Aegir. Here the aim is to obtain the coupled differential equations between bending, torsion and axial tension. The first steps is to define the different terms that should be taken into account in the energy balance. The total potential energy of an elastic body is defined as the sum of all strain and potential energies.

$$\Pi = U + EP \tag{2.49}$$

Strain energy is stored in a body due to deformation. A pipeline during reeling is subjected to axial, bending, shear and torsional strain[14][15][16][17]. It is assumed that the contribution of the shear strain is negligible and therefore is not taken into account. Two second order terms also contribute to the energy of the system: these terms are linked to the effect of tension on rotation angles and large deflections[18]. Furthermore, the potential energy of the pipeline in the form of its own weight is taken to account during the formulation of the energy balance. Adding all these terms together in one obtains the following potential energy equation (see Equation 2.51).

$$\Pi = U_b + U_r + U_a + U_{t1} + U_{t2} + EP \tag{2.50}$$

$$\Pi = \frac{1}{2} \left[EI \int_{0}^{L} \left(\frac{d\theta}{ds} \right)^{2} ds + GJ \int_{0}^{L} \left(\frac{d\phi}{ds} \right)^{2} ds + EA \int_{0}^{L} \left(\frac{d^{2}u}{ds^{2}} \right)^{2} ds + \frac{TI}{A} \int_{0}^{L} \left(\frac{d\phi}{ds} \right)^{2} ds + T(\theta(s))^{2} \right] + \rho Agz ds \quad (2.51)$$

The bending angle derivative $\frac{d\theta}{ds}$ is equal to the curvature along the suspended pipeline during lay operations:

$$\frac{d\theta}{ds} = \kappa(s) \tag{2.52}$$

In the next step, the residual curvature can be added to the total curvature of the pipeline. This is done similarly as in Equation C.2 in the literature study:

$$\kappa(s,\phi_0,\kappa_r) = \kappa_n(s) + \kappa_r \cdot \cos\left(\phi(s)\right) \tag{2.53}$$

The main assumption here is that the amount of residual curvature κ_r is constant and applied over the total length of pipeline installed. This assumption is based on the following: as mentioned previously, during Aegir reel-lay operations a segment of 12m of the pipeline is cut for inspection before the whole pipeline is lowered. This segment is laid on deck and checked if its amount of residual curvature is within standards. If not, the process is repeated until by iteration the adequate straightener settings have been established. In the end, a completely straight pipeline is a 'pipe dream', and so one must always take into account a certain amount of residual curvature caused by the straightener settings and which is constantly applied over the whole pipeline length. It is also assumed that the rigidity of the straightener is maintained during the complete lay process. Substituting the bending angle derivative term for the curvature term and expanding the equation gives us the following total potential energy equation:

$$\Pi = \frac{1}{2} \left[EI \int_{0}^{L} \left(\kappa_{n}(s) + \kappa_{r} \cdot \cos\left(\phi(s)\right) \right)^{2} ds + GJ \int_{0}^{L} \left(\frac{d\phi}{ds}\right)^{2} ds + EA \int_{0}^{L} \left(\frac{d^{2}u}{ds^{2}}\right)^{2} ds + \frac{TI}{A} \int_{0}^{L} \left(\frac{d\phi}{ds}\right)^{2} ds + T(\theta(s))^{2} \right] + \rho Agz ds$$

$$(2.54)$$

One way to find the ultimate values of the potential energy of the elastic body is by taking the derivative with respect to the twist angle ϕ , and afterwards set the equation to zero (Equation 2.55).

$$\frac{d\Pi}{d\phi} = 0 \tag{2.55}$$

Terms that are not coupled with the torsion angle will disappear in the differentiation, and therefore will not contribute in finding the twist angle corresponding to the minimal potential energy. This is the case for the axial strain and the potential energy. The second order terms are also disregarded, for their contribution is considered negligible. Therefore the potential energy equation with terms that are coupled to the twist of the pipeline is given as follows:

$$\Pi(\phi) = \frac{1}{2} \left[EI \int_{0}^{L} \left(\kappa_{n}(s) + \kappa_{r} \cdot \cos\left(\phi(s)\right) \right)^{2} ds + GJ \int_{0}^{L} \left(\frac{d\phi}{ds}\right)^{2} ds \right]$$
(2.56)

Another way of finding the minima of the total energy of the system, is by calculating the total potential energy over the pipeline as a function of the torsion angle, and seek its minima without differentiation. This second method is used to solve the system. For the curvature of the vertical suspended pipeline prior to residual curvature addition κ_n , one has seen that there are several methods to obtain its approximation. These methods have been discussed in section 2.6. It is chosen to use the enhanced stiffened catenary equations for further analysis. For the Enhanced Stiffened Catenary, the final derived curvature equation can be analytically implemented into obtaining the minimal twist angle. For the energy minimization of a suspended pipeline during lay operations as approximated by enhanced stiffened catenary, the bending strain contribution can be written as follows:

$$U_b = \frac{1}{2} E I \int_0^L \left(\kappa_n(s) + \kappa_r \cos\left(\phi(s)\right) \right)^2 ds$$
(2.57)

$$U_{b} = \frac{1}{2} EI \int_{0}^{L} \left(\frac{a}{a^{2} + s^{2}} - \frac{a}{(a + wd)^{2}} e^{\left(\frac{s}{\gamma} - \frac{L}{\gamma}\right)} - \frac{a}{(a + s)^{2}} \frac{e^{\left(\frac{L}{\lambda} - \frac{s}{\lambda}\right)}}{e^{\left(\frac{L}{\lambda}\right)}} + \kappa_{r} \cos\left(\phi(s)\right) \right)^{2} ds$$
(2.58)

Here, the function $\phi(s)$ represents the development of the torsional angle along the pipeline from the seabed until the clamping of the pipeline at the tensioner. The function is based on the previously discussed literature study (Appendix C). Here is it assumed that the maximum torsional angle ϕ_0 occurs at the seabed, and decreased towards the surface. The torsional twist therefore has its maximum at the seabed, and the torque is assumed to be zero at the sea surface where the pipeline is constrained by the tensioner. The boundary conditions used are as follows:

$$\phi(0) = \phi_0$$
$$\frac{d\phi}{ds}(0) = 0$$
$$\phi(L) = 0$$

Here, *s* is the distance along the pipeline, where s = 0 is located at the seabed, and s = L is located at the sea surface. The torsional angle is given by the a second order polynomial (Equation 2.59) and is plotted in Figure 2.12 for a torsional angle of 123 [°]:

$$\phi(s) = -\frac{\phi_0}{L^2}s^2 + \phi_0 \tag{2.59}$$

The torsional strain energy is equal to:

$$U_r = \int_0^L GJ\left[\frac{d}{ds}(\phi(s))\right]^2 ds = 4GJ\frac{\phi_0^2}{L^4}\int_0^L s^2 ds$$
(2.60)

The total energy equation for the torsional related energies is given by Figure 2.9. Here the total energy U_T is dependent on the amount of residual curvature κ_r , the torsional angle at seabed ϕ_0 , material properties of the pipeline and its geometric configuration. By iteratively varying the torsional angle ϕ_0 while keeping the other parameters constant, an angle is found for which a given system has its minimum of energy. One can see in Figure 2.9 that the calculated torsional angle is equal to 123°. The corresponding curvature along the pipeline in comparison to a system without residual curvature, can be seen in Figure 2.10. The geometric shape of the pipeline is plotted in Figure 2.11 and in Table 2.1 the properties of the example are given.



Figure 2.9: The total, bending and torsional strain energies for the pipeline. The minimum point is marked with a cross, and indicates the expected torsional angle ϕ_0 .



Figure 2.11: Geometric configuration of pipeline during laying



Figure 2.10: The curvature along the pipeline before residual curvature addition (red) and after residual curvature addition in its final state (blue). Here the final state is achieved after a certain amount of torsion.



Figure 2.12: Development of torsion angle along pipeline

Table 2.1: Geometric, material and situational properties of exemplary pipeline during lay operations

Parameter	Value	Unit
D_o	8	["]
t_{st}	0.875	["]
t_c	2.5	["]
E_{st}	200	[GPa]
I_{st}	$6.74e^{-5}$	$[m^{4}]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
$ ho_w$	1025	$\left[\frac{kg}{m^3}\right]$
α	85	[°]
wd	2100	[<i>m</i>]
κ _r	10 ⁻³	$\left[\frac{1}{m}\right]$

2.8.2. FINITE ELEMENT ANALYSIS OP A PIPELINE DURING LAYING, USING ELASTIC BEAM ELEMENTS WITH RESIDUAL CURVATURE

As a comparative study, a pipeline with residual curvature after the straightening procedure is modeled in the finite element program Abaqus. It is investigated if the energy minimization through torsion during lay operations also occurs in this type of analysis, and how it may differ from the previously executed analyses in subsection 2.8.1, where the studies have their fundamentals in analytically derived equations. The actual results of the finite element models will be discussed elaborately in chapter 4. In this subsection, only the details of the models will be discussed.

Similarly as in section 2.5, a 6DOF/element FE model was created using beam elements to approximate the pipeline. Here the same type of beam elements, material properties, mesh density and pipeline-seabed interaction properties are used as in section 2.5. The model comprises several steps, each with their own boundary- and load conditions. These steps along with the aforementioned values will be discussed in the following. Furthermore, the boundary conditions and loads are given in Table 2.2 and the steps can be seen in Figure 2.13 till Figure 2.18.

1. Initial Conditions

In the primary step of the model, an initially curved beam is loaded into the 3D space (see Figure 2.13). This curved beam is formed by defining an arc with a certain curvature radius, representing the residual curvature that is present in the pipeline after exiting the straightener. The arc length is equal to the total suspended pipeline length for a certain lay configuration: the curvature is distributed evenly along the entire length of the pipeline.

2. Application of gravity

Instability issues during the simulations have lead to the gradual build-up of certain loads and boundary conditions during modeling. Therefore a step is introduced in which gravity is applied to the whole system M_T with a ramp function (see Figure 2.14). Here, the same equivalent gravity constant after buoyancy compensation is used as in section 2.5. The pipe end at the surface P_s is clamped in at a given top angle α , and the lower pipe end P_b is constrained in all translational directions. In this step the whole pipeline M_T is constrained in the translational z-direction (see Figure 2.19) for stability purposes. For the higher curvature models during the sensitivity analysis, the final gravity applied to the system was built up in two steps.

3. Lowering of pipeline

In the third step, the boundary condition on the whole model M_T and the lower pipe end P_b boundary conditions are removed, with exception of the translational constraint in z-direction (see Figure 2.19) at the lower pipe end. The pipeline is lowered and ends up in a vertical suspended state (see Figure 2.16). During the lowering, a small, evenly distributed load perpendicular to the in-plane movement is applied along the pipeline (F_i). This load is applied to instigate the pipeline to twist in an energetically unstable configuration, while the magnitude of it remains of negligible influence to the final results.

4. Application of horizontal tension

In the fourth step, the horizontal load is applied in x-direction (see Figure 2.19) to the lower pipe end P_b . Depending on the analysis, the pipeline is either bent against or in the same direction as the curvature of the initial configuration. The horizontal load T_H represents the horizontal tension in the pipeline during laying in a certain configuration, and is calculated using the enhanced stiffened catenary theory. See Figure 2.17.

5. Interaction with seabed

In the concluding step, the interaction with the seabed occurs (see Figure 2.18). The seabed is raised, and has interac-

tion with the last few meters of the pipeline, giving it the characteristic catenary shape. Please note that the interaction of the beam elements with the rigid seabed is modeled as frictionless. The same interaction conditions apply as for the model in section 2.5. After completion, the total amount of twist is deduced in the system, and its build up along the system.



Figure 2.16: End of lowering step

Figure 2.17: Application of horizontal tension

Figure 2.18: Interaction with seabed

In Table 2.2 and overview is made of the boundary conditions and loads applied on the different instances of the model for the model steps. The table must be read as follows:

- Boundary conditions can be applied in the 6 DOF directions. When an instance is constrained in a certain direction, the constraint is marked with a dot (•). If the instance is free in a given direction, a horizontal dash is given (-). The instances of the model can be seen in Figure 2.13.
- Load conditions can also be activated in the 6 DOF directions. Here the same applies: a dot (•) indicates a load is engaged and a dash (-) signifies no load is present. Once a load is engaged, also the type of load is given in the table



Figure 2.19: Global coordinate system that is used for the modeling of the pipeline in Abaqus using FE analysis.

Model step	Instance	В	oun	dary	r con	ditic	ons			Lo	ads				
		x	у	z	θ	ψ	φ	х	у	z	θ	ψ	φ	Туре	Unit
1	P_s	L	•	•	•	•	•	+							
	P_b	•	•	•	•	•	•					-		-	-
	M_T	•	٠	•	٠	٠	٠	-	-	-	-	-	-	-	-
2	P_s	•	•	•	•	•	•								
	P_b	•	•	•								-			
	M_T	-	-	٠	-	-	-	-	•	-	-	-	-	g eq	$\left[\frac{m}{s^2}\right]$
3	P_s	•	•	•	•	•	•	-							
	P_b			•		-		•						T_{H}	[<i>N</i>]
	M_T	-	-	-	-	-	-	-	•	-	-	-	-	geq	$\left[\frac{m}{s^2}\right]$
		-	-	-	-	-	-	-	-	•	-	-	-	F_i	[N]
4	P_s	•	•	•	•	•	_ • _								
	P_b			•				•				-		T_{H}	[<i>N</i>]
	M_T	-	-	-	-	-	-	-	•	-	-	-	-	g eq	$\left[\frac{m}{s^2}\right]$
		-	-	-	-	-	-	-	-	•	-	-	-	F_i	[N]
5	P_s	•	•	•	•	•	_ • _								
	P_b			•				•				-		T_{H}	[<i>N</i>]
	M_T	-	-	-	-	-	-	-	•	-	-	-	-	geq	$\left[\frac{m}{s^2}\right]$
		-	-	-	-	-	-	-	-	٠	-	-	-	F_i	[N]

Table 2.2: The boundary conditions and loads als defined per step on the different instances

3

ANALYSIS TWIST DEVELOPMENT OF A VERTICAL FREELY SUSPENDED PIPELINE DURING LOWERING OPERATIONS

In the first stage of pipe lay during reel-lay operations with the Aegir, the pipeline is lowered vertically and remains freely suspended until touchdown as is schematically depicted in Figure 3.1. Usually the first end of the pipeline is fitted with an end terminal in the Reel-J-lay tower before lowering. This end terminal can have both a large mass, increasing the overall tension in the system significantly, and a high surface area, making the system sensitive to current loading. In order to approximate the geometric properties of the pipeline under different loadings and with residual curvature present in the pipeline, methods with linear and non-linear equations are used. These are discussed in the following chapter and compared with one another.



Figure 3.1: Pipeline lowering with the DCV Aegir

Once the geometric configuration of the pipeline is approximated, the twist development during lowering can be analyzed. Using total potential energy minimization and finite element analysis an attempt is made to obtain similar twist data as observed during the Lucius pipeline installation operations (see Figure 3.2).



Figure 3.2: Twist data of pipeline installation operations with the DCV Aegir. Left from dotted line: twist data during pipeline lowering, Right from dotted line: twist data during laying (not to be considered in this chapter).

3.1. APPROXIMATION PIPELINE LOWERING USING LINEAR EQUATIONS OF MOTION

A pipeline during lowering under current loading can be analyzed comparatively with an axially tensioned beam subjected to a uniform lateral load^[19].

3.1.1. COMPARISON VERTICAL SUSPENDED PIPELINE TO AXIALLY TENSIONED BEAM

The self weight of the pipeline and its end terminal causes an axial force which is dependent on the location x. To understand the effect of an axial force P(x, t) on a laterally loaded beam, let us consider the configuration as given by 3.3(a): a clamped horizontal beam under a uniform lateral load and tension P(x, t). Using the beam element 3.3(b) the equations of motion are computed.



(a) Clamped beam under axial tensile load



(b) Beam element under axial tensile load

Figure 3.3: Bending of beam under uniform lateral load and axial tension[19].

The equation of motion in the lateral direction z are given by the following:

$$V - \left(V + \frac{\partial V}{\partial x}dx\right) + f dx + \left(P + \frac{\partial P}{\partial x}dx\right) \cdot \sin\left(\theta + \frac{\partial \theta}{\partial x}dx\right) - P \sin\theta = \rho A dx \frac{\partial^2 w}{\partial t^2}$$
(3.1)

Which is equivalent to,

$$V - (V + dV) + f dx + (P + dP) \cdot \sin(\theta + d\theta) - P \sin\theta = \rho A dx \frac{\partial^2 w}{\partial t^2}$$
(3.2)

For small deflections the following applies:

$$sin(\theta + d\theta) \cong \theta + d\theta = \theta + \frac{\partial\theta}{\partial x}dx$$
$$\theta + \frac{\partial\theta}{\partial x}dx = \frac{\partial w}{\partial x} + \frac{\partial^2 w}{\partial x^2}dx$$
(3.3)

By filling in *sin* θ and *sin*(θ + $d\theta$) of Equation 3.2 one obtains the following:

$$-dV + f dx + (P + dP) \cdot \left(\frac{\partial^2 w}{\partial x^2} dx\right) + dP \frac{\partial w}{\partial x} = \rho A dx \frac{\partial^2 w}{\partial t^2}$$
(3.4)

For the rotational motion around the point O, the equation is motion is:

$$\left(M + \frac{\partial M}{\partial x}dx\right) - M - \left(V + \frac{\partial V}{\partial x}dx\right) \cdot dx + f \, dx \cdot \frac{dx}{2} = 0$$
$$dM - (V - dV) \cdot dx + f \frac{dx^2}{2} = 0 \tag{3.5}$$

From elementary beam theory for the bending of beams (better known as Euler-Bernoulli or thin beam theory), the relation between bending moment and deflection can be expressed as Equation 3.6

$$M(x,t) = EI(x)\frac{\partial^2 w}{\partial x^2}(x,t)$$
(3.6)

By writing dV and dM as partial derivatives once more and disregarding terms involving second powers in dx, Equation 3.5, Equation 3.6 and 3.4 are combined to obtain a single differential equation of motion:

$$\frac{\partial^2}{\partial x^2} \left(EI \frac{\partial^2 w}{\partial x^2} \right) + \rho A \frac{\partial^2 w}{\partial t^2} - P \frac{\partial^2 w}{\partial x^2} = f(x, t)$$
(3.7)

The load and mass terms of the differential Equation 3.7 are now fitted for the given system of a pipeline with an end terminal. The mass of the beam is now dependent on three contributions: the mass of the pipeline itself m_p , the hydrodynamic added mass m_a and the mass of the pipeline content m_{ct} . Axial force is equal to the effective tension term of the pipeline $T_e(z)$, with is dependent on it position along the pipeline. The beam model which represents the vertical pipeline is given in Figure 3.4.



Figure 3.4: Tension development in the pipeline over the waterdepth

Now that the axial tension in the system is dependent on its position along the z axis, a small alteration has to be made to one of the assumptions for small deflections. Equation 3.3 will now be written as the following for the vertical suspended pipeline as depicted in Figure 3.4:

$$\theta + \frac{\partial \theta}{\partial z} dz = \frac{\partial x}{\partial z} + \frac{\partial}{\partial z} \left(\frac{\partial x}{\partial z} \right) dz$$
(3.8)

Finally, this alteration leads to the system's equation of motion as given by Equation 3.9

$$EI\frac{\partial^2}{\partial z^2} \left(\frac{\partial^2 x}{\partial z^2}(z,t) \right) - \frac{\partial}{\partial z} \left(T_e(z)\frac{\partial x}{\partial z}(z,t) \right) + (m_p + m_a + m_{ct})\frac{\partial^2 x}{\partial t^2}(z,t) = f(z,t).$$
(3.9)

As has been done during the analysis of the suspended pipeline during laying, only the static configuration will be discussed. Therefore the time dependent mass terms will further not be taken into account and will be discarded from Equation 3.9. The effective tension in the pipeline is dependent on its submerged weight W_s and its bottom tension T_b , caused by the weight of the end terminal. See Equation 3.10.

$$T_e(z) = T_b + (A_{st}\rho_{st} - A_{ex}\rho_{ex} + A_i\rho_i)g(L-z)$$

$$T_e(z) = T_b + W_s(L-z)$$
(3.10)

Where

 $\begin{array}{l} A_{st} = \text{Cross-sectional area pipeline } \begin{bmatrix} m^2 \end{bmatrix} \\ A_{ex} = \text{External cross-sectional area pipeline } \begin{bmatrix} m^2 \end{bmatrix} \\ A_i = \text{Internal cross-sectional area pipeline } \begin{bmatrix} m^2 \end{bmatrix} \\ \rho_{st} = \text{Density steel } \begin{bmatrix} \frac{kg}{m^3} \end{bmatrix} \\ \rho_{ex} = \text{Density external medium pipeline } \begin{bmatrix} \frac{kg}{m^3} \end{bmatrix} \\ \rho_i = \text{Density internal medium pipeline } \begin{bmatrix} \frac{kg}{m^3} \end{bmatrix} \\ L = \text{Submerged pipeline length } \begin{bmatrix} m \end{bmatrix} \\ W_s = \text{Submerged weight of pipeline } \begin{bmatrix} \frac{N}{m} \end{bmatrix} \end{array}$

During lowering, the pipeline is hermetically sealed, therefore unlike similar analysis of functioning risers where a fluid is present in the pipeline, the only medium present in the pipeline during lowering is air. Its weight contribution is negligible and therefore the buoyancy of the system will effectively be determined by the displaced water volume. A flooded condition of the pipeline is not considered. Next the lateral load is computed. Using the Morison's equation for slender beams [20] the horizontal load on the beam representing the pipeline can be given as:

$$f(z,t) = f_I(z,t) + f_D(z,t) f(z,t) = C_m \cdot \rho_w \frac{\pi}{4} D^2 \cdot \frac{dv}{dt}(z,t) + \frac{1}{2} C_d \cdot \rho_w D \cdot v(z,t) |v(z,t)|$$
(3.11)

Since the problem is described by a static system, the inertia component of the Morison Equation is neglected. Furthermore, the current is assumed constant over the water depth. This is proven to be a valid assumption given the metocean data at the Lucius project site (Appendix G). The drag force coefficient for pipelines as given by DNV in [21] is equal to 0.7. The lateral current loading is simplified to the following:

$$f = \frac{1}{2} C_d \cdot \rho_w D \cdot v |v|$$
(3.12)

Filling the static drag forces of the Morison Equation along with the effective tension terms into the equation of motion 3.9, and disregarding the time dependent mass terms, one obtains the following:

$$EI\frac{\partial^4 x}{\partial z^4}(z) - (T_b + W_s(L-z))\frac{\partial^2 x}{\partial z^2}(z) + W_s\frac{\partial x}{\partial z}(z) = \frac{1}{2}C_d \cdot \rho_w D \cdot v|v|$$
(3.13)

For the given problem, four boundary conditions can be computed. At the clamped start point at x = 0 the displacement is zero and the angle is equal to $\frac{\pi}{2}$ rad minus the top angle α in [rad] of the system. For the end at x = L a shear force is created by the horizontal displacement of the end terminal as a results of external loading. The bottom tension T_b can be decomposed into a vertical contribution and a horizontal contribution, where its absolute values are dependent on the displacement angle at the end of the pipeline. The displacement angle at the end of the pipeline is assumed small, and therefore has been linearized, as can be seen in Equation 3.18. Usually the COG of the end terminal is not exactly in line with the COG of the pipeline. However, for the pipeline end terminals used at Lucius, the PLET's COG offset from the pipeline's centerline is relatively small. Therefore, no resulting moment from an end terminal COG offset will be defined in the boundary conditions. The current loading on the large surface area of the PLET however, must be taken into account. The surface area of the mudmat of the PLET for the PF-01 is used for current load calculations (Appendix E). The drag force coefficient C_d is higher for the PLET than for the pipeline itself. A drag force coefficient of 1.7 is used by Heerema for PLETs [22]. The horizontal force induced by the current on the PLET is added to the shear boundary conditions. The force is given by Equation 3.14.

$$F_{H,PLET} = \frac{1}{2} \cdot \rho_w \cdot C_{d,PLET} \cdot A_{H,PLET} \cdot v_c^2$$
(3.14)

The boundary conditions for the pipeline with pipeline end terminal under current loading are equal to:

$$\begin{aligned} x|_{z=0} &= 0 \\ \partial x|_{z=0} &= 0 \end{aligned} \tag{3.15}$$

$$\left. \frac{\partial x}{\partial z} \right|_{z=0} = \frac{\pi}{2} - \alpha \tag{3.16}$$

$$\left. \frac{\partial^2 x}{\partial z^2} \right|_{z=L} = 0 \tag{3.17}$$

$$\frac{\partial^3 x}{\partial z^3}\Big|_{z=L} = -T_b \frac{\partial x}{\partial z}(L) + F_{H,PLET}$$
(3.18)

3.1.2. Approximation pipeline lowering with residual curvature using linear equations of motion

Now that the linearized equation of motion for a beam under tensile force and lateral loading has been obtained, residual curvature will be added to the system. Identically to the analysis done for the suspended pipeline during laying, the residual curvature simulates the effect of a straightener which doesn't bend the pipeline back completely straight after reeling. One must note that the residual curvature added will be linearized, and therefore simplified in comparison to a non-linear system with large deflections. In a linear system, the length of the pipeline remains equal to the projected length of the pipeline, permitting the linearization of the curvature as done in Equation 3.19. This gives a fundamental difference between non-linear beam theory and linearized beam theory such as Euler Bernoulli [17], where the rotations are considered small and the actual length of the beam and the projected length are considered equal.

$$\kappa(z) = \begin{cases} \frac{d^2 x}{dz^2} & \text{if } \frac{d\theta}{ds} \cong \frac{d\theta}{dz} & \text{(Linearized system)} \\ \frac{\frac{d^2 x}{dz^2}}{\left(1 + \left(\frac{dx}{dz}\right)^2\right)^{\frac{3}{2}}} & \text{(Non-linear system)} \end{cases}$$
(3.19)

To approximated the shape of a pipeline which is curved due to plastic deformation, a system is modeled where a beam is pre-curved before it is subjected to loads. To achieve this pre-shape, a moment is added to the system which gives the beam a specific residual curvature $\kappa_r(z)$. It is assumed that the residual curvature after the pipeline has passed the straightener is constant, and therefore the applied moment M(z) to the system to approximate the curved shape

will also be constant. The relations between a moment applied on a beam and the corresponding curvature is seen in Equation 3.20.

$$\frac{\partial^2 x}{\partial z^2} = \frac{M(z)}{EI} = \kappa(z)$$
(3.20)

The added residual curvature will once again be made dependent on the torsion of the pipeline, similar to the analysis of the suspended pipeline during laying. The total curvature of the pipeline now consists out of the curvature of a tensioned beam under lateral loading, and the added residual curvature.

$$\kappa(z) = \frac{\partial^2 x}{\partial z^2} + \kappa_r(z)$$

The derivative terms of the residual curvature are obtained, and are added to the equation of motion of the system.

$$\kappa_r(z) = \kappa_r \cos(\phi(z)) \tag{3.21}$$

$$\frac{\partial \kappa_r(z)}{\partial z} = -\kappa_r \frac{\partial \phi(z)}{\partial z} \sin(\phi(z))$$
(3.22)

$$\frac{\partial^2 \kappa_r(z)}{\partial z^2} = -\kappa_r \frac{\partial^2 \phi(z)}{\partial z^2} \sin\left(\phi(z)\right) - \kappa_r \left(\frac{\partial \phi(z)}{\partial z}\right)^2 \cos\left(\phi(z)\right)$$
(3.23)

Where the development of the torsion angle has its origins in the linearization of Equation 2.59 in subsection 2.8.1. The function itself however has undergone a transformation, complying with the new starting point for this analysis. Contrary to the sagbend analysis in subsection 2.8.1, the starting point is now at the surface, where the pipeline is clamped by the tensioners. This has also been done to comply with convergence issues with the numerical solver, for starting the analysis at a free end point leads to instability. The linearized, transformed torsional angle function can be seen in Equation 3.24.

$$\phi(z) = \phi_0 \left(1 - \frac{(z-L)^2}{L^2} \right) \tag{3.24}$$

The equation of motion of a linearized beam under axial tension, lateral loading and with residual curvature is equated by the the following:

$$EI\left(\frac{\partial^2}{\partial z^2}\left(\frac{\partial^2 x}{\partial z^2} + \kappa_r(z)\right)\right) - (T_b + W_s(L-z))\left(\frac{\partial^2 x}{\partial z^2} + \kappa_r(z)\right) + W_s\frac{\partial x}{\partial z}(z) = f$$
(3.25)

$$EI\left(\frac{\partial^4 x}{\partial z^4} - \kappa_r \frac{\partial^2 \phi(z)}{\partial z^2} \sin\left(\phi(z)\right) - \kappa_r \left(\frac{\partial \phi(z)}{\partial z}\right)^2 \cos\left(\phi(z)\right)\right) - (T_b + W_s(L-z)) \left(\frac{\partial^2 x}{\partial z^2} + \kappa_r \cos\left(\phi(z)\right)\right) + W_s \frac{\partial x}{\partial z}(z) = f \qquad (3.26)$$

For $\phi_0 = 0....2\pi$

3.1.3. Solving the linear equation of motion for a vertical pipeline with residual curvature using numerical operations

To solve the boundary value problem of the ordinary differential equation, a built-in numerical solver was used of MAT-LAB. The so called BVP4C solver implements a collocation method for the solution of boundary value problems[23][24]. To use the solver, one must define three components of the function:

- The boundary conditions of the system;
- The vector of ordinary differential equations which make up the system;
- · Estimated initial values for the ordinary differential equations

The solver has been verified by comparing the analytically derived results for a tensioned vertical pipeline under current loading without residual curvature. The output of the solver was compared with the solutions found from the analytical derivation and were almost identical. To solve the 4^{th} order differential equation given in Equation 3.26, the system has to be written as a vector of 1^{st} order differential equations. Here the index of the vector is linked to the order of the derivative of the displacement equation. So the 1^{st} order differential equation vector becomes:

$$\vec{dx} = \begin{bmatrix} \frac{\partial x}{\partial z} \\ \frac{\partial^2 x}{\partial z^2} \\ \frac{\partial^3 x}{\partial z^3} \\ \frac{\partial^4 x}{\partial z^4} \end{bmatrix} = \begin{bmatrix} x(2) \\ x(3) + \kappa_r \cos(\phi(z)) \\ x(4) \\ \kappa_r \frac{\partial^2 \phi(z)}{\partial z^2} \sin(\phi(z)) + \kappa_r \left(\frac{\partial \phi(z)}{\partial z}\right)^2 \cos(\phi(z)) \dots \\ \dots + \frac{1}{F_l} (T_b + W_s(L-z)) \left(x(3) + \kappa_r \cos(\phi(z))\right) - \frac{W_s}{F_l} x(2) + \frac{f}{2F_l} \end{bmatrix}$$
(3.27)

The boundary conditions are also added as a vector. Here the boundary conditions at the clamped begin point of the beam are assembled as a vector *ya*, and the boundary conditions at the end point of the beam are noted as *yb*. Here once again the index of the vector denotes the order of the derivative. For the estimated initial values of the derivatives, the values are filled in as expected given the boundary conditions. The initial estimation has been found to influence the final results of the ordinary differential equations, bot can mostly improve or worsen the computation time of the solver.

$$\overrightarrow{v_{BC}} = \begin{bmatrix} x_a(1) \\ x_a(2) + \alpha - \frac{\pi}{2} \\ x_b(3) \\ x_b(4) + T_b x_b(2) - F_{H,PLET} \end{bmatrix} \qquad \overrightarrow{v_i} = \begin{bmatrix} 0 \\ \frac{\pi}{2} - \alpha \\ \kappa_r \\ 1 \end{bmatrix}$$
(3.28)

Using the solver, the displacement *x*, the rotation in the in-plane bending plane $\frac{\partial x}{\partial z}$, the moment $EI\frac{\partial^2 x}{\partial z^2}$ and the shear force $EI\frac{\partial^3 x}{\partial z^3}$ along the beam can be computed. In Figure 3.5 until Figure 3.8 the deflection, angular rotation, moment and shear are plotted for a 2100 m long beam with the specifications as given in Table 3.1.3. The modeled beam in the example is similar to a vertical, freely suspended pipeline as used during installation with the Aegir at the Lucius project. Important note: the graphs plotted are for a twist angle ϕ equal to zero.



Figure 3.5: Horizontal displacement along the beam







Figure 3.6: Displacement angle along the beam



Figure 3.8: Shear force along the beam

Parameter	Value	Unit
κ _r	0.002	$\left[\frac{1}{m}\right]$
D_o	8.625	["]
t_{st}	0.875	["]
t_c	2.5	["]
E_{st}	200	[GPa]
I_{st}	$6.74e^{-5}$	$[m^{4}]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
ρ_w	1025	$\left[\frac{kg}{m^3}\right]$
α	90	[°]
L	2100	[m]
v_c	0.1	$\left[\frac{m}{s}\right]$
W_{PLET}	28631	[mT]
$A_{H,PLET}$	87.15	$[m^2]$
C_d	0.7	[-]
$C_{d,PLET}$	1.7	[-]

Table 3.1: Properties of pipeline during lowering operation.

3.2. Approximation pipeline lowering using non-linear equations of motion

The vertical suspended pipeline during lowering can also be approximated by a non-linear model, similar to the one used in section 2.4. In this case, one assumes that the pipeline can be modeled as a beam, subject to possible large horizontal deflections due to forcing. For the modeling of a vertically tensioned beam under large deflections, similar governing equations like 2.19 and 2.21 from section 2.4 for a beam segment can be used. Given the different axes definition in comparison with section 2.4, the equilibrium equations along with the corresponding governing equations are derived once more for this system. The beam segment with the new axis definition can be seen in Figure 3.9, and the governing equations are given by Equation 3.31 and Equation 3.32. The boundary conditions will evidently be different for a vertical suspended pipeline in comparison to a pipeline during laying. At the pipe end at the surface, the pipeline will be clamped by the tensioners in the top angle α at which the lowering takes place. At the end of the pipeline, the pipeline will experience a shear force contribution due to the end terminal and the current loading on the PLETs surface area. Given the axes and angle definition, the PLET's weight shear force contribution will be equal to the cosine of the angle that the end terminal's bottom tension makes with the pipeline. The current loading on the PLET is again given as the shear force contribution $F_{H,PLET}$. Similarly to the linearized pipeline, the resulting moment caused by the offset of the PLET's COG is regarded as negligible and is not included into the boundary conditions. The system for the vertical suspended pipeline is schematically drawn in Figure 3.10 and its boundary conditions can be seen from Equation 3.33 to Equation 3.36.



Figure 3.9: Beam segment with axis definition for pipeline during lowering.



Figure 3.10: Schematic representation of vertical pipeline during lowering

$$\Sigma F_z = 0$$

$$Tsin\theta - (T + dT) \cdot sin(\theta + d\theta) + (V + dV) \cdot cos(\theta + d\theta) - V \cdot cos\theta + W_s \cdot ds = 0$$
(3.29)

The same equilibrium of forces is sought after in the horizontal plane,

$$\Sigma F_x = 0$$

$$T\cos\theta - (T + dT)\cos(\theta + d\theta) + V\sin\theta - (V + dV) \cdot \sin(\theta + d\theta) + f(s) \cdot ds = 0$$
(3.30)

$$\underline{G_{1:}} \qquad EI\frac{d^{3}\theta}{ds^{3}} - T\frac{d\theta}{ds} + W_{s}cos\theta - f(s)sin\theta = 0 \qquad (3.31)$$

$$\frac{G_2}{ds} = W_s sin\theta + f(s)cos\theta$$
(3.32)

$$\theta|_{s=0} = \alpha \tag{3.33}$$

$$\left. \frac{\partial \theta}{\partial s} \right|_{s=0} = 0 \tag{3.34}$$

$$\left. \frac{\partial \theta}{\partial s} \right|_{s=L} = 0 \tag{3.35}$$

$$\left. \frac{\partial^2 \theta}{\partial s^2} \right|_{s=L} = -T_b \cos(\theta(L)) + F_{H,PLET}$$
(3.36)

3.2.1. APPROXIMATION PIPELINE LOWERING WITH RESIDUAL CURVATURE USING NON-LINEAR EQUATIONS OF MOTION AND SOLVING THE SYSTEM NUMERICALLY

In the next step, residual curvature will be added to the non-linear governing equations of motion, just as has been done in subsection 3.1.2. Given our non-linear system, the curvature of the beam with residual curvature is equal to:

$$\kappa(s) = \frac{d\theta}{ds} + \kappa_r(s) \tag{3.37}$$

Where $\kappa(s)$ is the non-linearized version of the torsion angle Equation 3.24 times the absolute value of the residual curvature present in the pipeline after straightening:

$$\kappa_r(s) = \kappa_r \cos(\phi(s)) \tag{3.38}$$

$$\phi(s) = \phi_0 - \phi_0 \frac{(s-L)^2}{L^2}$$
(3.39)

Using the 1st and 2nd derivative of the residual curvature function, the governing equation Equation 3.31 is transformed into the governing equation of a system with residual curvature:

$$EI\frac{d^{2}}{ds^{2}}\left(\frac{d\theta}{ds} + \kappa_{r}(s)\right) - T_{e}(s)\left(\frac{d\theta}{ds} + \kappa_{r}(s)\right) + W_{s}cos\theta = -f(s)sin\theta$$
$$EI\left(\frac{d^{3}\theta}{ds^{3}} - \kappa_{r}\left(\frac{d\phi}{ds}\right)^{2}cos\left(\phi(s)\right) - \kappa_{r}\frac{d^{2}\phi}{ds^{2}}sin\left(\phi(s)\right)\right)$$
$$- (T_{b} + W_{s}(L-s))\left(\frac{d\theta}{ds} + \kappa_{r}cos\left(\phi(s)\right)\right) + W_{s}cos\theta = -f(s)sin\theta$$
(3.40)

The second governing equation for a vertical suspended system with residual curvature remains unchanged to a system with residual curvature. Given the fact that during lowering the length of the submerged pipeline is known beforehand, in contrary to the length of a submerged pipeline during laying, the second governing equation is not needed to solve the system. The system can be solved numerically once again using the MATLAB solver BVP4C[23]. The derivative vector that is used as input is given in Equation 3.41.

$$\vec{d\theta} = \begin{bmatrix} \frac{d\theta}{ds} \\ \frac{d^2\theta}{ds^2} \\ \frac{d^3\theta}{ds^3} \end{bmatrix} = \begin{bmatrix} \theta(2) + \kappa_r \cos(\phi(s)) \\ \theta(3) \\ \kappa_r \left(\frac{d\phi}{ds}\right)^2 \cos(\phi(s)) + \kappa_r \frac{d^2\phi}{ds^2} \sin(\phi(s)) + \frac{(T_b + W_s(L-s))}{EI} \left(\theta(2) + \kappa_r \cos(\phi(s))\right) - \frac{W_s}{EI} \cos\theta(1) - \frac{f(s)}{EI} \sin\theta(1) \end{bmatrix}$$
(3.41)

The boundary conditions as described earlier in Equation 3.33 to Equation 3.36 will change slightly once adding residual curvature to the system: the moment at the clamping point is now equal to κ_r , the rest of the boundary conditions remain the same. The boundary conditions for the system are given as input to the BVP4C solver. The boundary condition vector and the initial values vector can be seen in 3.42. Here, the clamped pipe end is seen as the begin point *a*, and the other pipe end as point *b*. Again the same important note: the graphs plotted are for a twist angle ϕ equal to zero.





Figure 3.14: Shear force along the beam

Similar to subsection 3.1.3, the displacement *x*, the rotation in the in-plane bending plane $\frac{\partial x}{\partial z}$, the moment $EI\frac{\partial^2 x}{\partial z^2}$ and the shear force $EI\frac{\partial^3 x}{\partial z^3}$ along the beam are computed. The results can be seen in Figure 3.11 until Figure 3.14. The same pipeline properties (Table 3.1.3) are used as in the linear approximation in subsection 3.1.3. A comparison of the results between the linear and non-linear approximation of the vertically suspended pipeline is made in the following section.

3.3. DISCUSSION RESULTS LINEAR AND NON-LINEAR APPROXIMATION VERTICAL BEAM WITH RESIDUAL CURVATURE AND CURRENT LOADING 31

3.3. DISCUSSION RESULTS LINEAR AND NON-LINEAR APPROXIMATION VERTICAL BEAM

WITH RESIDUAL CURVATURE AND CURRENT LOADING

Comparing the non-linear Equation 3.40 with the similar linear Equation 3.26 for the vertically tensioned pipeline with residual curvature and current loading, one can see the similarity. Besides the fact that the non-linear equation is 3^{rd} order and the linearized equation is 4^{th} order, one can find upon inspection the similar contributing terms: both contain a term concerning the bending stiffness, axial tension, self weight and external load. It is therefore not surprising that model comparison shows quite similar results.

3.3.1. MODEL COMPARISON CONCERNING SHEAR DEVELOPMENT ALONG THE PIPELINE

When comparing the shear development along the pipeline from the linear model (Figure 3.8) with the one of the nonlinear model (Figure 3.14), one sees that the are almost identical. Both the absolute shear values as the development of more or less the same. The steep increase of the shear at the end of the graph is in both models caused by horizontal current loading on the PLET.

3.3.2. MODEL COMPARISON CONCERNING MOMENT DEVELOPMENT ALONG THE PIPELINE

Also the moment development along the pipeline is almost identical for both models (Figure 3.7 and Figure 3.13). It can be seen that the moment in the pipeline is predominantly determined by the residual curvature in the pipeline, giving the moment development a constant character. From this it can also be concluded that the current loading has little effect on the moment, at least for the current value used for the analysis.

3.3.3. MODEL COMPARISON CONCERNING DISPLACEMENT ANGLE DEVELOPMENT ALONG THE PIPELINE

Upon comparison of the models regarding the displacement angle, one sees a difference. The development along the pipeline shows as similar trend, only the displacement angle graph of the non-linear model (Figure 3.12) gives higher absolute values throughout in comparison with the similar linear graph (Figure 3.6). Although there is a difference in absolute values, one must note that the maximum angle in both models is still very small. Converted to degrees, the maximum angle for both models does not exceed the 1°. This must be taken into consideration during contemplation of the significance of the different absolute displacement angle values.

3.3.4. MODEL COMPARISON CONCERNING HORIZONTAL DISPLACEMENT ALONG THE PIPELINE

Lastly, the horizontal displacement along the pipeline is compared (Figure 3.5 and Figure 3.11). The development for both the models is quite similar, only the non-linear model gives greater absolute values due to greater values for the displacement angle as described earlier. As could be expected, the small displacement angles lead to a small horizontal displacement along the pipeline. Over a length of more than 2km, the pipeline has a maximum offset of only a few meters for both models.

3.3.5. OVERALL MODEL COMPARISON CONCLUSION

Comparing the results of the shear, moment, displacement angle and horizontal displacement of the linear and nonlinear models, one can state that there are quite similar. Due to the small displacement angles, the linearization of the system does not lead to large discrepancies in the results. The most important value for further calculations is the development of the curvature along the pipeline. The curvature (which has an identical development compared to the moment) is almost identical for the linear and non-linear model, and will therefore give the same amount of bending strain energy. It is therefore concluded that there will be no difference in results when using the linear or the non-linear MATLAB model for the calculation of the twist during pipeline lowering operations. Given this indifference, the linear model will be used for further calculations due to easier computations.

3.3.6. ESTIMATION OF THE TWIST DEVELOPMENT DURING PIPELINE LOWERING

The twisting of the pipeline already occurred during lowering operations from the vessel to the seabed. The twist in the pipeline is then mitigated by rotating the PLET back to its desired angle or by temporarily changing the heading of the vessel. Depending on the type of pipeline and how far it was during the lowering process, twist angles occurred between approximately 10 ° until 90 °. Similar to the twist estimation done in section 2.8, models are made to research the effect that residual curvature in the pipeline has on the development of twist. MATLAB models using the linearized beam equations with residual curvature as described in chapter 3 together with total potential energy minimization are used to estimate the twist. And once again, finite element models in Abaqus with the same pipeline properties as used in the equivalent MATLAB models are used for results comparison.

3.3.7. POTENTIAL ENERGY MINIMIZATION FOR A PIPELINE DURING LOWERING

With the equations of motion for a vertical suspended pipeline with residual curvature and current loading having been derived, a closer look is taken to the energy balance of the system. Similar to the potential energy minimization done in subsection 2.8.1, the aim is to obtain the coupled differential equations between the different strain contributions of the system. The total potential energy equation for a pipeline during lowering operations is given in Equation 3.44. Note that the second order contributions as described in subsection 2.8.1 have once again been disregarded for the total potential energy minimization of the pipeline.

$$\Pi = U_b + U_r + U_a + EP \tag{3.43}$$

$$\Pi = \frac{1}{2} \left[EI \int_{0}^{L} \left(\frac{d\theta}{ds} \right)^{2} ds + GJ \int_{0}^{L} \left(\frac{d\phi}{ds} \right)^{2} ds + EA \int_{0}^{L} \left(\frac{d^{2}u}{ds^{2}} \right)^{2} ds \right] + \rho Agz ds$$
(3.44)

The bending angle derivative $\frac{d\theta}{ds}$ is equal to the curvature along the vertical suspended pipeline during lowering. In chapter 3 MATLAB models are made using analytical beam equations which can take into account residual curvature present in the pipeline after the pipeline has passed the straightener. This residual curvature is made dependent on the twist angle, making it possible to calculate the bending strain contribution of a system with residual curvature for different twist angles. Both the axial strain and the potential energy of the vertical suspended pipeline remain constant during pipeline twist. Ergo, since these terms are not coupled to the twist of the pipeline, they are disregarded in further analysis. The potential energy equation with terms that are coupled to the twist of the pipeline are therefore given as follows:

$$\Pi(\phi) = \frac{1}{2} \left[EI \int_{0}^{L} \left(\kappa(s,\phi(s)) \right)^{2} ds + GJ \int_{0}^{L} \left(\frac{d\phi}{ds} \right)^{2} ds \right]$$
(3.45)

Where $\kappa(s, \phi(s))$ is the curvature along the pipeline for a given twist angle as calculated using the linear MATLAB models for a vertical suspended pipeline with residual curvature. The estimated development of the twist angle is given in Equation 3.24. Identical to chapter 3, the twist angle is iteratively varied and the corresponding bending and torsional strain contributions are calculated. The combined minimum energy obtained of both contributions for a certain twist angle, gives the expected twist angle. In Figure 3.15 the torsional strain related energies are plotted against the twist angle, along with the expected twist angle. In Figure 3.16 the curvature in the pipeline is plotted for zero twist and for the twist angle related to the minima of the total potential energy of the model. The same is done for the horizontal displacement along the beam in Figure 3.17. Furthermore, the twist angle development is shown in Figure 3.18.



Figure 3.15: The total, bending and torsional strain energies for the pipeline. The minimum point is marked with a cross, and indicates the expected torsional angle ϕ_0 .



Figure 3.17: Geometric configuration of pipeline during lowering



Figure 3.16: The curvature along the pipeline before residual curvature addition (blue) and after residual curvature addition in its final state (red). Here the final state is achieved after a certain amount of torsion.



Figure 3.18: Development of torsion angle along pipeline

3.3.8. FINITE ELEMENT ANALYSIS USING ELASTIC BEAM ELEMENTS WITH RESIDUAL CURVATURE

Similar to subsection 2.8.2, the vertical suspended pipeline with residual curvature is modeled in the finite element program Abaqus for results comparison with the MATLAB models. The same type of beam elements, material properties and mesh density are used as in subsection 2.8.2. The model comprises several steps, each with their own boundaryand load conditions (see Table 3.2). These steps along with the aforementioned values will be discussed in the following. The actual results of the finite element analyses will be discussed thoroughly in chapter 5.

1. Initial conditions

In the primary step of the model, an initially curved beam is loaded into the 3D space, just as in subsection 2.8.2. At the end of the pipeline, an L - shaped rigid element is coupled to the pipeline. The element is a 2-node-3D rigid beam (*RB3D2*) of the linear geometric order[12]. The outer point of the L-shaped rigid, represents the center of gravity (COG) of an end terminal. The PLET PF-01-1A of the installed production flowline PF-01 is chosen to represent the end terminal. Specification of the end terminal can be viewed in Appendix E. PF-01 was the longest installed pipeline, and during the twist was measured several times. At the End terminal COG point P_e , loads will be activated in later steps. As for the coupling between the pipeline and the rigid element, this is a kinematic coupling which constrains all degrees of freedom of the rigid element in comparison to the movement of its reciprocal, the end point of the pipeline. Furthermore, the pipeline is constrained at both its ends (P_b and P_s), in all DOF. See Figure 3.19.

2. Application of gravity

The same instability issues during simulations in subsection 2.8.2 are present in this model, leading once more to the gradual build-up of certain loads and boundary conditions. In this step, gravity is applied to the whole pipeline M_T with a ramp function Figure 3.20. Here the same equivalent gravity constant after buoyancy compensation is used as in subsection 2.8.2. The pipe end at the surface P_s is clamped in at a given top angle α , and the lower pipe end P_b is constrained in al translational directions. In this step the whole pipeline M_T is constrained in the translational z-direction for stability purposes. For the higher curvature models during the sensitivity analysis, the final gravity applied to the

system was built up in two steps.

3. Lowering of pipeline

In the third step, the boundary condition on the whole modeled pipeline M_T and the lower pipe end P_b boundary conditions are removed completely. The pipeline is therefore solely constrained at the surface point P_s . The pipeline is lowered and ends up in a vertical suspended state (see Figure 3.21). During the lowering, a small, evenly distributed load perpendicular to the in-plane movement is applied along the pipeline (F_i). This load is applied to instigate the pipeline to twist in an energetically unstable configuration, while the magnitude of it remains of negligible influence to the final results. The same was done in subsection 2.8.2. There is also another load applied to the pipeline in the form of a line-load (F_c). The magnitude of the line load, along with its direction (either in head of beam direction) is dependent on the analysis being done. The line load represent possible current loading on the pipeline.

4. Current loading of end terminal

In the fourth step, the pipeline together with its end terminal, is in its suspended configuration (see Figure 3.22). As a last step, the end terminal is loaded with a point load acting on the center of gravity of the pipeline. This point load F_e represents the current loading and the weight of the end terminal. The direction and magnitude of the current component is again dependent on whether the analysis represents head or sideways current loading. The current contribution of the load is the summation of the total pressure over the projected surface area of the end terminal to the current.



Figure 3.21: Lowering of pipeline

Figure 3.22: Application of current loading

In Table 3.2 and overview is made of the boundary conditions and loads applied on the different instances of the model for the model steps. The table must be read similarly as Table 2.2 in subsection 2.8.2.



 $P_{s} = \text{Surface point of pipeline}$ $P_{b} = \text{Bottom point of pipeline}$ $M_{T} = \text{Total model}$ $P_{e} = \text{End terminal COG point}$ $g_{eq} = \text{Equivalent gravitational constant} \left[\frac{m}{s^{2}}\right]$ $F_{i} = \text{Instigating load } [N]$ $F_{c} = \text{Current load on pipeline } [N]$ $F_{e} = \text{Current load and weight of end terminal } [N]$

Figure 3.23: Global coordinate system that is used for the modeling of the pipeline in Abaqus using FE analysis.

Model step	Instance	В	oun	dary	r con	ditic	ons			Lo	ads				
		х	у	z	θ	ψ	φ	х	у	Z	θ	ψ	φ	Туре	Unit
1	P_s	•	•	•	•	•	•	-		-		-		-	
	P_b	•	•	•	•	•_	•							-	
	M_T	•	•	•	•	•	•	-	-	-	-	-	-	-	-
	P_e	-	-	-	-	-	-	-	-	-	-	-	-	-	-
2	P_s	•	•	•	•	•	•								
	P_b	•	•	•								-		-	
	M_T	-	-	•	-	-		-	•	-	-	-	-	g eq	$\left[\frac{m}{s^2}\right]$
	P_e	-	-	-	-	-	-	-	-	-	-	-	-	-	-
3	P_s	•	•	•	•	•	•					-			
	P_b		-	-	-	-			-	-	-	-	-	-	
	M_T	-	-	-	-	-	-	-	٠	-	-	-	-	g eq	$\left[\frac{m}{s^2}\right]$
		-	-	-	-	-	-	-	-	•	-	-	-	F_i	[N]
								•						F_{c}	[<i>N</i>]
	Pe	-	-	-	-	-	-	-	-	-	-	-	-	-	-
4	P_s	•	•	•	•	•									
	P_b	ļ <u>-</u> .													
	M_T	-	-	-	-	-	-	-	•	-	-	-	-	g _{eq}	$\left[\frac{m}{s^2}\right]$
		-	-	-	-	-	-	-	-	•	-	-	-	F_i	[N]
		ļ <u>-</u> .						•						F_c	[<i>N</i>]
	Pe	-	-	-	-	-	-	•	٠	-	-	-	-	F_e	[N]

Table 3.2: The boundary conditions and loads als defined per step on the different instances

4

RESULTS OF ANALYSES OF TWIST DEVELOPMENT DURING LAY OPERATIONS

In chapter 2 different analytical approaches were discussed to approximate the curvature of a pipeline during lay operations. The enhanced stiffened catenary method was chosen to use in further estimations of pipeline twist in a 4DOF/element environment. As a comparison, also a 6DOF/element finite element model was built to estimate the pipeline twist that would occur in a similar system. In this chapter, parameters that may influence the amount of pipeline twist will be varied in a sensitivity study and the results of both models will be compared with one another.

The parameters to vary in the sensitivity study were chosen based on the conducted literary review (Appendix C), the research into the lay operations with the Aegir (Appendix B) and on the analysis of the different contributions to the final twist in the analytical equations.

4.1. RESIDUAL CURVATURE DUE TO UNDER STRAIGHTENING

In this section, the amount of residual curvature that is assumed to be present in the pipeline is varied. According to the residual curvature criteria for reeled pipes set by HMC, a minimum residual curvature radius of 500m. In steps of 100m, the curvature radius is increased until the 1000m is reached: the DNV minimum straightness standard for factory pipes [25]. Also three relatively high residual curvature radii are examined, respectively one and a half, twice and thrice the allowed DNV standard for factory pipes. During the straightening step of the pipeline, the pipeline is either under straightened or over straightened to a certain extent. The direction in which the residual curvature in the pipeline is present, will be influential for the bending process that occurs in the sagbend part of the lay configuration. Under straightening of the pipeline will lead to a residual curvature that will increase bending strain in the sagbend, for the pipeline is bent against the strains induced by the residual curvature during laying with the Aegir (see Figure 4.1). Upon inspection of the on board reel-lay configuration and professional reports, it is less strenuous for the system to obtain an under straightened pipe within the tolerances than the other way around: over straightening leads to much higher loads on the straightener tracks, the pipeline and the aligner. Out of this efficiency standpoint, the pipelines are therefore usually under straightened. Ergo, lay operations with the Aegir are mostly executed with the direction of residual curvature which theoretically gives the highest amount of bending strains.



Figure 4.1: Schematic representation of under straightening after straightening operations. Also note that the lay direction of the DCV Aegir causes the under straightened pipeline to be bent against the residual curvature.



Torque angles in MATLAB due to under straightening, 2100m water depth

Figure 4.2: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Torque angles in Abaqus due to under straightening, 2100m water depth

Figure 4.3: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

		1 [0] 4]	-	Parameter	Value	Unit
$R_r[m]$	$\varphi_{max}[\circ]$ MATLAB	$\varphi_{max}[\circ]$ Abaqus	-	D	8 625	["]
500	128.4	163.8		D_0	0.025	L J ["]
600	128.4	159.8		ι_{st}	0.075	
700	127.0	166.6		t_c	2.5	["]
700	127.9	155.7		E_{st}	200	[GPa]
800	126.4	151.7		Ist	$6.74e^{-5}$	$[m^4]$
900	124.8	147.7		-31	7050	kg_1
1000	122.8	143.8		$ ho_{st}$	7850	$\left[\frac{1}{m^3}\right]$
1500	110.3	124.2		$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
2000	95.30	104.9		ρ_w	1025	$\left[\frac{kg}{m^3}\right]$
3000	59.70	59.66		α	85	[°]
				wd	2100	[m]

Table 4.1: Radii of residual curvature applied due to under straightening, along with the obtained maximum torsion angles in both MATLAB and Abaqus.

Table 4.2: Properties of pipeline and lay operation.

Looking at the results of the simulations, the following can be said. A lower residual curvature radius (higher κ_r) gives a higher maximum torsion angle, for both models. For the calculated angles with the 4DOF/element MATLAB model, the maximum torsion angle seems to converge to a certain value when decreasing the residual curvature radius. For the Abaqus model a more linear trend throughout is displayed. When comparing the absolute values of twist of the two models, one sees that a decreasing residual radius leads to a greater deviation in the maximum twist values between the two models. Where at $R_r = 3000m$ the calculated twist is practically the same, at $R_r = 500m$ the deviation of twist is 35°, or 21% difference from the Abaqus model. When looking at the shapes of the graphs, keep in mind that shape of the MATLAB torsion angle development graphs are estimated by a predetermined function. Upon comparison of the shapes between the two models, one sees that the Abaqus graphs have a more linear slope initially and appear to converge quicker at the end of the pipeline.

4.2. RESIDUAL CURVATURE DUE TO OVER STRAIGHTENING

The opposite to under straightening is over straightening. Here the residual strains will be in the same direction as those induced by the bending in the sagbend given the lay direction with the Aegir (see Figure 4.4). This will lead to reduced bending strains in the sagbend, along with a reduction of the overall bending energy of the lay operation. The results can be seen in Figure 4.5 and Figure 4.6.



Figure 4.4: Schematic representation of over straightening after straightening operations. Also note that the lay direction of the DCV Aegir causes the over straightened pipeline to be bent with the residual curvature.



Figure 4.5: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Figure 4.6: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

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4. RESULTS OF ANALYSES OF TWIST DEVELOPMENT DURING LAY OPERATIONS

$R_r[m]$	$\phi_{max}[^{\circ}] MATLAB$	ϕ_{max} [°] Abaqus
500	60.20	0.3208
600	46.14	0.3136
700	26.12	0.3066
800	0.000	0.2994
900	0.000	0.2928
1000	0.000	0.2864
1500	0.000	0.2584
2000	0.000	0.2353
3000	0.000	0.2001

Parameter	Value	Unit
D_o	8.625	["]
t_{st}	0.875	["]
t_c	2.5	["]
E_{st}	200	[GPa]
Ist	$6.74e^{-5}$	$[m^4]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
ρ_w	1025	$\left[\frac{kg}{m^3}\right]$
α	85	[°]
wd	2100	[m]

Table 4.3: Radii of residual curvature applied due to over straightening, along with the obtained maximum torsion angles in both MATLAB and Abaqus.

Table 4.4: Properties of pipeline and lay operation.

The bending of the pipeline in the same direction as the induced residual curvature gives some interesting results when comparing them to the previously obtained data in section 4.1. A significant reduction of the final rotation angle has taken place, to the extent that the twist in the Abaqus model can be assumed negligible. For the MATLAB simulations, one sees a comparatively large torsion angle for the three smallest residual curvature radii. Of the remaining radii, the results all converge to zero, which in comparison is very close to the Abaqus results. The deviating twist values for the smallest residual curvature radii can be explained by the way twist is calculated in the models. When the development of the system's total potential energy against the twist angle does not show a clear trough and is relatively constant for a range of twist angles, it has trouble finding a twist angle with minimal potential energy I. This can lead to a relatively large estimated angle, even though the energy difference between the calculated value and zero twist can be very small. Ergo, even for the maximum occurring twist angle in the MATLAB models (60 °), the energetic advantage compared to no twist is very low.

4.3. WATER DEPTH

Varying the water depth will impact the system in multiple ways. Deeper water will lead to a higher axial tension in the system for there is a larger amount of pipeline suspended from the clamping point. It also changes the curvature profile along the pipeline, as has been shown in section 2.6. The steeper departure angle leads to a more concentrated curvature peak close to the interaction with the seabed. The macro term of the torsional rigidity Q_{tr} as given in Equation 4.1 decreases with increasing water depth, for it is dependent on the pipeline length *L*, making a longer suspended pipeline more prone to twist. Therefore, the amount of torsion needed to achieve a certain twist angle will be higher for a pipeline in shallow water. As in section 2.6, water depths comparable to project site of Ichthys and Lucius will be used in the sensitivity study. The Aegir lay configuration for a top angle of 70 ° is schematically represented in Figure 4.7. The results can be seen in Figure 4.8 and Figure 4.9.

$$Q_{tr} = \frac{GJ}{L} \tag{4.1}$$



Figure 4.7: Schematic representation of the lay configuration of the DCV Aegir in shallow water. Note that the tests are done once more with under straightened pipelines.



Torque angles in MATLAB due to under straightening, 200m water depth

Figure 4.8: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Figure 4.9: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

$R_r[m]$	$\phi_{max}[^{\circ}]$ MATLAB	$\phi_{max}[^{\circ}]$ Abaqus
500	0.000	0.1730
600	0.000	0.1207
700	0.000	0.0927
800	0.000	0.0764
900	0.000	0.0644
1000	0.000	0.0554

Parameter	Value	Unit
D_o	8.625	["]
t _{st}	0.875	["]
t_c	2.5	["]
E_{st}	200	[GPa]
I_{st}	$6.74e^{-5}$	$[m^{4}]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
$ ho_w$	1025	$\left[\frac{kg}{m^3}\right]$
α	70	[°]
wd	200	[m]

Table 4.5: Radii of residual curvature applied due to under straightening, along with the obtained maximum torsion angles in both MATLAB and Abaqus.

Table 4.6: Properties of pipeline and lay operation.

The MATLAB models find no twist at all for the tested residual curvature radii and the Abaqus models give negligible amount of twist. Please note that in Abaqus an instigating load of 1N/m pipeline is exerted laterally. Although little, this disturbance in the system can give the small twist angles as observed in Figure 4.9. The obtained Abaqus twist values are therefore considered to be due to a model disturbance caused by the instigating load and not due to the residual curvature. Ergo, the twist angles with the Abaqus models are considered to be negligible and thus the same results are obtained with the MATLAB and Abaqus models upon model comparison. The reason why no twist is found at the shallow water depth, is explained with Figure 4.10 and Figure 4.11 below. Figure 4.10 represents the strain energy development for twist angles in deep water with the pipeline properties and configuration as given in Table 4.2 and Figure 4.11 represents the laying of a pipeline in shallow water with the exact same pipeline properties and the configuration as described by Table 4.6. Note that the given graphs are calculated by using total potential energy minimization with the analytical equations in MATLAB. Comparing the two graphs, a few things are noticed. In accordance with the macro term of torsional rigidity (Equation 4.1), it is seen that for shallow water the torsional strain energies are higher for a certain twist angle in comparison with the equivalent graph in deep water. Ergo, there is a higher resistance to twist for shallower water depth, and therefore for the given tests no twist occurs. The second thing that can be noted, is that the reduction in the overall bending energy strain through twisting is much more profound for deep water. Deep water results in long freely suspended pipeline lengths that contain residual curvature. This in turn give a large reduction of the bending strains if twist occurs. Both the lower resistance to torsion and the higher reduction gradient of the bending strain energy with respect to the twist angle result in twist occurring only in the deep water scenario and not in the shallow water. This signifies that for the residual curvature radii that are tested, pipeline twist is a deep water problem.



Figure 4.10: The development of the strain energy contributions with respect to the twist angle for a pipeline in deep water (wd=2100m).



Figure 4.11: The development of the strain energy contributions with respect to the twist angle for a pipeline in shallow water (wd=200m).

4.4. AXIAL TENSION

As discussed in the literary review in Appendix C, there are two important, opposing relations regarding pipeline tension dependency on rotation. The first being that increased tension results in a decreased curvature in the sagbend and therefore decreases the bending strains. In turn, this could lead to the reduction of twist because torsion will give a greater decrease in the bending strain energy. The second states that increased tension results in a lower torsional resistance for more pipeline is suspended, making the pipeline more prone to twist. During the sensitivity study, one hopes to obtain which of the discussed relations is more dominant during pipe lay operations. The tension in the pipeline is varied by changing the top angle of the tower: increasing the top angle leads to a decrease in the axial tension. For four different top angles, the twist development is investigated. The results can be seen in Figure 4.12 and Figure 4.12.





Figure 4.12: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Torque angles in Abaqus due to under straightening at different top angles, 2100m water depth

Figure 4.13: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

				Paramet	er Value	Unit
				$\overline{D_o}$	8.625	["]
α[°]	T[kN]	$\phi_{max}[^{\circ}]$ MATLAB	ϕ_{max} [°] Abaqus	t_{st}	0.875	["]
70	1771	110.0	140.4	t_c	2.5	["]
70	1//1	110.0	140.4	E_{st}	200	[GPa]
75	1572	120.8	141.6	Ist	$6.74e^{-5}$	$[m^4]$
80	1410	122.3	142.6	-31	7050	$r kg_1$
85	1276	122.8	143.8	ρ_{st}	7850	$\left[\frac{\overline{m}}{\overline{m}^3}\right]$
				$ ho_c$	800	$\left[\frac{\kappa g}{m^3}\right]$
Table 4.7	7: Top angle	es of lay tower during laying,	along with the ob-	$ ho_w$	1025	$\left[\frac{kg}{m^3}\right]$
tained n	naximum to	rsion angles in both MATLA	B and Abaqus. The	R_r	1000	[m]
residual	curvature is	s introduced due to under st	raigntening.	, ,	0100	

Table 4.8: Properties of pipeline and lay operation.

wd

2100

[m]

Looking at the results, it can be concluded that the axial tension in the pipeline does not have a significant effect on the development of pipeline twist. Both the MATLAB and Abaqus models show limited increase in the amount of twist while decreasing the tension in the system. The decrease in tension is achieved via increasing the top angle of the tower, which lead to a shorter suspended pipeline length. Upon model comparison, it can be concluded that both models show the same trend, only the twist value around which the graphs for different tension values hover, is different for the models.

4.5. BUOYANCY

The pipelines used for the Lucius project have a relatively thick coating of 2.5" GSPU, making the pipeline significantly more buoyant while the increase of the bending stiffness is negligible. Also having the pipeline non-flooded, increases the buoyancy. An increase of the buoyancy of the pipeline leads to a decrease of the submerged weight of the pipeline, which in turn decreases the axial tension and increases the suspended length of the pipeline. The two opposing relations as described in the previous section 4.4 also apply for buoyancy and once again it will be investigated which relation is more dominant. For the sensitivity study, analysis will be done with or without the GSPU coating to investigate the influence of buoyancy. The results can be seen in Figure 4.14 and Figure 4.15.



Figure 4.14: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Figure 4.15: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

				Parameter	Value	Unit
				OD	8.625	["]
				t _{st}	0.875	["]
Coating["]	$W_s[kg/s^2]$	$\phi_{max}[^{\circ}]$ MATLAB	ϕ_{max} [°] Abaqus	E_{st}	200	[GPa]
2.5	555.0	100.0	1/2 0	Ist	$6.74e^{-5}$	$[m^4]$
0	555.0 679.4	122.8	143.9	$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
				$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
Table 4.9: Top ang	les of lay tower d	luring laying, along with the	e ob-	ρ_w	1025	$\left[\frac{kg}{m^3}\right]$
tained maximum	torsion angles in	both MATLAB and Abaqus.	The	R_r	1000	[m]
Testudai cui vature	is introduced ut	ie to under straightening.		wd	2100	[m]
				α	85	[°]

Table 4.10: Properties of pipeline and lay operation.

It appears that buoyancy has no effect on the development of pipeline twist during laying. Again upon model comparison, it can be seen that both show the exact same indifference in twist development between a coated pipeline and a non-coated pipeline.

4.6. WALL THICKNESS

Increasing the wall thickness while maintaining the same outer diameter, increases the second moment of area and therefore the bending stiffness of the pipeline. Furthermore, increasing the wall thickness leads to an increase of the self weight and axial tension. In the sensitivity study, the outer diameter is kept constant while the thickness of the pipeline is increased. The results can be seen in Figure 4.16 and Figure 4.17.



Figure 4.16: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Figure 4.17: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

wt["]0.6

0.8	122.8	143.8
1	122.8	143.9
1.2	122.8	143.9

residual curvature is introduced due to under straightening.

 $\phi_{max}[^{\circ}]$ Abaqus

143.4

 $\phi_{max}[^{\circ}]$ MATLAB

122.3

Parameter	Value	Unit
OD	8.625	["]
t_c	2.5	["]
E_{st}	200	[GPa]
I _{st}	$6.74e^{-5}$	$[m^{4}]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
ρ_w	1025	$\left[\frac{kg}{m^3}\right]$
R_r	1000	[m]
wd	2100	[m]
α	85	[°]

Table 4.12: Properties of pipeline and lay operation.

Once more it seems that tested parameter does not effect the development of pipeline twist: for the wall thicknesses tested both the Abaqus and MATLAB twist developments along the pipeline show almost no change. Again, the mean value of the twist estimated with Abaqus is higher in comparison with the MATLAB model results. The trend of twist development for the different wall thicknesses however is identical for the models.

4.7. OUTER DIAMETER

As mentioned in section B.3 in Appendix C, the range of pipe diameter that can be reeled onto the drums is 6" to 16". Increasing the diameter of the pipeline will result in an increase of the second moment of area and consequently the bending stiffness. When dealing with a pipeline with a relatively thick buoyant coating, it is not directly clear if increasing the diameter will also increase the submerged weight of the pipeline, along with the tension. The 2.5" GSPU coating is maintained while varying the outer diameter of the pipeline. The wall thickness of the pipeline is also kept constant. The results can be seen in Figure 4.18 and Figure 4.19.



Figure 4.18: Array of graphs concerning the twist development along the pipeline as calculated with MATLAB



Figure 4.19: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus
				_	Parameter	Value	Unit
OD["]	$W_s[kg/s^2]$	ϕ_{max} [°] MATLAB	$\phi_{max}[^{\circ}]$ Abaqus	-	t _{st}	0.875	["]
6	421.4	123.3	143.7		t_c	2.5	["]
8.625	555.0	122.8	143.8		E_{st}	200	[GPa]
10	597.0	122.3	143.7		I _{st}	$6.74e^{-5}$	$[m^4]$
10	623.5	122.3	143.7		$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
14	609.3	121.8	143.6		$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
16	554.4	121.3	143.5		$ ho_w$	1025	$\left[\frac{kg}{m^3}\right]$
					R_r	1000	[m]
Table 4.13:	Top angles of la	ay tower during laying, alor	ng with the		wd	2100	[m]
obtained m The residua	naximum torsion al curvature is int	n angles in both MATLAB at troduced due to under straig		α	85	[°]	

Table 4.14: Properties of pipeline and lay operation.

The outer diameter seems to be of negligible effect on the development of twist: it is difficult to spot the change in twist when varying the outer diameter in the result graphs. Again, the MATLAB and Abaqus models show the exact same trend regarding twist development.

4.8. Clarification difference in twist development during laying between MATLAB and Abaqus models

Upon result comparison of the sensitivity analysis between the models, the following can be concluded. For certain parameter studies, such as those done for the outer pipeline diameter, the wall thickness, the axial tension and the buoyancy, the results show the exact same trend in both models. However, the maximum values of the torsion angle are different. This difference is best seen in the tests done for the different radii of residual curvature formed by under-straightening. Decreasing the curvature radius from 3000m to 500m, lead to an increase in the difference of the torsion angle approximation between the two models. An explanation for the discrepancy was sought after by looking at the difference between the two models. A quintessential difference between the models is that the MATLAB models contain 4DOF elements and those in Abaqus contain 6DOF. The abaqus model therefore has an additional bending plane in which the pipeline can have a curvature development. This out-of-plane bending is schematically illustrated in Figure 4.20. It was investigated what the influence is of out-of-plane bending of the pipeline during lay operations, a contribution that cannot be taken into account in the 4DOF/element MATLAB models but may well play a role in the development of twist.



Figure 4.20: The in-plane and out-of-plane bending of the pipeline in a 3D space

4.8.1. CONTRIBUTION OF OUT-OF-PLANE BENDING TO THE PIPELINE CURVATURE

Out-of-plane bending is only possible in the 6DOF/element Abaqus models, and therefore a closer look will be taken to the Abaqus models for an explanation of the offset in results between MATLAB and Abaqus. To obtain a better understanding of the out-of-plane bending contribution, the in- and out-of-plane curvatures are examined of the Abaqus tests with residual curvature due to under straightening. The same pipe properties are used as given in Table 4.2. The

curvatures are plotted for R_r = 500m, the radius which shows the biggest difference in twist angle approximation between the models, and $R_r = 2000$ m, which shows far less difference.



Figure 4.21: The in-plane and out-of-plane curvature development of the pipeline in Abaqus for R_r = 500m. The pipeline is formed due to under straightening and the water depth is 2100m.

When comparing the in-plane and out-of-plane curvatures with one another, one sees that the amount of out-of-plane bending curvature is quite significant. Along approximately the first 700m it is apparent that a reduction of the in-plane curvature simultaneously occurs with the negative increase of the out-of-plane curvature. Afterwards the out-of plane curvature goes from negative to positive values and takes on a similar shape to the in-plane curvature upon approach of the seabed interaction. The peak of the out-of-plane curvature is around two-thirds (\approx 67%) of the absolute value of the in-plane curvature peak.



Curvature in-plane and out-of-plane in Abaqus for R_r = 2000m due to under straightening

Figure 4.22: The in-plane and out-of-plane curvature development of the pipeline in Abaqus for $R_r = 2000m$. The pipeline is formed due to under straightening and the water depth is 2100m.

For the curvature graphs with a higher residual curvature radius, the development is less wavy. Before the curvature peaks near the seabed, the in-plane curvature slightly decreases as the out-of-plane curvature negatively increases. The ratio between the peaks is lower than in the previous Figure 4.21: the out-of-plane peak is around 40% of the in-plane peak. Reduction of the residual curvature in the system appears to lead to a reduction of the out-of-plane curvature. In the next subsection its effect on the total bending work will be discussed.

4.8.2. CONTRIBUTION OF OUT-OF-PLANE BENDING TO TOTAL BENDING WORK

The strain energies of the curvatures can be computed by using the bending strain equation as given by Equation 2.51. The work of the in-plane, out-of-plane and the summation of the two are plotted for the residual curvature radii R_r = 500m and $R_r = 2000m$. Again the same pipe properties as described by Table 4.2 are used.



Figure 4.23: The work due to in-plane and out-of-plane bending for $R_r = 500m$ in Abaqus. The pipeline is formed due to under straightening and the water depth is 2100m.

The work of the out-of-plane bending proves to be significant for $R_r = 500$ m. If the work of the plastic deformations, occurring in the first meters of the pipeline due to the model constraints, is subtracted from the in-plane bending work, the out-of-plane work along the pipeline is actually higher. This indicates that in the total energy balance of the pipeline, the contribution of the out-of-plane bending strain energies should be taken into account when estimating the twist angle via potential energy minimization of the system.



Work in-plane and out-of-plane in Abaqus for R_r = 2000m due to under straightening

Figure 4.24: The work due to in-plane and out-of-plane bending for $R_r = 2000m$ in Abaqus. The pipeline is formed due to under straightening and the water depth is 2100m.

The same graphs were made for the higher residual curvature radius of 2000m. Here we see a totally different development of the out-of-plane work along the pipeline (Figure 4.24). Ignoring the sharp increase of the total work in the beginning due to plastic deformations caused by the model boundary constraints, the in-plane bending work increases less in total than the in-plane bending work at the lower residual curvature radius. The out-of plane work is in comparison a lot less than the in-plane contribution, and therefore the overall bending work is not so different than the work done by in-plane bending only. It appears that for higher residual curvature radii, the contribution of the out-ofplane bending to the overall bending energy becomes less dominating. The approximation of the bending work in a 4DOF/element system which only takes in-plane bending into account, therefore becomes more accurate at higher R_r .

4.8.3. CONTRIBUTION OF OUT-OF-PLANE BENDING TO PIPELINE TWIST

One has seen that the increase of the amount of residual curvature in the system (by decreasing the residual curvature radius) leads to an increase of the out-of-plane bending work. For the 4DOF/element MATLAB models, the out-of-plane motion of the pipeline is not taken into account, an essential difference with the 6DOF/element Abaqus Models. The potential energy balance therefore has an extra energy contribution in the 6DOF/element model: the bending strain energy due to out-of-plane bending. Lacking this term in 4DOF/element, upon minimization of the total energy of the system the torsion angle is underestimated. The significance of this term increases with a decreasing residual curvature radius, giving more and more offset between the obtained twist angles of the models. The derivation of the twist angle using energy minimization of the R_r = 500m under straightening MATLAB model with the pipe properties of Table 4.2



is given in Figure 4.25. The value of the twist angle obtained in the corresponding Abaqus model is also plotted.

Figure 4.25: The total contributions of the in plane bending, torsion and the sum against the torsion angle.

At the moment one can only speculate on the development of the out-of-plane bending energy graph against the twist angle. If one looks at the development of the in-plane bending strain energy versus the twist angle, one can see that a quadratic cosine term is present (see Equation 2.57). This quadratic cosine term leads to the reduction of the residual curvature along the pipeline through twisting. The development of the bending strain due to this quadratic term dependent on the twist angle is clearly visible in Figure 4.25. If one assumes that the reduction of the in-plane bending energy leads to the increase of the out-of-plane energy in a complementary way, the out-of-plane bending energy would then be represented by a function which also has a quadratic cosine term. A phase difference of $\pi/2$ in the quadratic cosine term of the out-of-plane bending strain in comparison to the one of the in-plane can be expected: the given phase difference results in a shift of the quadratic cosine term so that the out-of-plane strain energy starts increasing at $\phi = 0$. An estimation of the out-of-plane strain energy development against the twist angle is given in Figure 4.26 by the green colored graph. Adding the contribution of the out-of-plane bending gives a shift of the total energy minima as calculated by MATLAB to the right: the minimum total potential energy would then be reached at a higher rotation angle. The 6DOF/element Abaqus models do take the out-of-plane bending strain energy into account, and therefore its development of the total potential energy is different than for the 4DOF/element MATLAB models. Ergo, subtracting the values of U_{bop} (green) from the Π_A (blue dotted) will give us the values of Π_M (blue solid). A schematic representation tation of the total potential energy development versus the pipeline twist in Abaqus is given by the blue dotted line in Figure 4.26.



Figure 4.26: The total contributions of the in plane bending, torsion and the sum against the torsion angle.

5

RESULTS OF ANALYSES OF TWIST DEVELOPMENT DURING PIPELINE LOWERING

Similar to the sensitivity analysis done for pipeline twist during lay operations, the amount of twist during pipeline lowering is estimated for different amounts of residual curvature in the pipeline. Again the 4DOF/element MATLAB models for the vertical pipeline will be compared with equivalent Finite Element 6DOF/element Abaqus models. The properties of an installed pipeline at the Lucius project with corresponding end terminal are given as input for both the models. A restriction which must be taken into account when comparing the different models, is that the MATLAB model can only be subjected to currents in the in-plane direction. A current with a out-of-plane component can only be modeled with the Abaqus models, for in MATLAB the pipeline can only be subjected to loads in 2D space. Therefore, certain current related sensitivity studies will be performed solely with the Abaqus models. All the tests have been performed for a pipeline length of 2100m, which is approximately the maximum length that the pipeline in vertical suspended state can achieve before seabed interaction given the conditions encountered at the Lucius project. The maximum pipeline length has been chosen because the vertical pipeline's torsional resistance decreases with increasing length. Ergo, twist development as well as the effect of sensitivity studies will be most visible at the maximum vertical pipeline length.

5.1. RESIDUAL CURVATURE

In Figure 5.2 and Figure 5.3 the amount of twist is estimated for different residual curvature radii. The residual curvature is applied as under straightened curvature, and the current acts in the positive in-plane direction (current to bow), against the residual curvature introduced in the pipeline as is depicted in Figure 5.1.



Figure 5.1: Schematic representation of pipeline lowering with the DCV Aegir. Note that the tests are done once more with under straightened pipelines and a constant current acting in positive x-direction.



Figure 5.2: The twist development along the pipeline for different residual curvature radii as calculated with MATLAB for a vertical suspended pipeline with end terminal and positive in plane current loading. The pipeline is formed due to under straightening and the water depth is 2100m. loading.



Figure 5.3: The twist development along the pipeline for different residual curvature radii as calculated with Abaqus for a vertical suspended pipeline with end terminal and positive in plane current loading. The pipeline is formed due to under straightening and the water depth is 2100m.

			Parameter	Value	Unit
			D_o	8.625	["]
			t_{st}	0.875	["]
		([0] 4]	t_c	2.5	["]
$R_r[m]$	$\phi_{max}[\circ] MATLAB$	$\phi_{max}[\circ]$ Abaqus	E_{st}	200	[GPa]
500	95.80	0.4078	Ist	$6.74e^{-5}$	$[m^{4}]$
750	82.73	0.2776	$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
1000	66.69	0.2089	$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
able 5.1: Ra	adii of residual curvature app	blied due to under straight-	$ ho_w$	1025	$\left[\frac{kg}{m^3}\right]$
ning, alon	g with the obtained maxim	um torsion angles in both	α	90	[°]
IATLAB an	d Abaqus.		L	2100	[m]

 $egin{array}{cccc} lpha & 90 & [^{o}] \ L & 2100 & [m] \
u_{H} & -0.1 & [rac{m}{s}] \
W_{PLET} & 28631 & [mT] \end{array}$

Table 5.2: Properties of pipeline during lowering operation.

Looking at the results, one can see that there is a large difference in results between the models. Where almost no twist occurs with Abaqus, the MATLAB gives significant twist for all residual curvature radii. The large discrepancy in the results lead to the abridgement of the sensitivity analysis: only three residual curvature radii were compared. While the absolute values differ greatly, one does see the same trend with decreasing residual curvature radius: the amount of pipeline twist increases, similarly as for the pipeline during lay operations.

5.2. CURRENT TO BOW LOADING

In this section the effect of current on the torsion angle development along the vertical suspended pipeline is investigated. As mentioned before in the introduction of chapter 5, for model comparison only in plane current loading can be used. In the following Figure 5.4 till Figure 5.8 the models are compared for different current speeds in the positive direction (current to bow) against the under-straightened curvature of the pipeline as well in negative direction (current to stern). Besides from the current loading, all pipeline and model properties are equal to the ones given in Table 5.1 in the previous section. The results of different current magnitudes acting in positive direction similar as depicted in Figure 5.1, are given in Figure 5.4 and Figure 5.5.



Figure 5.4: The twist development along the pipeline as calculated with MATLAB for a vertical suspended pipeline with end terminal for different positive in-plane current speeds. Here, the current loading is against the direction of the residual curvature in the pipeline.



Figure 5.5: The twist development along the pipeline as calculated with Abaqus for a vertical suspended pipeline with end terminal for different positive in-plane current speeds. Here, the current loading is against the direction of the residual curvature in the pipeline.

$R_r[m]$	$v_c[\frac{m}{s}]$	$\phi_{max}[^{\circ}]$ MATLAB	ϕ_{max} [°] Abaqus
1000	0	95.77	0.4029
1000	0.1	95.62	0.4078
1000	0.2	94.76	0.4183

Table 5.3: The obtained maximum torsion angles in both MATLAB and Abaqus for different positive in-plane current speeds. Here, the current loading is against the direction of the residual curvature in the pipeline.

The results of the negative current (current to stern) sensitivity study are given in Figure 5.7 and Figure 5.8. A schematic representation of the current direction with respect to the pipeline lowering is given in Figure 5.6.



Figure 5.6: Schematic representation of pipeline lowering with the DCV Aegir. Note that the tests are done once more with under straightened pipelines and a constant current acting in negative x-direction.



Figure 5.7: The twist development along the pipeline as calculated with MATLAB for a vertical suspended pipeline with end terminal for different negative in-plane current speeds acting in same direction of residual curvature in the pipeline.



Figure 5.8: The twist development along the pipeline as calculated with Abaqus for a vertical suspended pipeline with end terminal for different negative in-plane current speeds acting in same direction of residual curvature in the pipeline.

$R_r[m]$	$v_c[\frac{m}{s}]$	$\phi_{max}[^{\circ}] MATLAB$	ϕ_{max} [°] Abaqus
1000	0	95.80	0.4029
1000	-0.1	82.73	0.2776
1000	-0.2	66.69	0.2089

Table 5.4: The obtained maximum torsion angles in both MATLAB and Abaqus for different in-plane current speeds. Here, the current loading is in the same direction as the residual curvature in the pipeline.

For both the current loading against as in the same direction as the residual curvature in the pipeline, one sees a large discrepancy between the models. While the Abaqus model predicts almost no pipeline twist in all the load cases, the MATLAB models predict a large twist angle. The effect of the current speed or its direction in-plane seem to be of little influence for both models: only minor difference at the current loadings are found. Again the sensitivity analysis is abridged due to the large discrepancy in the results and the minor influence of the current in the models.

5.3. TRANSVERSE CURRENT LOADING

The effect of out-of-plane current loading can only be taken into account in the 3D Abaqus model. For different current velocities, the pipeline and end terminal are subjected to a positive sideways current loading (transverse current), perpendicular to the previous in-plane loading. Whether the out-of-plane current loading is positive or negative does not make a difference in the tests, due to the models symmetry. The results can be seen in Figure 5.9.



Figure 5.9: Array of graphs concerning the twist development along the pipeline as calculated with Abaqus

$R_r[m]$	$v_c[\frac{m}{s}]$	ϕ_{max} [°] Abaqus
1000	0	0.4029
1000	0.1	0.5080
1000	0.2	0.8340

Table 5.5: The obtained maximum torsion angles in Abaqus for different out-of-plane current speeds. Here, the current loading is positive and perpendicular to the in-plane

Once more it can be seen that the amount of twist in the pipeline is quite small. With still less than a degree of twist for all current speeds tested, a more elaborate out-of-plane current analysis was canceled. It was chosen to further investigate the reason of the offset in results between the models, which will be discussed in the next chapter.

5.4. Clarification difference in twist development during lowering between MATLAB & Abaqus models

During the sensitivity analysis of the twist development during lowering (see chapter 5), it became apparent that there is a large discrepancy in results between the MATLAB models and the Abaqus models. The MATLAB models show significant twist development in the order 60 to 100°[4], while the maximum twist for the Abaqus models did not exceed the 1°. An explanation was sought after to explain this large difference between the models, and to determine which gave the most accurate result in comparison to real lowering operations.

5.4.1. IN-PLANE CURVATURE COMPARISON BETWEEN THE VERTICAL LOWERING ABAQUS AND MAT-LAB MODELS

A closer look is taken to the development of the curvature along the pipeline for both models. One compares the models for the pipeline and loading properties as described in Table 3.1.3. For the Abaqus models, there was virtually no torsion along the pipeline. Therefore the residual curvature in the pipeline remained almost completely in the in-plane bending plane as can be seen in Figure 5.10. For the MATLAB models, the residual curvature in the in-plane bending plane is made dependent on the twist angle. Iteratively the bending and torsional strain energy in the system are calculated for different twist angles, and by minimizing the total energy of the system an expected twist angle is obtained. Twist leads to the reduction of the residual curvature in the in-plane bending plane, as can be seen in Figure 5.11. For a twist angle of 90° the residual curvature is decreased to zero along the pipeline, and for a twist angle of 180° the residual curvature contribution goes from $+\kappa_r$ to $-\kappa_r$. When comparing Figure 5.10 with Figure 5.11, one can see that for zero twist the curvature development along the pipeline is virtually constant. Ergo, the analytical equations used in the MATLAB do accurately represent a vertical suspended beam with residual curvature when no pipeline twisting occurs.



Figure 5.10: The curvature along the pipeline for a twist angle which is almost equal to zero.



Figure 5.11: The curvature along the pipeline for different twist angles as obtained in the MATLAB models.

5.4.2. Effect out-of-plane bending contribution on discrepancy results Abaqus and MAT-LAB models

The culprit is found in the lack of out-of-plane bending contribution in the MATLAB 4DOF/element models. By twisting, the residual curvature in the pipeline is reduced in the MATLAB models. In reality the decrease of the in-plane bending curvature by twisting would lead to the increase of the curvature in the out-of plane direction. However, the latter increase is not taken into account in the MATLAB models. Therefore twist is energetically perceived as more beneficial than it would be in reality. When disregarding the residual curvature in the pipeline, the in- and out-ofplane curvature along the pipeline is very similar, even with head or sideways current loading: due to the long pipeline length, the submerged weight of the pipeline and that of the PLET, the pipeline stays incredibly straight. This means that pipeline twist would energetically give no benefit. Ergo, twist would lead to the transfer of residual curvature from the in-plane to the out-of-plane of the system, while the torsional strain increases. Upon inspection of the Abaqus model, it is seen that the small instigating load F_i of 1N mostly leads to the bending of the beam in the out-of-plane direction and only a small amount of twist. A schematic overview of the difference between the models can be seen in Figure 5.12.



Figure 5.12: Schematic overview of difference between 3D Abaqus vertical pipeline models and 2.5D MATLAB models.

6

PIPELINE TWIST MODELS WITH PRE-CURVED BEAMS FORMED BY PLASTIC DEFORMATION

From the sensitivity analyses performed on the beam models during lay operations in chapter 4, it was seen that residual curvature causes pipeline twist in amounts representative to the values found during operations at the Lucius project. It is apparent that bending an under straight pipeline in the opposite direction in the sagbend (as is the case with laying with the Aegir) has the most profound influence on the twist. In the subsequent chapter 5, the onset of twist during lowering was investigated. Here the amount of twist calculated differed greatly between the models, and after further research it was concluded that the MATLAB models were insufficient given its dimensional simplification. Since the MATLAB models contain 4DOF elements, the increase of out-of-plane curvature was not taken into account when decreasing the in-plane residual curvature due to torsion. It was decided to continue further research with the Abaqus models. The amount of twist that occurred with pre-curved beams representing pipelines with residual curvature during lowering operations was negligible however. Even with static loading onto the pipeline with end terminal due to current form different directions and velocities, the amount of twist never surmounted the 1°. During lowering operations with the Aegir, the maximum twist that occurred was in the order of 90 degrees, far more than what was found during the models tests. It was therefore decided to pursue the investigation of other possible factors that may influence the onset of pipeline twist, particularly during lowering operations. In this chapter the influence of plastic deformation will be investigated. For the Abaqus models, all of them contained pre-curved beams shaped directly into the residual curvature radius that is to be tested. The beams in the initial phase, before the application of loads or gravity, contained no stresses or strains whatsoever to keep them in their shape. During the tests, all strains were kept elastic (with the exception of the first few meters at the clamping point which caused local plastic deformation by the boundary conditions itself). These models do not take into account the stress-strain history of the pipeline during reel-lay operations. As was seen in the literature study, the double bend cycle during reel-lay operations causes significant plastic deformations before the pipeline enters the moonpool. To investigate the effect of prior plastic deformation, the onset of twist for a model with an plastically pre-curved beam and one with an elastically pre-curved beam with the same initial residual curvature radius are compared. The method of obtaining the pre-curved beams with plastic deformation is however not the same as the double bend cycle occurring during reel-lay operations with the Aegir. It is therefore deemed as a qualitative study, where one seeks to see if plastic deformation increases the maximum twist angle.

6.1. FINITE ELEMENT ANALYSIS USING PLASTICALLY DEFORMED BEAM ELEMENTS WITH RESIDUAL CURVATURE

To obtain a beam with residual strains that retains a curved shape in no load conditions, the beam must be plastically deformed. Given the large computational time associated with large plastic deformation for long beam models, it was chosen to only bend the pipeline plastically once. Also the amount of residual strain introduced to the system is not equivalent to those induced during the actual reeling on process. In the Abaqus model, a reel of 39m is used to bend the pipeline plastically, in comparison to 8m of the reel-drum used during spooling on for the Aegir. The reasons for this increased reel radius are twofold. First of all, in order to only model one time plastic bending, the reel radius cannot be too large in order to avoid excessive residual curvature after un spooling: one does not want to bend the beam for a second time to obtain a residual curvature radius which is in the limits of acceptability for HMC: this would lead to more model complexity, convergence issues and last but not least a higher computational time. A reel radius of 8m will give a residual curvature radius without straightening that is far lower that the 500m standard of HMC. The second reason is computation time: larger plastic deformations along the pipeline length will lead to more iterative steps for Abaqus to converge to an answer and thus gives a higher computational time. The steps undertaken, along with the model properties will be given in the following subsections.

6.1.1. ELEMENTS

The pipeline is modeled using Abaqus PIPE31 beam elements[12], similarly to all the previous Abaqus models.

6.1.2. MESH

For a mesh density of 1 element per m, as is used in all the other models, the pastic deformation model is unstable. A mesh convergence study lead to a finer mesh of 5 elements per m over the entire length of the pipeline.

6.1.3. MATERIAL PROPERTIES

For the material of the beam, a bilinear plastic deformation model representing X-65 steel is used. Here the yield stress is set to be 450 MPa, and at 535 MPa a plastic strain of 0.1 occurs. A piecewise linear representation or other more advanced material models would give a more accurate representation of plastically deforming X-65 steel, but since we are dealing with a qualitative analysis a material model with linear strain hardening will suffice. Furthermore, the material is assigned a density, Young's Modulus and Poisson's ration equivalent to the prior models.

6.1.4. INTERACTIONS

In the model two different interactions take place: one between the beam and a circular 3D analytically rigid element representing a spool, and one between the beam and a 3D analytically rigid flat surface representing the seabed. Both rigid elements cannot deform during interaction with the pipeline and the interactions are frictionless. Also, all the interactions are defined as hard contacts [12].

6.1.5. MODEL STEPS

In a series of steps one obtains the desired beam shaped to a certain residual curvature due to residual strains in the beam. Once obtained, further steps of bringing the pipeline into its configuration during lay operations will be exactly the same. Therefore, only the steps up to the initial phase will be discussed, the phase where in comparison with the elastic models the pre-curved beam was loaded into the model. The steps will be discussed briefly in the following, along with the loads and boundary conditions.

1. Load in parts in assembly

The pipeline, rigid spool and rigid seabed are loaded into the model. The bottom point of the pipeline P_b is constrained in all directions, and the rest of the pipeline until the top point is only constrained in z-direction (see Figure 3.23). The reel is constrained in all directions via a rigid coupling to the center point of the reel P_r , which is fully constrained. The seabed is also constrained completely. The top point of the pipeline contains a kinematic coupling with the center point of the reel. Here, the top point is constrained in all directions except rotation around the z axis in comparison to the movement of its coupled center point.

2. Reeling on

During reeling on, the pipeline is spooled onto the spool. The center point of the reel P_r rotates around the z-axis and winds up the pipeline by plastically deforming it over the reel. The y-constraint of the bottom point P_b is released to facilitate the movement of the pipeline. A point load T_b in negative z direction is active at the bottom point of the pipeline. This load represents the back tension. The back tension was calculated as given in Equation 6.1. Here, 1.3 is the safety factor used by HMC for back tension calculation. The back tension is kept constant during spooling on.

$$M_p = \sigma_y \cdot t_{st} \cdot (D_o - t_{st})^2$$

$$T_b = 1.3 \cdot \frac{M_p}{R_{sp}}$$
(6.1)

3. Reeling off

At the end of spooling on, most of the pipeline is wound up plastically on the spool. Small end sections of the pipeline close to the ends have not been plastically deformed, but will be removed in a later step to achieve a pipeline with a uniformly distributed residual curvature over its entire length. In the reel off phase, the center point of the reel P_r starts to rotate in the counter-clockwise z-axis, leading to the unspooling of the pipeline. The rotation around the z-axis constraint is removed from the bottom point P_b . This is done to avoid plastic back bending of the pipeline once unspooling begins. Once again, the back tension T_b is maintained.

4. Pipeline clamping and tension relieve

In the next couple of steps, loads and boundary conditions are gradually removed to obtain a pipeline shaped in its residual curvature radius. To avoid convergence issues, this cannot be done in solely one step. As mentioned before, small segments close to the pipe ends where not plastically deformed. In this step, a clamping point P_{cl} is defined on the pipeline, the point where the pipeline would be clamped in by the tensioner. This clamping point is constrained in all directions and is situated below the area which was not plastically deformed. Also the tension is removed from the pipeline in this step.

5. Introduction of a small gravitational field

At the end of the previous step, the y-constraint of the bottom point P_b is released, while the constraints in x and z still remain active. A gravitational field is introduced to the model. This gravitational field has a low equivalent gravity constant and is solely introduced to maintain stability to the pipeline when removing the y-constraint of the bottom point.

6. Release of x-constraint bottom point

In this step the sole change that is implemented is the removal of the x-constraint at the bottom point P_b of the pipeline. The bottom point is now only constrained in z-direction.

7. Removal gravitational field

The small gravitational field is removed and the pipeline gently moves to its final curved position.

8. Trimming of beam and obtaining final curved shape

As mentioned before, the ends of the pipeline close to the boundary conditions were not properly deformed plastically. At the top of the pipeline, a new clamping point P_{cl} below the elastic zone was formed in the pipeline clamping and pipeline displacement step. For the end of the pipeline, a different method is used to get rid of the elastic region: the elements in the region are removed from the beam by trimming. At the end of the trim, a point P_b^* is defined and is constrained fully. Once the step has been completed, one has a uniform plastically deformed beam in the shape defined by its residual curvature radius. This end state is equal to the initial shape in the elastic pipeline models, where an elastic beam is loaded into the model. Only now, residual stresses are present.



Figure 6.4: Pipeline clamping and tension removal.

0

Figure 6.5: Introduction of small gravitational field.

Figure 6.6: Release of x-constraint bottom point.



Figure 6.8: Trimming of beam and obtaining the final curved shape.

0

In Table 6.1 and overview is made of the boundary conditions and loads applied on the different instances of the model for the model steps. The table must be read similarly as Table 2.2 in subsection 2.8.2. $\dot{P_b}$ = Bottom point of pipeline



 P_r = The reel's center point P_{cl} = Clamping point on the pipeline P_{b}^{*} = New bottom point of pipeline after trimming M_T = Total model $g_s =$ Small gravitational constant $\left| \frac{m}{s^2} \right|$ F_i = Instigating load [N] T_b = Back tension [N] θ_s = Rotation of center point spool during spooling on [*rad*]

Figure 6.9: Global coordinate system that is used for the modeling of the pipeline in Abaqus using FE analysis.

- θ_u = Rotation of center point spool during unspooling [*rad*] b_x = Displacement of bottom point in x-direction [m]
- b_{y} = Displacement of bottom point in y-direction [m]

Table 6.1: The boundary conditions and loads als defined per step on the different instances

Model step	Instance	E	Bound	lary	cond	ditior	15			Lo	ads				
		х	у	Z	θ	ψ	ϕ	x	у	Z	θ	ψ	ϕ	Туре	Unit
1	P_b	•	•	•	•	•	•				-		-		-
	M_T	•	•	•	•	•	•								
	P_r	•	•	•	•	•	•				-				
	P_{cl}	-	-	-	-	-	-	-	-	-	-	-	-	-	-
2	P_b	- <u>-</u> -		•	•									T_b	[<u>N</u>]
	M_T	L		•											
	P_r	•		•	•	•	θ_s								
	P _{cl}	-	-	-	-	-	-	-	-	-	-	-	-	-	-
3	P_b	•		•	•	•			•					T_b	[<i>N</i>]
	M_T	L		•							-				
	P_r	•	•	•	•	•	θ_u				-	-	-		-
	P_{cl}	-	-	-	-	-	-	-	-	-	-	-	-	-	-
4	P_b	b_x	b_y	•	-										-
	M_T			•	-								-	-	
	P_r	•	•	•	•	•	•	-	-	-	-	-	-	-	-
	P_{cl}	•	•	•	•	•	•	-	-	-	-		-		-
5	P_b	•		•											
	M_T	-	-	•	-	-	-	-	•	-	-	-	-	gs	$\left[\frac{m}{s^2}\right]$
	P_r	•	•	•	•	•	•	-	-	-	-		-	-	-
	P_{cl}	•	•	•	•	•	•	-	-	-	-		-		-
6	P_b			•		-					-			-	
	M_T	-	-	•	-	-	-	-	•	-	-	-	-	gs	$\left[\frac{m}{s^2}\right]$
	P_r	•	•	•	•	•	•	-	-	-	-	-	-	-	-
	P_{cl}	•	•	•	•	•	•	-							-
7	P_b			•	-	-			-		-	-	-	-	-
	M_T	-	-	•	-	-	-	-	-	-	-	-	-	-	-
	P_r	•	•	•	•	•	•	-	-	-	-				-
	P_{cl}	•	•	•	•	•	•								
8	P_b*	•	٠	٠	•	•	•	-	-	-	-	-	-	-	-
	M_T	•	•	•	•	•	•	-	-	-	-	-	-	-	-
	P_r	•	•	•	•	•	•	-							
	P_{cl}	•	•	•	•	•	•	-							-

6.2. RESTART FOR PIPELINE DURING LAYING MODEL

The data from the final step of the plastic deformation model as described in section 6.1 is used as starting point for a model similar to the previously used elastic models for simulating the pipelay process as described in subsection 2.8.2. This process is known as a restart in Abaqus. A restart is done to obtain the final curved shape obtained with the plastic deformation model. From here on out the steps to obtain the pipeline in its lay configuration are identical to those used in the elastic pipeline during laying model. These steps will not be discussed, and can be read in subsection 3.3.8.

6.3. Comparison pipeline during laying model containing plastically precurved beam versus elastic pre-curved beam

A pre-curved beam with residual plastic strains is formed as described in section 6.1. A similar model is made for an elastically pre-curved pipeline containing the same residual curvature radius. The sole difference between the two is the history before the initial phase: for the plastic model, reeling the pipeline over a spool has lead to the residual stress introduction of permanent plastic strains which keep the pipeline in the desired radius. For the elastic model, the beam has been loaded into Abaqus into the desired radius directly and therefore does not contain any strains and stresses initially. From the initial phase onwards the model steps, loads and boundary conditions are identical for the two models. The results of the final amount of twist can be seen in Figure 6.10.



Figure 6.10: The twist development along the pipeline for a 721.3m residual curvature radius as calculated with Abaqus elastically pre-curved beams and plastically deformed pre-curved beams.

$R_r[m]$	$\phi_{max,e}[^{\circ}]$	$\phi_{max,p}[^{\circ}]$	Δ [%]
721.3	154.9	155.2	0.1937

Table 6.2: The maximum twist angles of an elastic pre-curved Abaqus beam model versus a plasticly deformed pre-curved Abaqus beam model.

Parameter	Value	Unit
Do	8.625	["]
t_{st}	0.875	["]
t_c	2.5	["]
E_{st}	200	[GPa]
Ist	$6.74e^{-5}$	$[m^{4}]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
ρ_w	1025	$\left[\frac{kg}{m^3}\right]$
α	90	[°]
L	2302	[m]

Table 6.3: Properties of pipeline during lay operations.

Looking at the maximum twist angles, it is apparent that they differ only slightly. The maximum angle has increased marginally by around 0.2%. As mentioned before, a different mesh density was used in order to obtain a convergent system during plastic deformation. With an element size of 0.2m used in the plastic model compared to the 1m element size used in all the elastic models, the mesh was a lot more fine. To test the effect of the mesh density on the final twist results, a small mesh density test was done. The element size of the elastic model was decreased and compared with the results of the equivalent elastic 1m element size model. An element size of 0.2m made the elastic system instable, so after a mesh convergence study it seemed that 0.5m elements and larger gave converging results for the elastic model. The 0.5m and 1m element models were run and compared and the results can be seen in Table 6.4. The higher mesh density decreased the maximum roll angle insignificantly. It is therefore concluded that the difference in mesh density

is not the reason for the offset between the plastic and elastic model. Possibly the residual strains in the plastic model have lead to the slight increase of twist in the pipeline. However, one must remember that the amount of residual strain in the pipeline is quite small and is present at the outer areas of the pipes cross-sectional area. During actual reel-lay operations, the pipeline is plastically deformed over a reel with a much smaller radius and therefore obtains plastic deformations to almost the hart of its cross-sectional area. Furthermore, during reel-lay, the pipeline undergoes two bend cycles with a total of four times plastic bending instead of the modeled singular plastic deformation bending in Abaqus. It is therefore hard to translate the result obtained with the plastic model to what can be expected in reality, especially since only a minor difference between results is obtained.

$R_r[m]$	$\phi_{max,e}[^{\circ}] \ S_{el} = 1m$	$\phi_{max,p}[^\circ]$ $S_{el} = 0.5m$	Δ [%]
721.3	154.9	154.9	$7.396 \cdot 10^{-3}$

Table 6.4: Outcome effect mesh density on maximum twist angle for elastic Abaqus model

6.4. RESTART FOR PIPELINE LOWERING MODEL

In the restart model for the pipeline during lowering there are some lessons learned incorporated, giving a slightly different lowering model. The rigid element representing the end terminal of the pipeline is removed, and replaced by a point load an a moment in the last step of the model. The point load on the new bottom point of the pipeline (the trim point as formed in the plastic deformation model) acts in negative y-direction and represents the submerged weight of the end terminal. The moment is also activated on the same bottom point, and represents the moment contribution due to the offset of the end terminal's COG in comparison to the pipeline. Although the model improvement only has minor differences in comparison with the elastic lowering models, the loads and boundary conditions of the improved model are given in Table 6.5 for the sake of completeness. The stages of the lowering model look identical to the screenshots given in subsection 3.3.8.

 $\begin{array}{l} P_{s} = \text{Surface point of pipeline} \\ P_{b} = \text{Bottom point of pipeline} \\ M_{T} = \text{Total model} \\ g_{eq} = \text{Equivalent gravitational constant} \left[\frac{m}{s^{2}}\right] \\ F_{i} = \text{Instigating load } [N] \\ F_{c} = \text{Current load on pipeline } [N] \\ F_{e} = \text{Current load and weight of end terminal } [N] \\ M_{e} = \text{Overturning moment induced by offset COG of end terminal from pipeline } [Nm] \end{array}$

Table 6.5: The boundary conditions and loads als defined per step on the different instances

Model step	Instance	В	oun	dary	o con	ditio	ons			Lo	ads				
		х	у	z	θ	ψ	φ	х	у	Z	θ	ψ	ϕ	Туре	Unit
1	P_s	•	•	•	•	•	•	L -	-	-	-				
	P_b	•	٠	٠	٠	٠	•	-	-	-	-	-	-	-	-
	M_T	•	•	•	•	•	•	-	-	-	-	-	-		
2	P_s	•	•	•	•	•	•	L -							
	P_b	•	•	٠	-	-	-	-	-	-	-	-	-	-	-
	M_T	-	-	•	-	-	-	-	•	-	-	-	-		$\left[\frac{m}{s^2}\right]$
3	P_s	•	•	•	•	•	•	-			-		-		
	P_b	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	M_T	-	-	-	-	-	-	-	•	-	-	-	-	geq	$[\frac{m}{s^2}]$
		-	-	-	-	-	-	-	-	•	-	-	-	F_i	[N]
		-	-	-	-	-	-	•	-	-	-	-	-	F_c	[N]
4	P_s	•	•	•	•	•	•	L -	-				-		
	P_b	-	-	-	-	-	-	•	•	-	-	-	-	F_e	[N]
		-	-	-	-	-	-		-	-	-	-	•	M_e	[Nm]
	M_T	-	-	-	-	-	-	-	•	-	-	-	-	geq	$\left[\frac{m}{s^2}\right]$
		-	-	-	-	-	-	-	-	•	-	-	-	F_i	[N]
		-	-	-	-	-	-	•	-	-	-	-	-	F_c	[N]

6.5. COMPARISON PIPELINE DURING LOWERING CONTAINING PLASTICALLY PRE-CURVED BEAM VERSUS ELASTIC PRE-CURVED BEAM

As described in section 6.4, a restart is done from the plastic deformation model to obtain a model of a pipeline during lowering with prior plastic deformation. Since there are lessons learned incorporated in the model, an identical model for an elastic pre-curved pipeline was made. This model contains the same pipeline length, the same amount of residual curvature and the same boundary and load conditions. Once again, the sole difference with the plastic model is that the pre-curved shape is loaded in Abaqus without any strains or stresses in the pre-curved shape. Since the same prior plastic deformation model is used for the restart of the pipeline lowering plastic deformation model, it is chosen to use a longer pipeline length for the model comparison. The length that is chosen is 2302m, which is equal to the length of the pipeline used for the restart of the pipeline during laying with plastic deformation model.

For the model comparison, the elastic and plastic lowering model are subjected to the same end terminal contribution as used in the prior elastic lowering models. The pipeline and end terminal are subjected to a head current, acting against the residual curvature in the pipeline. The results of the models can be seen in Figure 6.11.



Figure 6.11: The twist development along the pipeline for a 721.3m residual curvature radius as calculated with Abaqus elastically pre-curved beams and plastically deformed pre-curved beams.

$R_r[m]$	$\phi_{max,e}[^{\circ}]$	$\phi_{max,p}[^{\circ}]$	Δ [%]
721.3	0.2735	0.2002	-72.93

Table 6.6: The maximum twist angles of an elastic pre-curved Abaqus beam model versus a plasticly deformed pre-curved Abaqus beam model during lowering operations.

Parameter	Value	Unit
D_o	8.625	["]
t_{st}	0.875	["]
t_c	2.5	["]
E_{st}	200	[GPa]
I _{st}	$6.74e^{-5}$	$[m^4]$
$ ho_{st}$	7850	$\left[\frac{kg}{m^3}\right]$
$ ho_c$	800	$\left[\frac{kg}{m^3}\right]$
$ ho_w$	1025	$\left[\frac{kg}{m^3}\right]$
α	90	[°]
L	2302	[m]
v_c	0.1	[m/s]
T_b	28.631	[Tonnes]

Table 6.7: Properties of pipeline during lowering.

The elastic model gives higher maximum torsion angles than the plastic model. The difference is around minus 73 %, but the absolute values of twist remain insignificant: the energetic difference in torsion of a 0.07° twist variance is very small. Therefore the influence of plastic deformation on the final twist contribution remains inconclusive. The amount of twist is negligible and still nowhere near the twist angles as obtained during the lowering of pipelines during the Lucius project. Once again there is a difference in mesh distribution between the elastic and the plastic model. Similar as in section 6.3 as mesh convergence study was done for the results of the elastic vertical suspended pipeline. Again it was concluded that the mesh density did not give a significant difference in results and is not the reason for the variance in results of the maximum twist angle.

7

THE INFLUENCE OF THE SPIRAL WISE WALL THICKNESS DISTRIBUTION IN SEAMLESS PIPES

Since the amount of twist obtained via the Abaqus models for the pipeline lowering operation was negligible compared to the amount of twist observed during actual operations at the Lucius Project, it was decided to investigate other possible instigators more thoroughly. One of the possible instigators as mentioned in section A.1, is the influence of the spiral wise wall thickness and yield strength distribution along seamless manufactured pipes[26]. The distribution is caused by the roller motion during manufacturing, which is continuous and therefore the imperfection is introduced in the same direction in all the pipes. A batch of pipes will therefore have the spiral rotating in the same direction along the pipeline. Upon connection of pipes from the same batch, the pipeline will have one long spiral in the same rotation direction as the individual pipes. Ergo, two pipes with the spiral in the same rotating direction, will form one spiral in the same direction independently of the way they are joined. For long pipelines such are the ones used at Lucius, a variable wall thickness could lead to a twist upon being loaded with increasing tension during lowering. It is therefore decided to investigate what the influence of tension is on the twist development of a pipeline with a spiral wise variable wall thickness distribution.

The idea that the spiral wise wall thickness distribution along the pipeline could lead to twist originates from the studying of the behavior of composites under axial loading [27]. Consider parallel fibers in a certain direction in a laminate plate as schematically depicted in Figure 7.1. When the plate is loaded by a tensile force at an angle ϕ_{σ} compared to the fibers in the laminate, a shear force is created (see Figure 7.2). The shear is due to a difference in the material properties of the laminate and the fibers, which leads to a shear force when the composite is loaded in anything but the principle stress directions. The wall thickness variation can be seen as a somewhat similar system, only the variations are not in the material but in the geometry along the pipeline.



Figure 7.1: Schematic representation of a composite plate, consisting out of a laminate, and fibers of a different material [27].



Figure 7.2: Schematic depiction of the material in the plate that is axially loaded at an angle ϕ_{σ} compared to the principle stress directions [27].

7.1. Small scale Abaqus pipe model with spiralwise distribution of wall thickness using measured data

Currently HMC is actively investigating the influence of imperfections on the behavior of pipes under certain loading conditions. One of those imperfections is the spiral wise wall thickness variation that could be present in seamless pipe. It is therefore decided not to disclose all the specifics of the model and the data obtained to represent the spiral imperfection. Only the essential information needed to understand the workings of the model are shared.



Figure 7.3: Schematic representation wall thickness variation in small scale Abaqus shell model. The difference in colors indicate a difference in wall thickness. Due to confidentiality no legend of the thickness values is provided.

A 12" outer pipe with a wall thickness of 15.9mm and a length of 0.8m is used for the small scale Abaqus model. The model contains standard quad node shell elements which use reduced integration (S4R [12]). Over the pipe, the spiral wise wall thickness distribution is added as an imperfection to the shell elements. The absolute value of the wall thickness variation and the sequence of the imperfections along the pipeline are based on prior research done by HMC on seamless pipes and will not be disclosed. The spiral is represented by a wall thickness increase along the pipe. Here, the thickness increase is defined using a sine function. A schematic representation is given in Figure 7.3. At one pipe end, the boundary conditions allow the pipe end to be displaced solely in the radial direction, but not in axial or rotational directions. For the other pipe end, the boundary conditions are defined such that the pipe is allowed to displace radially, longitudinally and twist around its own axis. No load conditions are present on the pipe. Furthermore, the material characteristics of X-65 steel are used to represent the material of the outer pipe. For a control model, the exact same properties and conditions are used, only without the spiral imperfections along the shell elements of the pipe. Both models are subject to the same axial displacement δ_A and the amount of angular twist is measured for both models and compared. Due to confidentiality, the actual Abaqus shell models cannot be depicted, but a schematic representation of the two can be seen in Figure 7.5 and Figure 7.6.



Figure 7.4: Segment of pipe crosssection (**blue**) with wall thickness increase due to the spiral (**red**)

Figure 7.5: Small scale Abaqus pipe model with spiralwise wall thickness variation

Figure 7.6: Small scale Abaqus pipe model without imperfections

7.2. TRANSLATING THE MAXIMUM AXIAL STRAIN IN THE PIPELINE DURING LOWERING TO EQUIVALENT AXIAL STRAIN IN ABAQUS MODEL

It is assumed that the highest amount of twist will occur when the pipeline reaches its longest freely suspended length during lowering. Here, the torsional resistance is at its lowest and the axial strain along the pipeline is at its highest. The amount of axial strain is not constant of the pipeline, but decreases from the tensioner downwards due to the submerged weight contribution. As a simplification, the total elongation of the pipeline is calculated and used to obtain an average strain in the pipeline. The elongation of a pipeline with similar characteristics as one to use with end terminal is given by the following:

$$\delta_r = \int_0^L \frac{T(z)dz}{A_r E_r} \tag{7.1}$$

Where the tension T(z) along the pipeline is given by:

$$T(z) = T_b + W_s(L-z)$$

Integrating gives us the following elongation and average strain of the pipeline:

$$\delta_r = \frac{T_b L_r}{A_r E_r} + \frac{W_s L_r^2}{2A_r E_r}$$

$$\epsilon_r = \frac{\delta_r}{L_r} = \frac{T_b}{A_r E_r} + \frac{W_s L_r}{2A_r E_r}$$
(7.2)

The average axial strain in the pipeline is set equal to the axial strain to be induced on the Abaqus model. Given this strain, the total elongation of the Abaqus pipe can be calculated.

$$\epsilon_r = \epsilon_A$$

$$\delta_A = \epsilon_A L_A$$
(7.3)

Elongation of the pipe modeled in Abaqus by δ_A gives an axial strain which is equivalent to the average strain in the real life pipeline. The pipe elongation is performed by adding a pure axial displacement step in both models (the one with and the one without the imperfection) and the amount of twist occurring is compared with one another.



Figure 7.7: Schematic representation of the Abaqus small scale models with spiral imperfections (**left**) and the real life pipeline with axial strains as encountered during lowering (**right**).

7.3. Results model test with measured imperfection data

As is mentioned in the previous section, both small scale Abaqus models are subjected to a pure axial displacement representing the strain that occurs during pipeline lowering operations. The angular displacement, or twist, is measured and the results are as follows:

- No twist for the model without imperfections (as expected).
- Twist for the model with spiral imperfections.

The amount of twist occurring for the model with spiral imperfections is measured by looking at the amount of angular displacement that occured for a row of elements along the pipeline after the axial displacement step. The results can be seen in Figure 7.8. Note that the twist development near the boundaries at the pipe ends have been disregarded due to influence of the boundary conditions. A few observations are made. First of all, it can be seen that the effect of the Poisson ratio is more significant than the wall thickness variation: the average twist angle caused by the Poisson ratio varies relatively little over the length of the pipeline. Furthermore, the angular displacement along the pipeline due to the wall thickness variation appears to be linearly increasing, but is highly oscillatory. Looking at the frequency of the thicker bands along the pipeline and the oscillatory motion in the twist development, it corresponds quite well. Along a thickening on the pipeline in longitudinal direction, the resistance to torsion increases and therefore the twist reduces locally. After the thickening, the torsional resistance is lowered again and the amount of twist locally increases again.

The thickening on the pipeline is added using a sine function where only its peaks have been added. One can clear see the result of this in the graph: the peaks have a sinusoidal shape, and the troughs are sharp due to the cut-off of the sine function.



Figure 7.8: Schematic representation abaqus wall thickness variation. The difference in colors indicate a difference in wall thickness. Due to confidentiality no legend of the values is provided.

To calculate the twist development due to the wall thickness variation, the average slope over the pipe is taken (ϕ_A). Using the total twist angle along with the pipe properties, the torsion in the model is calculated. This amount of torsion is set equal to the amount of torsion that would be encountered at the real life pipeline. With the properties of the pipeline, the expected torsion angle is calculated for derived torsion.

$$T_{A} = \frac{G_{A}J_{A}}{L_{A}}\phi_{A}$$

$$T_{A} = T_{r}$$

$$T_{r} = \frac{G_{r}J_{r}}{L_{r}}\phi_{r}$$

$$\phi_{r} = \frac{T_{r}L_{r}}{G_{r}J_{r}}$$
(7.4)

The calculated twist angle of the real life pipeline is equal to 0.017°, which is a negligible amount of twist. So it turns out that wall thickness variation along the pipeline could not be the reason for the large twist angles as measured during the Lucius operations.

7.4. Small scale Abaqus pipe model with increased spiralwise distribution of wall thickness

In additional research, the spiral wise wall thickness variation is exaggerated in comparison to the data that HMC measured. This is done to approximate the wall thickness distribution as presented in the paper [26]. The wall thickness distribution that is given in the paper can be seen in Figure A.1 in Appendix A. In Appendix A, it is mentioned that the wall thickness variation of a seamless pipe can be between +15% to -12.5%. Unfortunately no data is disclosed about the absolute wall thickness along the pipeline for Figure A.1. It is therefore chosen to take the extreme wall thickness tolerance that is allowed of 15% and to approximate the spiral as depicted in Figure A.1. The function that defines the wall thickness variation in the previous model (section 7.1) is therefore changed so that a maximum wall thickness increase of 15% is realized. Furthermore, the period of the sinusoidal wall thickness wave is lengthened, creating a higher spiral angle \angle_{sp} with respect to the longitudinal pipe edge (see Figure 7.9 and Figure 7.10). Three different spiral angles \angle_{sp} are modeled of 30°, 45° and 60°. The length of the shell model is increased to 12 meters to get rid of the end effects caused by the boundary conditions. Afterwards, again an axial displacement is executed as done in the previous models described in section 7.1. The same axial strain is achieved as in section 7.1.



Figure 7.9: A zoomed in side view of the left pipe end of the Abaqus shell model with the exaggerated spiralwise distribution of wall thickness. The spiral angle \angle_{sp} is equivalent to 45 °.



Figure 7.10: The wall thickness variation along the pipeline for the different spiral angles \angle_{sp} . Please note that the variations are only plotted over 4m of the Abaqus shell model.

The twist for the different spiral angles are given below in Figure 7.11. Here one can see that the amount of twist that occurs is significantly higher than as seen from the results of the shell model with the actual HMC measured wall thickness imperfection data (Figure 7.8). The highest twist ϕ_A in the small scale model is at a spiral angle of 30° and is equivalent to 0.08° over the 12 meter shell model length.



Figure 7.11: Schematic representation abaqus wall thickness variation. The difference in colors indicate a difference in wall thickness. Due to confidentiality no legend of the values is provided.

The highest twist ϕ_A that occurred during the model tests is translated to the expected twist for the 'real life' pipeline ϕ_r that was used earlier in section 7.3. Using Equation 7.4, the maximum pipeline twist ϕ_r is calculated to be 17.3°.

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7.5. DISCUSSION SPIRAL WISE WALL THICKNESS DISTRIBUTION

The model tests using the imperfection data as measured by HMC turn out to give very small twist along the modeled pipe segment. When translated to a realistic pipeline during lowering operations in deep water, the contribution of the imperfection to twist is deemed negligible. An important note however is that the pipeline measuring tests executed by HMC were done by a geometric scan of the internal pipe surface area. Although the internal variation of the spiral imperfection was identified using this method, no information was obtained about the eccentricity along the pipeline. This eccentricity can give large wall thickness variations along the pipeline, as mentioned in Appendix A. Figure A.1 is obtained via Automatic Ultrasonic Testing (AUT), where the wall thickness is measured circumferentially along the pipeline. Comparing the data from the HMC scans with the AUT scan, it appears that the development of the spiral wall thickness variation is different. It could be that during the fabrication process, a second, more dominating spiral wise wall thickness variation with a longer period is present. This was modeled with the increased spiral wise wall thickness models, and showed that for large wall thickness variations more significant twist angles can occur. Since no actual data about the eccentricity of the seamless pipes used at the Lucius project is available (which would give information about a possible second spiral), the latter models remain a qualitative study. The model does indicate that if extreme wall thickness variations are present, twist angles during lowering of tens of degrees can occur. Whether the increased variations are realistic, is subject for further research: in a long pipeline, not every joint will have the maximum wall thickness variation due to the eccentricity change along the pipeline.

8

THE POSSIBLE INFLUENCE OF CURRENT ON PIPELINE END TERMINALS

Research conducted on the onset of pipeline twist during pipeline lowering has not yet lead to twist results comparative to that observed during operations with the Aegir. The effect of residual curvature and the wall thickness variation along the pipeline do not seem to be the primary causes of pipeline twist during lowering. In the MATLAB and Abaqus models, the end terminal was simplified to force and moment contributions based on its weight, the offset of the end terminal 's COG in comparison to the pipeline and the drag force contributions due to current. All these analyses do not take into account the fluid dynamics to which the end terminal is subjected to during lowering. Thoroughly studying the effect of fluid motions around a porous structure such as an end terminal is regarded as a separate research, and due to its complexity along with a finite amount of research time available, will not be investigated in depth. However, using fluid behavior relations, the DNV code and recent in house research one is able to estimate the possible influence that current on the pipeline and its end terminal has on the development of pipeline twist.

8.1. POSSIBLE EFFECT OF PIPELINE END TERMINAL 'S CENTRE OF PRESSURE ON PIPELINE TWIST

During pipeline lowering the pipeline as well as the end terminal are subjected to wave drift forces and current loading. Upon investigation of the twist data during lowering, it is seen that significant twist of several tens of degrees occurs at a water depth of 700m and lower. The metocean data (Appendix G) shows that around 700m water depth, the effect of wave drift forces can be disregarded and that current is the most dominant and can be assumed constant during further lowering. The pipeline end terminal used for investigation of pipeline twist (see Appendix E) roughly consists out of the mudmat, central hub and the padeye. As a whole, the end terminal is a porous structure but will be simplified to a flat plate with a surface area similar to that of the mudmat. The current loading on the end terminal results in a drag and lift force, giving a resultant normal force which works perpendicular to the loaded surface. According to DNV-RP-H103 [21], the normal force, lift and drag coefficients for flat plates can be described by the following relations:



$$C_N = \begin{cases} 2\pi t a n \theta, & \text{for } \theta < 8^\circ \\ \frac{1}{0.222 + \frac{0.23\theta}{\sin \theta}}, & \text{for } 90^\circ \ge \theta > 12^\circ \end{cases}$$
(8.1)

$$C_L = C_N \cdot cos\theta \tag{8.2}$$

$$C_D = C_N \cdot sin\theta \tag{8.3}$$

Figure 8.1: Thin flat plate inclined to flow as given in DNV-RP-H103[21]



Figure 8.2: The normal force coefficient as a function of the angle of attack θ as defined by DNV.

Please note that DNV [21] does not give any information about the normal force coefficient between the angles $8 \ge \theta \le 12$. No reason is given for this absence in information and in further calculations the data for the undefined angles is estimated by interpolation between the two normal coefficient relations as given in Equation 8.1. Using the angle dependent normal force coefficient, the normal force for flow from a certain angle of attack can be calculated. Assuming that the PLET is only subject to current forces, the normal force of the pipeline is equal to:

$$F_N(\theta) = \frac{1}{2} \cdot \rho_w \cdot C_N(\theta) \cdot A_{PLET} \cdot v_c^2$$
(8.4)

The surface area of the PLET is simplified by taking the surface area of its mudmat, which forms the most dominant surface of the end terminal. The normal force acts on the center of pressure, which is the point where the total sum of a pressure field acts on a body. From aerodynamics it is known that the center of pressure of an airplane wing changes with respect to its angle of attack[28][29][30], see Figure 8.3 and Figure 8.4. The deviation of the center of pressure (COP) from the center of gravity (COG) in an airplane wing gives a pitch moment.





Figure 8.3: The center of pressure (CP) and center of gravity (CG) of an airplane wing for small angle of attack [31].

Figure 8.4: The center of pressure (CP) and center of gravity (CG) of an airplane wing for large angle of attack [31].

A similar mechanism could take place for current-PLET interaction during lowering, as is schematically shown in Figure 8.5. Current flow at a certain angle of attack will result in an offset of the COP from the COG located on the symmetric axis in the middle of the PLET. The offset times the normal force for the given angle of attack will give a yaw moment, which is transmitted as a torsion moment to the pipeline. In order to obtain the yaw moment with respect to the angle of attack, an estimation has to be made for the center of pressure along the PLET for different angles of attack. No research has been found concerning the COP of offshore structures subjected to flow from a certain angle of attack. Research done on thin flat plates in an air flow at different angles of attack [29] concludes that assuming the center of pressure to be equal to the center of gravity is quite accurate. However, one must take into account that the testing conditions of the plate along with the flow medium (air) are quite different from what can be expected during lowering. Furthermore, as has been mentioned before the shifting of the COP from the COG is a known phenomena in aerodynamics. Missing comparative research for a PLET in a current and not being able to use the data from the flat plate analysis from Ortiz et al. [29], an attempt is made to estimate the COP graph with respect to the angle of attack. It must be noted that this estimation is roughly based on the shift of the COP for airplane wings and general speculation. The results will not be used for quantitative analysis of pipeline twist, but solely to obtain an idea of the possible contribution a shift of the COP might have on the twist development. The hypothetical development of the COP against the angle of attack is given in Figure 8.6 and is schematically represented in Figure 8.5.



Figure 8.5: **Top View**. Schematic overview of the hypothetical change of the center of pressure (COP) with respect to the angle of attack of the current flow. The offset of the COP with respect to the center of gravity (COG) gives a moment arm defined as r_{COP} .



Figure 8.6: The center of pressure (COP) as a percentage of half the width of the mudmat (the maximum horizontal distance between COG and the end of the structure). By multiplying the percentage with half of the width of the mudmat, the absolute value of the moment arm is obtained.

Now that the COP graph has been estimated, the yaw moment for different angles of attack can be estimated. The yaw moment M_{yaw} , which is a function of the angle of attack, is given by the following:

$$M_{yaw}(\phi) = \frac{1}{2} \cdot \rho_w \cdot C_N(\phi) \cdot A_{PLET} \cdot v_c^2 \cdot r_{COP}(\phi)$$
(8.5)

During lowering, the torsional resistance of the pipeline decreases due to its increasing length. The deeper the pipeline goes, the larger the torsional angle will become for the same amount of torsion on the pipeline. Ergo, a constant yaw moment induced by current loading on the PLET will lead to an increasing twist during lowering. This progressive trend is also visible in the observed twist data from the Lucius project. The equilibrium between the yaw moment induced by current loading at a certain angle and the pipeline torsion induced by twisting to the given equilibrium angle can be used to estimate the amount of twist occurring for different directions of current loading (see Equation 8.6).

$$\Sigma M = 0$$

$$M_{yaw}(\phi) = T_p(\phi, L)$$

$$T_p(\phi, L) = \frac{GJ}{L}\phi$$
(8.6)

Given the pipeline properties of the 8" pipeline used at Lucius project (Table 5.1 in chapter 5), the pipeline torsion with respect to its twist angle and pipeline length is plotted together with the development of the yaw moment for different flow angles. The result can be seen in Figure 8.7. A relatively high current velocity (given the metocean data) of 0.2 m/s is used, and the PLET is initially loaded in sideways direction. The twisting of the PLET leads to linear increase of the pipeline torsion T_P and the development of the PLETs yaw moment as defined by Equation 8.5. Taken into account all the assumptions made, it is seen that for increasing water depth potentially large twist angles could occur ($\approx 60^\circ$ at 2100m).



Figure 8.7: The pipeline torsion development for different waterdepths wd together with the development of the yaw moment M_{yaw} with respect to the angle of attack of the flow. The intersection points give the torsion angle (which is equal to the angle of attack) that is obtained during equilibrium. The current flow is initiated sideways on the PLET.

8.2. IN HOUSE RESEARCH CONCERNING PLET TWIST

An inhouse study done by M. Rodermans on the hydrodynamic behavior of inline structures has shown that significant pipeline torsion can occur in an oscillating flow [2]. Given the pipeline parameters (Table 8.1), the end terminal parameters (Table 8.2) and the test conditions as given in (Table 8.3), torsion magnitudes of 15kNm per m of FLET length (so 225 kNm in total) were measured.

Parameter	Value	Unit							
OD	18	["]				Parameter	Value	Unit	
t _{st}	0.0219	[m]	Parame	eter Value	Unit		1	[<u>m</u>]	
W_s	73.6	$\left[\frac{kg}{m}\right]$	LEIE	т 17.8	[<i>m</i>]	u (i)-	0.2141	$\begin{bmatrix} r & s \end{bmatrix}$	
E	$2.0 \cdot 10^{11}$	$\left[\frac{N}{m^2}\right]$	Winings	1 11.0 down 95	[m]	ω_1	0.3141	$\begin{bmatrix} \frac{1}{s} \end{bmatrix}$	
EI	$1.583 \cdot 10^8$	$[Nm^2]$	Wings,	6 025	[<i>m</i>]	ω_2	0.6283	$\left[\frac{s}{s}\right]$	
ν	0.3	[-]	Wwing.	s, up = 0.023	[m]	β	30		
GI	$1.218 \cdot 10^{8}$	$[Nm^2]$	M	T = 0.4	[mT]	$T_{max,r}$	15	$\left[\frac{\kappa m}{m}\right]$	
ĒĂ	$6.760 \cdot 10^{9}$	[N]	vvs,FL	ET 117.55	[<i>m</i> 1]				
L	200	[m]	Table 8.2: ELET parameters as defined in Po			Table 8.3: Input	Table 8.3: Input parameters for oscil-		

Table 8.2: FLET parameters as defined in Rodermans' research [2]. Table 8.3: Input parameters for oscillating flow, along with maximum torsion as given in Rodermans research [2].

Table 8.1: Pipeline properties for the pipeline used in inhouse research done by M. Rodermans [2].

The flate plate used in the analysis was constrained against twist. If this would not be the case, the maximum torsion magnitude would result in a twist of around 25 degrees for the 17.8m FLET length and a pipeline length of 200m, which is significant. For the sake of argument, the torsion obtained in Rodermans' research is translated to a scenario with a PLET as used at the Lucius project together with the corresponding smaller OD pipeline.

$$T_{max,r} = T_{PF-01}$$

$$T_{PF-01} = \frac{GJ\phi_{PF-01}}{L_{PF-01}}$$

$$\phi_{PF-01} = \frac{T_{PF-01}L_{PLET}}{GJ}L_{PF-01}$$
(8.7)

Parameter	Value	Unit		
G	$7.576\cdot10^{10}$	$\left[\frac{N}{m^2}\right]$		
J	$1.3485 \cdot 10^{-4}$	$[m^4]$		
L_{PF-01}	200	[m]		
L_{PLET}	16.76	[m]		
W_{PLET}	5.2	[m]		
ϕ_{PF-01}	282	[°]		

Table 8.4: Input parameters for oscillating flow, along with maximum torsion as given in Rodermans research [2].

Calculations give that a static torsion of 15kNm per meter of PLET length would give a twist of 282° to 200m of PF-01 pipeline together with its first end PLET as installed at Lucius. Of course, 282 ° of pipeline twist is unrealistic given a constant current direction, but it does give an indication on the severity of torsion due to current loading on an end terminal. Note that the pipeline parameters are different, but the width and length of the PLET (or FLET in the case of Rodermans research [2]) are comparative (Table 8.4). Of course, there are many variables from Rodermans' research [2] which are different from the deep water conditions seen at Lucius. Dynamic loading is used in Rodermans' research, whereas at the deep water Lucius site the loading is mostly due to constant current. It should be noted that the PLET is simplified in Rodermans' research [2] to a flat plate, and that only one angle of attack (30°) at one current speed for two different oscillation frequencies. The research was done for relatively harsh conditions, and therefore the torsional values obtained are more at the extreme side at what could be expected. Nonetheless, it is evident that the influence of an oscillating flow on the torsion development can be significant. Whether a constant flow can also lead to significant torsion cannot be concluded from Rodermans' research [2] and additional research will have to be done.

9

DISCUSSION

The research has led to a better understanding of the development of twist during reel-lay operations. In this chapter, the validity of the research will be discussed, along with its limitations.

9.1. DIFFERENCE BETWEEN THE ANALYTICAL AND FINITE ELEMENT MODELS

Results comparison between the analytical and the finite element models show that the DOF/element simplification in the analytical approach has an impact on the validity of the results. For a pipeline during laying, the out-of-plane bending contribution becomes more prominent with increasing residual curvature in the pipeline. The analytical models have a significantly smaller computational time in comparison to the finite element models, and its input parameters can quickly be adjusted to specification. Another advantage is that the energetic benefit of twisting to a certain rotation angle can be seen, for one is able to plot the development of the strain energies against the twist angle. For the finite element models, only the end result of the twist is given. So although the analytical approach for pipelay operations gives less accurate results for high residual curvature, one must not forget the practical advantages the method has. The impact of underestimating the twist by a maximum of $\approx 35\%$ is something that can be questioned. The validity of the analytical approach for the lowering of the pipeline however is proven to be dissimilar to the expected twist behavior in reality during the the same stage of pipeline installation. The lack of out-of-plane bending contribution overestimates the amount of twist significantly, which leads to unrealistic results for the analytical lowering models.

9.2. CHOICE OF ELEMENTS FOR FINITE ELEMENT ANALYSIS (FEA)

For the finite element analysis, beam elements with pipe properties are used to approximate the geometry of the pipeline. In comparison with reality, the discretization of the amount of integration points along the cross-section of the pipeline leads to less information about the stress and strain development over the cross-section. However the overall curvature of the individual elements seem to be in accordance with the curvature profile of a pipeline in similar configuration as estimated with analytical equations. So for curvature calculations without stress concentrations or asymmetric stress patterns over the cross-section of the pipeline due to plastic deformation history, the assumption is proven to be valid.

9.3. The FEA model with plastically pre-curved beams

To incorporate the effect of plastic strains, models with residual curvature as a result of plastic bending were created. The amount of plastic strain in the pipeline is relatively low, compared to the strains induced during the actual reeling process. Also, the pipeline was bent plastically only once, instead of the four times which is the case during the actual reeling procedure. The simplifications are made with good reasons. To induce higher strains in the pipeline than used in the model, it is necessary to increase the mesh density along the beam model to ensure solution convergence. Together with the increase in iterative calculations for higher bending strains, this will lead to higher computational times. Further incorporating the other three bend steps as present in Aegir's reel cycle will increase the computational time even more. Given that the twist phenomena occurs at high water depths, bending only a small segment of pipeline will not give relevant twist data. For the simplified model, the computational time is around seven days. After speaking to the creator of the reeling model for the Aegir, it is estimated that modeling a pipeline during deep water pipelay with the entire history of the reel-lay process, would take a lot more computational time. Also the stability of the system is questioned. The plastic strain induced over the beam elements gives an average curvature along each element, but gives no information about the stress and strain development over its cross-section. Whether the asymmetric stress pattern along the cross-section which is present after reeling is influential on the twist development is unknown. The minimization of the potential energy of the system states that the work of the curvature along the pipeline is quintessential for determining the pipeline twist, regardless to the history of the residual curvature. This would mean that the twist development for the models with plastically induced residual curvature would be equal to those with elastically induced curvature. Also the literature review and prior research does not mention a possible influence, but given the discrepancy between the model results for pipeline lowering and the actual twist data, it is something that could be subject for future research.

9.4. The assumption of a static environment

All analyses done consider a static environment. At the beginning it was believed that the twist development is the result of static equilibrium of either strain energies, load or moments. The actual data during lowering was collected at depths where only (non-dynamic) current is present, and the twist accumulation during laying was measured by relieving the static torque in the system. These are all valid arguments that a static environment is sufficient for analyzing the twist development. Now that all research is completed, all static assumptions made for the pipeline during laying are still considered accurate. For the lowering however, there appears to be a possibility that the simplification leads to inaccurate modeling. The current loading is simplified to constant loads, not taking into account possible fluid dynamics. Current research shows that potentially large twist angles can occur. Given the discrepancy between the actual twist data and the results of the static FE models, it could very well be that the current interaction with the end terminal plays a vital role in the development of pipeline twist. But one must not forget that the energetic value of the twist is relatively low, and that the twist would be elastic: Removal of the current would lead to the twisting back of the pipeline to its initial configuration. To definitively conclude that the static assumptions for the pipeline during lowering is valid, the influence of current-structure interaction would have to be researched with computational fluid dynamics or another fluid structure interaction based approach.

9.5. COMPARING MODEL RESULTS WITH ACTUAL DATA

One must be careful when comparing the modeled data with the actual twist data. Eye witness accounts explain that the accuracy of the twist measurements during pipeline lowering is limited to $+/-10^{\circ}$ due to parallax from viewing the end terminal at a distance with an ROV. Also, for the pipeline during laying models it is decided to only model the pipeline from the vessel to the interaction with the seabed. In none of the models also a piece of pipeline is laid additionally to suspended pipeline during laying configuration. The results of the models already showed twist amounts comparative to the actual data, and it was therefore seen as unnecessary to further develop the laying models. Moreover, the discrepancy between actual data during lowering and the results of the equivalent models lead to a shift in research focus. One must also take into account that laying a segment of pipeline on the seabed means that pipeline-seabed interaction properties must be considered. This is decided to be outside the research scope.

9.6. DATA REGARDING SPIRAL WISE WALL THICKNESS DISTRIBUTION

The spiral wise wall thickness distribution was measured by HMC via a geometric scan of the internal pipe surface area. The wall thickness increase due to the spiral is relatively small in comparison to the tolerances set for the wall thickness variations in seamless pipes. Comparing the data from the HMC scans with the AUT scan done by Technip as given in Figure A.1, it appears that the development of the spiral wall thickness variation is different. It could be that during the fabrication process, a second, more dominating spiral wise wall thickness variation with a longer period is present in the pipeline . Further research has to be done using AUT on seamless pipelines to verify this possibility.

9.7. Assumption of constant residual curvature

All residual curvature added to the system is assumed to be constant over the length of the pipeline. In reality, every pipe segment welded together to form a pipeline will have slightly different material and geometric properties. Residual curvature introduced to the pipeline after straightening will therefore vary around an average value along its length. How much the residual curvature deviates from its average is unknown, but it is considered to be of far lesser influence to the twist development.

10

CONCLUSIONS AND RECOMMENDATIONS

The aim set in this thesis was to find an answer to the question: "What causes twist in reeled pipelines during lowering and lay operations?". It was decided to investigate the two consecutive operations separately due to the difference in pipeline configuration and load cases. The conclusions that can be drawn from the research of the two systems are given in section 10.1 and section 10.2. For the twist occurring during lay operations, the primary reason has been found and has lead to a clear understanding on the preventive measures that can be taken to decrease the amount of twist. For the twist occurring during lowering, several suspected instigators have been ruled out, limiting significantly the amount of possible causes. The recommendations for practice regarding reel-lay operations with the DCV Aegir will be given in section 10.3. Lastly, recommendations for further research are discussed in section 10.4.

10.1. CONCLUSIONS - TWIST DEVELOPMENT DURING PIPELINE LAY OPERATIONS

It was chosen to use analytical equations in MATLAB to approximate pipeline twist during lay operations and to use similar finite element analysis models in Abaqus for result comparison. For the estimation of the twist development during lay operations using analytical equations, the method of total potential energy minimization together with enhanced stiffened catenary theory is proven to give comparative results to actual observed twist data during Lucius project. The same is concluded for the FE models. Upon model comparison however, a difference is observed in absolute twist values for the sensitivity studies (max 35%). Further research indicated that the out-of-plane bending contribution, which is not taken into account in the 4DOF/element MATLAB models (see subsection 2.8.1), plays a role in determining the amount of pipeline twist. It is concluded that an increase in the residual curvature in the pipeline leads to an increase in the out-of-plane bending contribution and therefore decreasing the accuracy of the 4DOF/element MATLAB models in comparison to the 6DOF/element Abaqus models (see subsection 2.8.2). Ergo, the theory of energy minimization using solely the in-plane bending strain contribution gives twist results comparable to actual measured data, but has its limitations regarding accuracy for high residual curvature in the pipeline.

The primary instigator of pipeline twist during laying is found to be residual curvature in the pipeline after straightening. The higher it is, the more torque is built up during laying and ultimately more twist is developed in the pipeline. Besides the magnitude, also the direction of the residual curvature with respect to the lay direction appears to be quintessential: under-straightening leads to high twist angles due to bending against the residual curvature in the sagbend, whereas over-straightening gives minimal twist because the residual curvature is in the same direction as the bend in the sagbend. Besides the residual curvature, also the water depth is of importance as no twist occurred in the shallow water depth models, even when the maximum amount of residual curvature allowed by HMC was tested. This explains why HMC has encountered no twist at the shallow water Ichthys project, while there was significant twist at the deep water Lucius project. Other pipeline parameters concerning twist development that were studied appear to be of lesser influence. The variation of the pipeline tension, buoyancy, wall thickness and outer diameter for a constant residual curvature have little effect on the twist development and on its magnitude.

Subsequently, the effect of plastic deformation in obtaining pipelines with the residual curvature was investigated using finite element analysis. The results of the research show almost no difference in twist in comparison with a similar model with residual curvature in the pipeline that was added elastically. It must be noted however that the plastic model was designed for a qualitative study. The process of inducing residual curvature in the pipeline is simplified to only one half bend cycle and the amount of plastic strain induced in reality is far higher. Given that the qualitative study gives no difference in results and the dissemblance of the plastic model in comparison to Aegir's reel-lay process, the test is deemed inconclusive.

10.2. CONCLUSIONS - TWIST DEVELOPMENT DURING PIPELINE LOWERING

For the approximation of the pipeline twist during lowering, again analytical equations together with total potential energy minimization theory were used to create MATLAB models and for comparative research the pipeline was also modeled in Abaqus using finite element analysis. For the analytical analyses, a pipeline with residual curvature subject to current loading was modeled using both linear and non-linear beam equations. It is concluded that linearizing the system does not have any adverse effects on the structural analysis, for the pipeline in deep water is subject to small displacements and bending angles. Upon comparison with the finite element models however, a large discrepancy in the results is observed. The finite element models show negligible pipeline twist during lowering under all tested static loadings and residual curvature conditions, while the analytical models show significant twist for the same tested conditions. Further investigation proves that once again the out-of-plane bending contribution, which is taken into account in the 6DOF/element finite element models (see subsection 3.3.8) but not the 4DOF/element MATLAB models (see section 3.1), plays a vital role for the pipeline twist estimation during lowering procedures. For the MATLAB models, a reduction of the in-plane residual curvature due to twisting together with the build-up of torsion does not take into account the increase of the out-of-plane curvature. Ergo, due to the 4DOF/element simplification twist is perceived as energetically more beneficial than is actually the case. The derived analytical approximations for pipeline lowering are therefore deemed as unfit for the estimation of pipeline twist during lowering.

The finite element models with residual curvature in the pipeline and static loading conditions show virtually no pipeline twist, which does not match the twist data obtained at Lucius Project. Unlike the twist during laying, it is concluded that residual curvature is not the primary instigator for twist during pipeline lowering. As a result other possible twist instigators were investigated. Given the wall thickness variations induced during manufacturing of seamless pipes, a finite element shell model was created. The model represents a piece of pipe with a spiral wise wall thickness variation along its length as measured with geometric internal pipeline surface tests by HMC. The model was subjected to tension and showed the development of twist. The properties and findings of the shell model were translated to conditions similar to pipeline lowering, and the calculations show that the amount of twist possibly occurring due to the measured spiral wall thickness variation along the pipeline is negligible. Additionally, the imperfection was exaggerated, but still within the limits of pipeline tolerances. For the same model conditions, a higher twist angle is obtained for a realistic pipeline during deep water lowering. Whether the exaggerated variations are realistic, is subject for further research. Given the HMC data for the imperfection, it is concluded that the measured wall thickness variation is not the primary reason for pipeline twist.

Similar to the additional research done for the pipeline during laying, the effect of plastically induced residual curvature is compared to elastic residual curvature. No significant additional twist is calculated for the plastic model under different loading conditions in comparison to a similar elastic model. However, again the amount of plastic deformation is small and the entire plastic deformation process of Aegir's reel-cycle is simplified. Given these factors and the fact that qualitatively no real difference in results is obtained compared to the elastic model, the effect of plastic deformations during the reeling process on the development of pipeline twist remains inconclusive.

Also a closer look was taken into the effect of current on the system. In prior models the effect of current was modeled using static loads, therefore not taking into account possible effect of fluid motions. The effect of an end terminal induced yaw moment due to an estimated traveling center of pressure shows potentially large pipeline twists can occur as seen in the actual data of the Lucius project. Since no actual computational fluid dynamics were done, the center of pressure development with respect to the angle of attack of the current remains a qualitative approximation. Comparable in house research with fluid dynamics taken into account ([2]) gives results which are in accordance with the aforementioned approximation, which is promising. It is noted that given the low torsional rigidity for the relatively small diameter pipelines in deep water, a relatively low instigating moment is needed for the lowering phase to give the twist angles recorded during Lucius Project. This would mean that relatively low energy instigators can give large twist angles during lowering, but that on the other hand mitigation procedures will not require large forces or moments for twist correction.
10.3. RECOMMENDATIONS FOR PRACTICE

The research has shown that residual curvature is the key instigator for pipeline twist during lay operations. The amount of twist can be mitigated by the following:

- Reduce the amount of residual curvature in the pipeline as much as possible. It is concluded that lowering the amount of residual curvature in the pipeline decreases twist in the pipeline during laying. Therefore it is recommended to increase the HMC minimum residual curvature radius criteria, which is currently equal to $R_r = 500m$.
- Keep the residual curvature in the over straightening domain if twist reduction is of primary concern. The research shows that having a residual curvature in the pipeline due to over straightening reduces the amount of twist during laying to a negligible amount, for the pipeline is bent in the same direction as the residual curvature. It must be noted however, that over straightening the pipeline brings higher strains to both the pipeline as well as the straightener itself. It would have to be investigated if they can cope with the over straightening, and whether the benefits of less pipeline twist outweighs higher strain related challenges. To keep track whether the pipelines are over- or under straightened, it is recommended to change the straightening trail procedures. By marking the pipeline on one side, one can keep track in which direction the pipeline is curved once laid on deck for straightness measurements.

10.4. RECOMMENDATIONS FOR FURTHER RESEARCH

The thesis research has also lead to recommendations regarding future research. Especially the investigation of the pipeline lowering has spiked enthusiasm in investigating alternative twist instigators. My principle recommendations for subsequent research are given below.

- Further investigate the effect of current on the development of twist during pipeline lowering. In my research the current-structure interactions have all been modeled statically and do not take into account fluid motions. It is therefore recommended to do a computational fluid dynamics study of an end terminal loaded by current of different magnitudes and directions. This would give valuable information on the amount of torsion, and subsequently pipeline twist, that can be induced due to current.
- Research the influence of the plastic bending history on the development of pipeline twist. In this research, simplified models give inconclusive results regarding the effect of plasticly bending pipelines to a certain residual curvature. It is therefore recommended to form pipeline models with residual curvature created via the complete reeling cycle of the Aegir. Modeling pipelines with the full reeling history will be a challenge from a computational point of view. It will require a more innovative approach of modeling. It is recommended to look into the possibility of transferring and multiplying the essential data (like the stress and strain profile) from a small segment of pipeline reeled with the in house reeling model to my pipeline laying models that have a longer pipeline length. Given the FEA complexity, further research would be more suitable for an experienced FEA engineer instead of a graduate student.
- Study the possibility of adding the out-of-plane bending contribution to the models that use analytical equations. This will lead to increased accuracy of the models and will also make it possible to approximate twist for pipelay operations where the pipeline is laid in a curve, an operation which is quite common in the offshore.
- Investigate the possibility of a second, more dominant wall thickness variation spiral along the pipeline due to eccentricity changes along the pipeline. Accurate wall thickness data will have to be acquired using AUT on seamless pipes, preferably comparative to those used during the Lucius project. A subsequent step would be to transfer data to an Abaqus shell model once more, an quantitatively identify its contribution to pipeline twist.

A

PROBLEM DESCRIPTION PIPELINE TWIST DURING REEL-LAY OPERATIONS

An overview of executed tests done to better understand the twist development during pipeline lowering and laying is given below.

A.1. PIPELINE PROPERTY TESTS

All the pipelines for the Lucius project have been made out of seamless pipes. The main advantages of using these type of pipes is the good track record in service and the absence of welds in the longitudinal direction[36][37]. The disadvantages however, are that they can have a fairly wide variation of the local wall thickness, typically +15% to -12.5% and out-of-roundness and -straightness [37]. Also, the outer surface of the pipe may be highly distorted such that when it is grit-blasted prior to coating, tiny slivers of steel rise up. These slivers can create an issue for coating bonding, especially when dealing with a thin anti-corrosion coating such as fusion-bonded epoxy (FBE). One of the concerns is the influence of variations in material- and geometric properties of the pipeline in both its circumference as in longitudinal direction regarding the onset of pipeline twist. Tests done on seamless pipes in 2009 by Technip [26] show that wall thickness variations due to pipe eccentricity can be quite severe. Also, it can be seen in Figure A.1 that there can be a significant variations both along and around the pipe joint regarding the wall thickness and the yield strength. What can also be seen in Figure A.1, is that the wall thickness appears to be spread in helix or spiral like bands in longitudinal direction. Internally, HMC has also done numerous tests concerning the distribution of the wall thickness over the circumference and longitudinal direction of the seamless pipes, and have also found the helix shaped variation of the wall thickness. These tests were done by 360 deg scans of the internal surface area of seamless pipes used for HMC projects. Due to confidentiality, no specifics can be given about the exact thickness differences and the spiral frequency. Given the fabrication steps of the seamless pipes where the pipe is rotationally driven through the pinch zone of two rollers which are also rotating, the peculiar spread of the longitudinal wall thickness is linked to the pipe's manufacturing method. The difference of geometric- and/or material properties in the circumference and longitudinal direction of the pipeline, may play a role in the onset of pipeline twist.



Figure A.1: **left:** Qualitative variations of the wall thickness over the circumference of the pipeline versus the pipeline length. **right:** Example of extreme yield strength variation through a pipeline cross section. every 10 degrees a sample was taken, given a total of 36 sample locations over the circumference. Source: [26]

A.2. SPOOLBASE TESTS

At the spoolbase in Carlyss, USA, the twist of the pipeline was measured during counter-clockwise spooling on of the pipe stalks onto the reel-drum [4]. Red stripes were painted at the 6 o'clock and blue stripes were painted at the 3 o'clock orientation, which repeated in constant intervals along the stalks. See Figure A.2 and Figure A.3. This was done for a test pipe of 16", and for the pipe stalks for the Production Risers PR-04 and PR-05 later used for installation at the Lucius Project. Results from the tests were obtained by visual inspection of the shift in position of the stripes over the reel in comparison with its initial position. The results are not given in the repository version. A few observations can be made. First of all, the amount of twist seems to be random. The production riser stalks all have the same properties and initial length, but only some of the stalks had pipeline twist and for the pipelines that did twist, the amount was non-equivalent to one-another. Another observation is that the twist always occurred counter clockwise, possibly indicating that the pipeline has a rotation preference. The test pipe used had a larger diameter and a different length than the production riser stalks, making it difficult to correlate the difference in occurred twist between the two types of pipeline with the change in geometric and/or material properties. But looking at the data, the length of the pipeline and the geometric properties (pipeline's outer diameter) could be of influence on the final amount of pipeline twist. Internal observations note that the pipe twist could possibly have been caused by the way of loading the pipeline onto the track towards the reel: Loaders were used to lift the pipeline and skid them onto the track for reeling-on. Due to the relatively low torsional rigidity of the long stalk lengths in combination with the small diameter of the pipeline, twist could have been introduced during transportation of the stalks onto the track.



Figure A.2: Markings made on pipe stalks before spooling on



Figure A.3: Visible rotation of markings on pipeline during spooling on

A.3. REELING TESTS

Another investigation into the twist phenomena took place during the spooling off of one of the Flowlines (PI-04) and the pipeline passing through the Reel/J-lay tower [38]. Again, red stripes were painted at the 6 o'clock orientation, on the pipelines that were accessible from scaffolding beneath the reel. In total, 8 markings were made, all on the same layer of pipeline on the reel (unequivocally the last layer). All pipelines contained an added 2.5" GSPU coating in comparison to the tests done at the spoolbase. The markings were monitored on top and bottom of the upper tensioner, and on the auxiliary welding station. At none of the monitored positions and for none of the 8 markings respectively, was twist of the pipeline monitored. The test was once more done by visual inspection.



Figure A.4: Markings made on PI-04



Figure A.6: Marking coming out of upper tensioner



Figure A.5: Marking entering the upper tensioner



Figure A.7: Marking entering lower tensioner

A.4. TWIST OBSERVATION DURING LOWERING

During the lowering of the pipeline to the target box at the seabed, the amount of pipeline twist was measured[4]. Via ROVs the rotation of the end terminals attached to the end of the pipelines was monitored during lowering. The ROV maintained a constant distance from the end terminal without rotating with the twist in the pipeline. Parallax, the phenomena of an object's apparent displacement due to a change in the observer's point of view, was mentioned as a possible influential factor regarding the accuracy of the measurements made. Along with the fact that the observations were done via visual inspection through the eyes of an ROV, the accuracy of the measurements was estimated at -/+ 10 deg. Depending on the type of pipeline, the twist was measured only once or numerous times during the descent to the seabed. Upon arrival, the rotation of the end terminal is rotated back using either the ROVs or by temporarily changing the heading of the vessel. Given the water depth at the site location (2100m), all measurements done for pipelenghts longer than the waterdepth have been derived using a different method. As part of a twist mitigation procedure, the second end of the pipeline was attached to a rotational swivel connected to the Abandonment and Recovery Winch of the Aegir before installation of the second end terminal. Upon release of the clamping constraint given by the tensioner tracks, the build up torsional energy was relieved in the form of rotation around its own axis (twist). The measured twist after torque relieve is given in Figure A.8 by the points after a waterdepth of 2100m.



Figure A.8: Measured twist during lowering of pipelines at Lucius project. **Note:** All data point before a length of approximately 2100m were derived via ROV visual inspection, and all data after via measuring the twist angle after torque relieval.[4]

B

DECOMPOSITION OF AEGIR'S REEL-LAY PROCESS

Heerema's newest addition to the fleet, the deepwater construction vessel Aegir is capable of executing complex infrastructure and pipeline projects in ultra-deep water and has sufficient lifting capacity to install fixed platforms in relatively shallow water. The vessel is equipped with a pipelay tower which is able to install pipelines via J-lay or Reel-lay. Only the reel-lay process will be of importance for the thesis assignment. In this chapter the different steps of the process, from fabrication of the pipe to the laying of the pipeline on the seabed will be briefly dicussed.



Figure B.1: HMC reel-lay vessel DCV Aegir

B.1. FABRICATION OF SEAMLESS PIPES

For the Lucius project, seamless pipes were used to create the pipe stalks. Seamless pipes are created by hot working a solid rod to form a pipe without any longitudinal welds. The two main methods, which are commonly used are the Mandrel Mill and the Mannesman Plug Mill process[35]. In both seamless pipe processes the pipes start out as a solid round rod called a billet, which is heated in a furnace. Afterwards the heated billets are rotationally driven through the pinch zone of two rollers which are also rotating. The rotary motions of the billet and rollers leads to a reduction of the stress needed for the piercer between the rollers to penetrate the rod and to shape it into a pipe. The first piercing step produces the primary pipe shape. Afterwards the pipe is brought to a finish in a number of steps, dependent on which process is used. See Appendix D for further fabrication information.

B.2. PIPE STALK FORMATION AND SPOOLBASE HANDLING

At the spoolbase, pipe segments are welded together to obtain pipe stalks of up to 2km[39]. The individual pipe segments before welding are 12m. The choice of pipe is predominantly dependent on the required diameter, thickness, and D/t ratio, but mostly seamless pipes are used for Aegir's reel lay operations. After each pipe stalk formation step, the stalks are shifted from one line to the next for further welding, testing or coating (see Figure B.2). The shifting is done using roller boxes suspended in excavators riding along the stalk length. Eye witness accounts suggest that the process of shifting with the excavators can incorporate twist in the stalk.



Figure B.2: Schematic overview of the Spool Base Layout (Top View)

B.3. AEGIR'S SPOOLING PROCESS

The spooling process comprises a few sequential steps from spooling on the pipeline until overboarding it. The steps will briefly be discussed in the following segment.

B.3.1. Spooling on of pipeline

The spooling process begins onshore [39][40]. The formed pipe stalks are rolled up, starting from underneath the reel drum. Separate stalks can be welded together to fill the reel drum with a few km of pipeline. The minimum radius of the reel drum is 8m and the pipeline stalks can be loaded to the reel up to the radius of 12m. The range of pipe diameter that can be reeled onto the drums is from 6" to 16". The reel drum is positioned on a barge moored at the spool yard, for the ease of subsequent transportation. During the spooling on process onshore, the pipe is first bent elastically, leading to the increase of the internal moment of the pipe. By further bending, the pipe is deformed plastically. This leads to the increase of the curvature of the pipe, but not in the significant increase of the internal moment. The first step of the spooling process is given by the blue line in Figure B.4, ending in point *A*.

B.3.2. TRANSPORTATION PIPE STALKS AND REELS

Once the pipe stalks have been spooled onto a reel drums a barge brings the reel to the Aegir's location where it can be transferred to the deck of the vessel by the 4000mT crane on board. Subsequently an empty reel can be loaded onto the barge for refilling. At a given time the Aegir has the capacity for three spools on the vessel, two fulls reels and one empty one. Once a reel is empty, it can be shifted onboard and the following reel can be loaded into the pipelay tower. It is therefore unnecessary to terminate the lay procedure for pipe logistical reasons such as sailing back on forth to the yard to refill the spool with stalks. When keeping a structure in a strained condition for a finite amount of time, like a pipeline on a reel drum, it is known that the internal stress response can decrease for the same amount of constant strain. This phenomena, called stress relaxation, could also occur during the transportation phase of the reel drums from the spoolyard to the Aegir vessel. This phenomena however is not taken into account during further analysis, and therefore is not of influence on the moment-curvature relation as plotted in Figure B.4.

B.3.3. PIPELINE UNSPOOLING

Once the reel drum is in place, the unspooling of the pipeline can commence or continue. The pipeline is pulled off the reel drum towards the aligner, see Figure B.3. As the pipeline leaves the reel-drum, the internal moment is reversed and the curvature of the pipe is decreased via the orange line segment in Figure B.4 to point *B*. Again, during the unspooling the pipeline deforms plastically, completing the first bend cycle.

B.3.4. BENDING OVER THE ALIGNER

The second bend cycle starts off with the plastic bending of the pipeline over the aligner wheel, which has a fixed radius of 9m. Once again, the curvature of the pipeline is increased up to point *C* inFigure B.4

B.3.5. PIPELINE STRAIGHTENING

The pipeline is plastically deformed for the last time at the straightener. Using three point bending, the pipeline is given an excess curvature, which upon relieve of the moment enables it to elastically spring to a configuration of zero residual curvature (in theory at least). As the pipeline leaves the aligner wheel, it comes in contact with the outer straightener. The internal moment is reversed and the curvature is reduced. At the second part of the straightener, the inner straightener, the pipe is bent to the point where a negative curvature is applied in order to compensate for the elastic deformations of the pipe at point D in Figure B.4. In the last step of the straightening procedure, all residual internal moment is relieved by the elastic rebound of the pipeline (point E). Theoretically, also all curvature in the pipeline will be removed.

B.3.6. THE TENSIONERS

The tensioners have two functions: they maintain the back tension in the pipeline so that no unwanted buckling occurs in the pipeline during reeling and they hold the weight of the pipeline catenary from the moonpool downwards (top tension). The tensioners have a maximum capacity of 800mT top tension and are capable of executing pipelay installation projects in ultra-deep water for infield flow lines and risers.



Figure B.3: Reel-lay tower Aegir



B.4. STRESSES AND STRAINS DURING SPOOLING PROCESS

[41][42][43][44] When a symmetric cross-section like a pipeline is solely loaded by a bending moment, the neutral axis is equal to the axis of symmetry. However, if that is also an axial force present, such as a back tension during reeling, the neutral axis will not be on the axis of symmetry. For a symmetric cross section loaded by a bending moment only the neutral axis is on the axis of symmetry. However, if the cross section is also loaded by an axial force the neutral axis is not on the axis of symmetry. During the double bend cycle of the reeling process, the pipeline is plastically deformed 4 times. The relationship between the bending moment and the curvature are given in Figure B.4. When denoting the step number with the subscript i and letting z = 0 be the location of the symmetry axis, then the bending moment and axial force with respect to the symmetry axis can be expressed by:

$$M_{z,i} = \int_{A} \sigma_i(z) z dA \tag{B.1}$$

$$N_{z,i} = \int_{A} \sigma_i(z) dA \tag{B.2}$$

The stress at a certain step i can therefore be expressed as,

$$\sigma_i(z) = \sigma_{i-1}(z) + \Delta \sigma(z) \tag{B.3}$$

Given that the strain at the axis of symmetry is denoted as *a*, the strain at each step and the strain difference is given as follows:

$$\epsilon_i(z) = a_i + \kappa z \tag{B.4}$$

$$\Delta \epsilon_i(z) = a_i - a_{i-1} + \kappa_i z_i - \kappa_{i-1} z_{i-1} \tag{B.5}$$

The location of the neutral axis d with respect to the symmetry axis at a certain step i is given by:

$$d_i = \begin{cases} \frac{a_i}{\kappa_i}, & \text{for } \kappa \neq 0\\ 0, & \kappa_i = 0 \end{cases}$$
(B.6)

Figure B.5 gives a schematic representation of the aformentioned relations between, strain, neutral axis and curvature.



Figure B.5: Schematic representation of strain development in steps *i*

During spooling on, the pipeline will first deform elastically; the stress will increase linearly with the strain increase.

$$\sigma = E\epsilon \tag{B.7}$$

However, upon further bending, plastic deformation will occur and the relation between strain and stress changes. The stress increase is then dependent on strain hardening relations^[45][44]. There are different ways of modeling the inelastic response of metals. The most commonly used in finite element analysis will be discussed: isotropic hardening, kinematic hardening, or a combination of the two (real metals exhibit both types of hardening).

B.4.1. ISOTROPIC HARDENING

When modeling with isotropic hardening, it means that plastic deformation will lead to an equal increase of the yield strength for the member tension or compression, irrespective of the type of plastic loading that occurred. The yield surface remains the same shape, but expands with the increasing stress. Isotropic hardening does not change the shape of the yield function. See Figure B.6. Therefore, an initial asymmetric yield function will stay asymmetric, but will only radially increase in size. Also the history of the material should be taken into account: every plastic bending step will lead to the increase of the yield surface. When modeling the reeling process with isotropic hardening, the first plastic bending cycle is modeled with a so called Lüders Plateau. This Plateau is caused by localized bands of plastic deformation in metals experiencing tensile stresses (Lüders bands) that require an amount strain before strain hardening occurs. A qualitative representation of the effect of isotropic loading on the double bend cycle of the reel-lay process is given in Figure B.8. It is seen that the Lüders plateau disappears after the first half bend cycle.



Figure B.6: Isotropic strain hardening



Figure B.7: Kinematic strain hardening

B.4.2. KINEMATIC HARDENING

For an isotropic model, if initially the yield strength in tension and compression are the same, the yield surface is symmetric about the stress axes, and will remain symmetric as the yield surface increases due to plastic strain. In order to take into account the Bauschinger effect or similar responses which are present during cyclic loading such as during spooling, kinematic hardening is used. The Bauschinger effect describes the following: When materials are loaded uni-axially in one direction into the plastic regime, unloaded to zero stress level, then reloaded in reverse direction, they may yield during reloading, at a stress level lower than if the reloading were carried out in the original direction [46]. Ergo, when a pipe is bent in one direction, it will yield at a lower stress during subsequent bending in the opposite direction. Hardening in tension will lead to a 'softening' in a subsequent compression: the yield surface remains the same shape and size but merely translates in the stress space. See Figure B.7.

B.4.3. STRESS - STRAIN CURVE DURING REELING

The strain hardening effects can been seen in the stress strain curve of the double bend cycle. In the first step, the aforementionned Lüders plateau is visible, followed by strain hardening. In the second step one sees a difference between Figure B.8 and Figure B.9. In the former, one sees the typical behavior for isotropic hardening during the bending back of the the pipeline: the elastic region ends at a stress equal to $-\sigma_1$ and afterwards the Bauschinger effect comes into play during plastic deformation. For the kinematic model, it can be seen that the strain hardening during the first step translates to an earlier onset of plastic derformation. In step 3 and 4 the bend cycle is repeated for both models and in step 5 one can see the decrease of the strain to zero, but a residual stress remains in the pipeline!



Figure B.8: Isotropic strain hardening in reeling cycle



Figure B.9: Kinematic strain hardening in reeling cycle

B.4.4. CROSS-SECTIONAL ASYMMETRIC STRESS DISTRIBUTION OP PLASTICALLY DEFORMED PIPE

During the reeling cycle, the pipeline is deformed plastically four times under a back tension. The consecutive bending under tension leads to the build-up of a cross-sectional asymmetric stress distribution. One must note that for zero curvature at the pipeline segment at a given pipe cross section is, asymmetric stress distribution will not result into an internal moment. The summation of the products of the cross-sectional stress increments with the corresponding arm lengths will give zero. A simple example is given below. Assuming a tensioned pipeline made out of an elastic-perfectly plastic material, the pipeline is bent into plastic deformation. In the second step, the pipeline is bent plastically the opposite direction, as such that when the bending moment is relieved, the pipeline elastically deforms to a state with zero residual curvature. Although the asymmetric stress distribution remains present over the circumference of the pipe due to the persistent tension, the summation of the moments over the area gives zero and so the overall strain is zero.

B.5. RESIDUAL CURVATURE IN THE PIPELINE

Once the first pipeline segments has been pulled through the entire reel system on board, iterative minimization of the pipelines residual curvature via testing commences. After the pipeline has passed the tensioner, a segment of 12 m pipe is cut from the pipe stalk. This segment is then laid on the deck, and measured for its out-of-straightness. The out of straightness of a pipeline can be converted to curvature either using Pythagorean theorem or intersecting chord theorem[47][48]. It is assumed that the curvature radius is constant over the measured piece of pipeline. It can therefore be seen as an arc on a circle with a radius equal to the radius of curvature R_{κ} . Here the horizontal length of the pipeline is considered to be equal to the sum of L_{AB} and L_{BC} , which are both equal to half of the arcs' horizontal length W. The out-of-straightness H is then measured in the middle of the arc. See Figure B.10



Figure B.10: Pipeline arc for curvature calculation using Pythagorean theorem

Finding the radius of curvature of the pipeline using Pythagorean theorem is given below. Using the arc height, which is equal to the out of straightness H, and the base length, which is equal to half of the arcs' horizontal length W, the radius of curvature R_{κ} is found.

$$R_{\kappa}^{2} = \left(\frac{W}{2}\right)^{2} + (R_{\kappa} - H)^{2}$$

$$R_{\kappa}^{2} = \frac{W^{2}}{4} + R_{\kappa}^{2} - 2HR_{\kappa} + H^{2}$$

$$2HR_{\kappa} = \frac{W^{2}}{4} + H^{2}$$

$$R_{\kappa} = \frac{W^{2} + 4H^{2}}{8H}$$
(B.8)

For pipelines coming out of the factory, certain standards for out-of-straightness are set by DNV [25]. These standards are given in Table B.1. For reeling however, no such standard was found concerning the maximum out-of-straightness. This is something that must be discussed with the client, to see what is acceptable. For the Lucius project, the maximum out-of-straightness for a 12*m* long pipe segment is 9*mm*, which gives a minimum residual curvature radius of $R_{\kappa} = 500m$ is used. The 12 m pipe segments are measured for its out-of-straightness iteratively: a segment is measured and when the minimum radius is exceeded, the straightener is adjusted, and another segments is reeled, cut and measured. This process is repeated until the accepted curvature is met, upon which continuous reeling will occur. Simply put, a reeled pipe can either be completely straight ($R_{\kappa} = \infty$), over straightened ($R_{\kappa} > 0$) or under straightened in the inplane bending plane of the pipeline over the straightener. An under straightened pipe is caused when the straightener tracks have not induced enough plastic bending, leading to a remainder of curvature in the pipe. For over straightening, the bending of the tracks of the straightener have been too effective, giving the pipeline a surplus of curvature in the opposite direction. See Figure B.11. Usually the pipeline is under straightened, for it will cause less strain to the system. In practice, a straight pipe is an ambitious goal, which is seldom met.

Table B.1: Out-of-Straightness tolerances for factory pipes set by DNV [25]. These standards do not apply for the out-of-straightness of a pipe segment during reeling



Figure B.11: Left: Under-straightening. Right: Over-straightening

B.6. The lowering of a freely suspended pipeline

The lowering of the pipeline with the attached 1^{*st*} end PLET is done in four phases during the Lucius Project. The phases will briefly be discussed here and have been illustrated in Figure B.12.

Phase I

In Phase one, the 1st end PLET (Appendix E) has been welded to the reeled pipe. The PLET is hanging via the upper tensioner, and a steering winch is connected to the mudmat of the PLET (Appendix E). Afterwards, the PLET and pipe stem are lowered by 17m by paying out the reeled pipe. The so called Pipe and Wire Centralizer is engaged, and the steering winch is pre-tensioned. At a water depth of 150 m the steering winch is finally slack, leading to the disconnecting of the winch.

Phase II

In order to cope with the increasing axial tension in the pipe during lowering, the lower tensioner is engaged. The Pipeline is lowered towards 20m above the seabed, ready for the hook up sling connection.

Phase III

The hook up sling is connected to the start up suction pile using the yoke hook on the PLET (Appendix E). The PLET is gently lowered to the seabed and the mudmat is opened. During this process, the tower angle is change to 85deg.

Phase IV

In the last phase, the PLET stem is lowered onto the mudmat and the normal lay configuration is built up.



Figure B.12: Different phase during pipeline lowering

B.7. PIPELINE CONFIGURATION DURING LAY OPERATIONS

After the pipeline passes the straighteners it is lowered in the sea through the moonpool. During the lowering of the pipeline the tensioner comes into play. It has the essential role of supporting the increasing top tension in the pipeline at the end of the tensioner, whilst maintaining a relatively constant back tension in the system prior to the tensioner. At a certain depth the pipeline will begin to gradually bend from vertical to its final horizontal position on the seabed. The gradual bend is called the underbend, or sagbend. Besides the material and geometric properties of the pipeline, the shape of the sagbend is dependent on the amount of applied horizontal tension. By increasing the horizontal

tension, the curvature along the pipeline will decrease and therefore the steepness of the sagbend slope. In the sagbend the internal moment and curvature increases again, causing strains in the pipeline. During lowering, the hydrostatic pressure on the pipeline increases and can be an important factor for the pipeline design in ultra deep water projects. Typical values for the external water pressure and sagbend strain ϵ are given in Figure B.13. The pipelay tower can be set to an angle as low as 50 degrees from horizontal, which makes installation of larger diameter pipelines in relatively shallow water possible.



Figure B.13: Schematic overview of the pipeline in the sagbend

C

LITERATURE STUDY OF PRIOR PIPELINE TWIST RESEARCH

The occurrence of pipe rotation is not a new phenomena. During S-lay and Reel-lay operations in deep water it is a known fact that the build up of torque can occur. For J-lay however, no reports have been found concerning twist during operations. Although there are theories for its occurrence, it is still difficult to estimate if and how much pipe rotation will occur. S-lay differs from Reel-lay in the way that the pipe is bent plastically at the overbend during S-lay, which does not occur with reel-lay. Although the lay operations share similarities and it is therefore possible to extrapolate relevant information from previously done research, it is important to keep in mind that the lay operations do have distinct differences. In the following chapter previous theoretical research will be discussed concerning the onset of twist during S-lay and Reel-lay operations.

Over the years the maximum lay depth has constantly been increasing. In shallow water, the pipeline wall thickness selection is usually determined by hoop stress due to the internal pressure. For deep water however, other factors play a governing role. Fractures in welds or local buckling with or without resulting collapse of the pipe can be governing failure modes due to combined bending, tension and external pressure during laying. An important consequence of deep water pipelay is that the need of a greater pipe strength capacity has lead to the use of strain criteria beyond the elastic limit of the pipe. And the reason for this change in criteria when looking at S-lay operations is obvious: since the overall size of the pipelay vessel's stinger and it's maximum tension capacity are costly to modify, the curvature of the stinger is the least expensive parameter that can be altered to adapt to deepwater requirements where the pipe must in some cases leave the firing line with almost 90 degrees from the horizontal plane. Small stinger curvatures for S-lay in deep water or the reeling of pipes with reel-lay can lead to plastic strains in the pipeline. Even when the lay configuration is set up to load the pipe within the elastic limit, the effect of inline structures such as valves, tees or buckle arrestors can create local strain concentrations. The residual plastic strain of the pipe can be high, but in most cases will not be noticeable when the pipe is resting on the seabed.

From pipelay experience it appears that the pipeline has a tendency to rotate with S-lay when the permanent curvature induced in the overbend becomes suspended in the sagbend section of the pipe. If a small lateral load is applied to the pipeline (in the form of a current for example), the pipe will experience a torsional moment due to the out-of-plane displacement and the gravitational force. In Damsleth et al. [34] it is mentioned that the rotation of a suspended pipe with S-lay is proportional to the plastic strain, the suspended length and the ratio D/t of the pipeline. Even a small value of permanent curvature in the pipeline (0.02% plastic strain) can cause enough rotation during installation to make it difficult to access inline components or can lead to the necessity of torque removal measures. Although the rotation can be severe, Endal et al. [33] concludes that the pipeline roll does not give any on-bottom instabilities of the as-laid pipe. Also, no reports have been found stating that the pipeline twist led to the deterioration of the structural integrity of the pipeline itself.

C.1. PIPELINE TWIST RESEARCH DONE BY GEIR ENDAL

Geir Endal, advisor pipeline technology at Statoil, has written about the twist phenomena during laying in several papers spanning from 1995 tot 2015 [33][10][1][49][50]. Several observations can be made from them, and his findings will briefly be discussed in this chapter.

C.1.1. TWIST ANALYSIS OF A GLOBAL BUCKLING MITIGATION DURING REEL-LAY

An important design concern for designing offshore oil & gas pipelines is the mitigation against thermal buckling. Pipelines under high pressure and temperatures will be exposed to axial compressive forces, which can lead to excessive lateral movement at a single location. The pipeline may form a localized kink with high curvature, also known as the phenomena global pipeline buckling. At the pipeline buckles, the stress & strain criteria may be exceeded, leading to possible undesired consequences such as pipeline collapse or rupture. The mitigation method of intermittently adding residual curvature during the reel-lay process was introduced by Endal and was patented by Statoil in 2002. With the

mitigation method came an investigation into the twist behavior of the pipeline.

By adjusting the reel ship's upper track of the straightener system during installation of the pipeline, local, in-plane residual curvature can be created. The result can be seen in bottom-right in Figure C.2. According to the patent of Statoil[50], the typical configuration would be to impose 0.20%-0.25% residual strain over a 40 m length every kilometer during reel-lay installation. As with the Aegir, the straightener system on the reel ship works via 3-point bending and by intermittently adjusting the straightener during laying the residual curvature can be applied. The residual curvature gives the pipeline locally less stiffness so that extension in axial direction due to temperature may occur in a distributed and controlled manner, causing pipe deflections without producing large compressive forces. If the straightener equipment of the 'Seven Oceans' is taken as an example, residual curvature can be added to the pipeline by adjusting the orientation and position of the upper track. By increasing the upper deflection of the upper track while keeping the lower deflection constant, an additional curvature is added to the pipeline. By subsequently decreasing the upper deflection and increasing the lower deflection of the track, the additional curvature is topped off and the pipeline continues to be laid in its normal curvature without residual curvature. See Figure C.3 and Figure C.1.



Figure C.3: Straightener components of the Seven Oceans[32]

Figure C.4: Schematic depiction of residual curvature in pipeline

The tendency to pipeline twist due to (intermittent) residual curvature can be estimated by a simplified analytical energy approach. The minimum total potential energy principle is a fundamental concept which briefly entails that a structure or body shall deform or displace to a position that minimizes the total potential energy of the system. For example, a horizontal beam loaded by additional weight will bend to a lower position, a position where the total potential energy is at it's minimum, and therefore the configuration is stable. The same principle is applied to the bending of the pipeline in the sagbend. Here, the overall strain energy $U_T(\phi_0)$ of the pipeline is used to estimate the pipeline twist. The total strain energy $U_T(\phi_0, \kappa_r)$, which can also be seen as the total amount of work $W_T(\phi_0)$ as the pipeline goes through the sagbend, is assumed to consist of a bending and a roll contribution. Both strain energies are dependent on the final torsion angle of the pipeline: see Equation C.1

$$U_T(\phi_0,\kappa_r) = U_B(\phi_0,\kappa_r) + U_R(\phi_0,\kappa_r)$$
(C.1)

The bending strain contribution $U_B(\phi_0, \kappa_r)$ is dependent on the curvature that the pipeline has in the sagbend. This curvature can be approximated by for example the natural catenary, giving us the curvature $\kappa_n(s)$ along the length of the pipeline. Furthermore, the residual curvature as described earlier, can be added to the pipeline curvature equation:

$$\kappa_T(s,\phi_0,\kappa_r) = \kappa_n(s) + \kappa_r \cdot \cos\left(\phi\left(s,\phi_0\right)\right) \tag{C.2}$$

The effect of a segment of pipeline with residual curvature on the total curvature of the pipeline in the sagbend is schematically illustrated below in Figure C.5. Here it is evident that at different torsional angles the curvature of the pipeline changes. The bending strain of the pipeline in the sagbend will therefore be higher for certain torsional angles, and lower for others.



Figure C.5: Schematic effect of κ_r on κ_T at different angles of rotation

Here $\phi(s, \phi_0)$ is the roll angle along the suspended section of the pipe, which is dependent on the roll angle at the touchdown point at the seabed ϕ_0 and the distance along the pipeline *s*. The roll angle increases from the vessel towards the seabed, assuming that the pipeline is constrained against twist at the surface and completely free at the bottom. Evidently, the maximum roll angle ϕ_0 occurs at the bottom. The torque therefore has its maximum at the sea surface where the pipeline is constrained by the tensioner, and the torque is assumed to be zero at the seabed. The development of the roll angle of the pipeline has three boundary conditions:

$$\phi(0) = \phi_0$$
$$\frac{d\phi}{ds}(0) = 0$$
$$\phi(L) = 0$$

Here, *s* is the distance along the pipeline, where s = 0 is located at the seabed, and s = L is located at the sea surface. The twist angle, which begins at its maximum angle at the seabed and decreases to zero at the sea surface, is given by the following second order polynomial:

$$\phi(s,\phi_0) = -\frac{\phi_0}{L^2} \cdot s^2 + \phi_0 \tag{C.3}$$

The bending strain contribution of the total work is given as:

$$U_{B}(\phi_{0},\kappa_{r}) = \begin{cases} \int_{0}^{L_{c}} E \cdot I \cdot \left[\kappa_{n}(s) + \kappa_{r} \cdot \cos\left(\phi\left(s,\phi_{0}\right)\right)\right]^{2} ds + \int_{L_{c}}^{L} E \cdot I \cdot \kappa_{n}(s) ds, & \text{if } L_{c} \leq L \\ \int_{0}^{L_{c}} E \cdot I \cdot \left[\kappa_{n}(s) + \kappa_{r} \cdot \cos\left(\phi\left(s,\phi_{0}\right)\right)\right]^{2} ds & \text{otherwise} \end{cases}$$
(C.4)

In practice, the residual curvature is added intermittently over sections of pipeline. For simplification, the residual curvature applied over the intermittent sections are bundled into an equivalent length of pipeline containing the same amount of curvature. This bundled L_c with residual curvature is used in the calculation of the bending strain energy U_B .

$$L_c = \sum_{n=1}^M L_{\kappa_r}(n) \tag{C.5}$$

(C.6)

It is assumed that the intermittent residual curvature sections are of equal length, and all have the same amount of residual curvature:

$$L_{\kappa_r}(n) = L_{\kappa_r}(n+1)$$
 for $n = 1...M$ (C.7)

For the torsional moment, the contribution to the total energy is as follows:

$$U_R(\phi_0) = \int_0^L M_\phi(s,\phi_0) \cdot \frac{d}{ds} \left(\phi\left(s,\phi_0\right)\right) ds$$
(C.8)

Which is equivalent to,

$$U_R(\phi_0) = \int_0^L G \cdot I_T \cdot \left[\frac{d}{ds} \left(\phi\left(s, \phi_0\right) \right) \right]^2 ds$$
(C.9)

By finding the minimum potential of the combined contributions, the amount of pipe twist rotation can be estimated.

$$\frac{d}{d\phi_0} \cdot U_T(\phi_0, \kappa_r) = 0 \tag{C.10}$$

The methodology can be viewed in detail in Endal et al. [1]. A schematic overview of the analytical approach has been made and can be seen in Figure C.7. In the same paper the total work of a 16" inch pipeline during lay operations is analytically approximated with varying values of residual curvature lengths. Curvature lengths L_C of 0m, 50m, 70m and 100m are used with a residual strain of 0.2% in the curvatures. The results can be seen in Figure C.6. A few conclusions were made about the total works graphs:

- When there is no residual curvature in the pipeline, the total work increases monotonically against the roll angle: no trough is visible in the graph. Consequently, the pipeline has a resistance towards roll, for there is no energetic benefit of twisting.
- For short residual curvature lengths the graph is relatively flat. The point of minimal total work against the roll angle is not too defined, which can make the pipeline relatively unstable: it may easily rotate to a roll angle with a higher total work.
- For higher residual curvature lengths, the graph has a clear minimum point of total work at a specific roll angle. The steep trough in the curve indicates that pipe twist will easily be initiated.



Figure C.6: The total work for a 16" pipeline for different residual curvature lengths (0.2% local strain) against the roll angle at touchdown (360m waterdepth) (add source)



Figure C.7: Schematic overview of shape functions for the simplified analytical approach estimating pipeline roll (add source)

C.1.2. TWIST ANALYSIS DURING S-LAY OPERATIONS

As mentioned before, twist has been known to occur during S-lay in deep water. Back in 1995, Endal did research on the relation between plastic pipeline deformation in the overbend and the occurrence of twist (Endal et al 1995)[33]. For the prediction of the pipeline twist, Endal used both the analytical approach of energy minimization as mentioned in the previous section and a numerical approach using the FE computer program Abaqus. The findings of his research will be briefly discussed.

The analytical approximation of the pipeline twist using energy minimization is similar to the method mentioned in the previous section. Only during the S-lay operations, the residual curvature is not added purposefully nor intermittent: it is dependent on the residual strain of the pipeline induced by the radius of curvature that the stinger has during operations. The nominal curvature κ_n and roll angle ϕ along the pipeline have been approximated with different functions in comparison to the previous section. The functions can be found in Endal et al 1995. For the further calculations, the same equations have been used as in the previous section: from Equation C.4 until Equation C.10.

For the numerical modeling approach the general non-linear FE-program ABAQUS in Endal et al (1995) is used. Different models along with a thorough sensitivity study to investigate the effect of plastic strain, waterdepth, tension etc.

Effect of plastic strains induced by stinger

A model is made to approximate the S-lay operation via static 3D simulation to investigate the influence of plastic strain and pipe tension on the roll angle. When looking at the results from Endal et al. (1995) [33]it can be seen that the decrease of the stinger radius (and therefore an increase of the pipe curvature at the overbend) lead to an increase in the total amount of pipe twist. In the paper two different diameter pipes are subjected to varying stinger radii and are analyzed at different depths. The results for the 20" pipe can be seen in the Table C.1. It is clear that the amount of pipe twist increases rapidly for early increasing increments of the stinger radius, and thus an increase of the amount of residual curvature, and that the increase slows down in the later increments. This would indicate that the twist phenomena is a typical instability problem, where once the roll is initiated it will increase rapidly, up to the point where it comes close to equilibrium, in which case it will start to converge.

Analysis			Output				
	Depth	R [m]	r/R [%]	T _{b,true} [kN]	$T_{b,eff}$ [kN]	T _{barge} [kN]	θ [deg]
1	300m	69.0	0.306	-333	103	294	116
2		72.0	0.293	-333	103	295	95.0
3		75.0	0.281	-333	103	295	48.0
4		76.0	0.278	-333	103	296	11.5
5		78.1	0.270	-333	103	296	2.2
6	400m	70.3	0.300	-483	98	346	137
7		74.0	0.285	-460	121	372	110
8		78.1	0.270	-437	144	396	52.0
9		82.0	0.257	-425	156	410	1.2

Table C.1: Results numerical model test of 16" pipe

In the paper Endal et al. (1995) [33] the development of the roll angle over the amount of installed pipes is given for analysis number 6 of Table C.1. The development can be seen in Figure C.8. A length of pipeline is installed of approximately 1300m at a waterdepth of 400m. After installation of 470m of pipeline, the cross-section which was initially placed on the stinger has passed the inflection point and is now in the sagbend of the lay (about 80 m from touchdown). Here it appears that the pipe twist is initiated: the plastic strains developed at the overbend are causing a combination of bending and rotation of the pipeline in the sagbend. It can be seen that once the onset of pipe twist occurs, it increases rapidly in the beginning, followed by a asymptotic decrease of the increase of the roll angles. When looking at the graph, it appears that there might be a maximum roll angle that can occur, but this cannot be concluded with certainty.



Figure C.8: The development of the maximum roll angle during installation simulation of 16" pipe [33].

Effect of tension

Another parameter that was investigated in the paper Endal et al. (1995) [33] was the effect of pipe tension on the pipe rotation. Two pipes with the values as given in Table C.2 were studied on the development of the roll angle against the effective bottom tension. The results can be seen in Figure C.9. The roll angle increases fast for both pipes from almost zero at a tension of 0.02 MN to maximum roll angles at 0.04 MN (pipe 1) and at 0.08 MN (pipe 2) bottom tension. Both pipes share the same dependency concerning the bottom tension. The difference between them is in the decrease of the roll angle after the maximum roll angle is reached: for pipe 1 the roll angle decreased to approximately zero with increasing bottom tension, but for pipe 2 the decrease was far less. This can be explained by the larger amount of submerged weight and the larger strains over the stinger for pipe 2.

Table C.2: Add caption

Pipe	D ["]	D/t	Steel	R [m]	ϵ_{nom} [%]	Depth [m]	W_s [N/m]
1	16	20.6	X65	78.1	0.27	400	584
2	25	35	X60	105.8	0.3	400	800



Figure C.9: The roll angle as function of the Tensile Force

Two important, opposing relations regarding pipe tension dependency on rotation are discussed:

- Increased pipe tension results in a decreased sagbend curvature. The amount of bending strain in the sagbend would therefore reduce, possibly causing the pipeline to have less energetic advantage by twist. Ergo, the amount of twist would reduce as the amount of tension is increased.
- Increased tension results in a lower torsional resistance due to a longer suspended spanlength: the touchdown point is horizontally displaced and the amount of torsional unconstrained pipeline is increase. This would mean that the pipe twist will as the tension is increased.

The contradictory relations between tension and twist can possibly be used to explain the progression of the graphs in Figure C.9. At very low pipe tension, there will be a very short suspended length and the pipeline will very suddenly go from vertical to horizontal. It is stated that the possible deformation controlled restrained forces that correspond to instability conditions will not arise at this geometric configuration, and so reversed bending takes places without the onset of roll. When the tension is increased, a proper sagbend in the pipeline is formed, with high bend strains in the sagbend. This along with the effect of lower torsional resistance when the tension is increased give the fast increase of the roll angle in the beginning. After the maximum roll angle is achieved, the decrease in the sagbend curvature with increasing tension becomes predominant, and the pipe twist decreases again.

Tension also has an effect on on-bottom instabilities. Particularly for lower tensile forces, the lateral resistance of the seabed will become more prominent, for the normal force of the pipeline at the touchdown area will increase. Also the effect of bottom topography in combination with tension should be taken into account. High (residual) horizontal tension develops both larger and more frequent freespans, making the pipeline more prone to twist.

Effect of geometric properties

In the same paper Endal et al. (1995) [33] another model is made of the pipeline. The pipe is divided into 2D beam elements where each beam element has two nodes with the vertical displacement and the rotation as nodal degrees of freedom. The more exact details of the model are not of importance. In one of the tests, the diameter of the pipeline was varied to see its effect on the variation of the total amount of roll in the pipeline. In the graph of Figure C.10 it can be seen that the pipeline roll decreases in a linear fashion when the diameter is decreased. During the tests, the D/t ratio is kept constant.



Figure C.10: Roll angle as Function of the Pipe Dimension [33].

A decrease of the pipe diameter with a factor of 2 leads to an increase of the bending stiffness with a factor of 16. This increase in stiffness has, similar to the increase of tension, two opposing effects on the pipeline twist.

- Increasing the stiffness of the pipe leads to a decrease of the curvature along the suspended pipe. The decrease of curvature along the pipeline will result in a reduction of the strain in the sagbend, and possibly therefore the amount of twist occurring.
- On the contrary, increased stiffness will lead to a longer suspended pipeline length. This length increase will result in a decrease in the torsional rigidity and resulting in a pipeline more prone to twist.

When looking at the results of Figure C.10 it seems that the governing relation is the former: the decreased curve has an inhibiting effect on pipeline rotation.

C.2. PIPELINE TWIST RESEARCH DONE BY DAMSLETH ET AL.

In the paper Damsleth et al [34], the emphasis is laid on the contribution of potential energy on the pipeline twist. When a pipeline is plastically deformed over the stinger at the overbend, the section will hang vertically higher upon arrival at the sagbend than in comparison with a pipeline section that has deformed elastically. The naturally upward convex shape of a plastically deformed pipe segment during S-lay would lead to the seeking of an energetic equilibrium via twist at the reverse bending in the sagbend. Ergo, the pipeline would 'fall' to a lower energy potential through twist, until a balance is found between the torsional, potential en bending strain energies. For the elastically deformed pipe the lowest energy potential is already met and therefore no onset of twist would occur for there is simply nothing to gain energetically. It is therefore according to Damsleth et al. [34] reasonable to conclude that the potential energy reduction mechanism underlies pipeline rotation during laying. To validate the assumption that plastic strain in the pipeline causes the onset of pipe rotation, 3D models have been made in Damsleth et al.[34]. Three simple models all representing a pipe with a certain length and D/t ratio are used, fixed at one end, and pinned at the other where a horizontal sliding condition is specified. The models span horizontally, where both ends are at equal height vertically. In two steps, forces are applied to the model: firstly a horizontal force corresponding to a sea current of 0.5 m/s is applied perpendicular to the in-plane bending plane. Secondly, the submerged weight of the pipeline is applied, along with an axial horizontal tension representative to a situational comparative lay tension. The three different models are:

- A Straight pipe
- Pre-curved 'overbend' pipe , R = 571m
- Pre-curved 'sabend' pipe, R = 571 m

The displacement and the rotation at the middle of the pipe are studied and compared for the different models. For the overbend pipe model, the pipe has been pre-curved and represents a pipe that has been plastically bend over the overbend of a stinger to obtain a 0.1% residual strain. The final equilibrium configurations represent the pipeline in the underbend section during pipelay, where the pipe is subjected to its submerged weight and axial tension. For the underbend model, the pipe represents a naturally stable case where gravity has caused the underbend configuration. Also in the second model, submerged weight and axial tension is applied to obtain the final equilibrium configuration. See Figure C.11.



(a) The pre-curved overbend model in its free and loaded conditions [34].



Figure C.11

The graph of the twist in the midpoint is given in Figure C.12, in which the rotation is plotted against the lateral horizontal force. It can be seen that the underbend pipe is subjected to negligible rotation, in the order of 1 percent for the maximum applied lateral force. The straight pipe shows no rotation, as expected. Lastly, for the overbend pipe we see a clear onset of rotation, up to 17 degrees. The following explanation is given in the paper: the rotation tries to 'tip over' the residual curvature from the overbend shape into the underbend shape in the effort to minimize its potential energy. Eventually an equilibrium will be met between the increase in potential energy due to rotation and the corresponding decrease of potential energy of the residual curvature. The increase of the rotation angle is linear to the increase of the

lateral force, indicating that the pipe twist phenomena is a stable process, and not an instable process as indicated in Endal et al. [33]. As a reader of the article, I would like to comment on the conclusions made: the bending strain for a pipeline bending from a pre-curved underbend model to a configuration with tension and submerged weight, is less in comparison to an pre-curved overbend model. In the overbend model, not only the magnitude, but also the direction of the curvature along the pipeline changes upon loading with self weight and equivalent lay tension. This difference in bending strains during the analysis and possibly thereafter could be of influence on the results found.



Figure C.12: Rotation at mid-span subject to a lateral force [34].

The development of the maximum roll angle during installation simulated in Endal et al. [33], is repeated in Damsleth et al. [34] for a stiffer pipe (D/t ratio of 39). Here, the onset of pipe rotation is slower and eventually after a few kilometers of pipelay, the amount of twist in the pipeline as observed at the rotationally free pipeline end seems to have converged (see C.13(c)). The figures below illustrate the twist phenomenon during laying of 2.4 km section of deep water pipeline with a lateral current of 0.5 m/s. Sadly, the depth at which the lay operations take place is not mentioned, but given the subject of the article it can be assumed that deep water is used in the simulation. The pipeline end is free to rotate and rotational friction is ignored. C.13(a) shows the total strain over the pipeline and permanent residual strain after the pipe has been subjected to elasto-plastic bending at the overbend. The torsional moments for the elasto-plastic and elastic bended pipelines are given in C.13(b). C.13(c) shows the rotation of the pipeline (max of 60 degrees) subjected to plastic residual strain and of a pipeline subjected solely to elastic bending (no rotation).



(a) Total axial strain and plastic strain in an elasto-plastic pipe from the free end on the seabed to the tensioner on the lay vessel [34].

(b) Torsional moment in the pipeline from the free end on the seabed to the tensioner on the laybarge [34].



(c) Axial rotation of the pipeline from the free end [34].

Figure C.13: Development of axial strain, moment and rotation during a s-lay installation simulation

D

PIPELINE FABRICATION

For the oil and gas industry, there are four types of fabrication routes that can be used to produce pipe.

- · Seamless welding
- Longitudinal welding using a submerged arc
- · Longitudinal welding by electrical resistance
- · Helical welding

Seamless welding

For the Lucius project, seamless pipes were used to create the pipe stalks. Seamless pipes are created by hot working a solid rod to form a pipe without any longitudinal welds. The two main methods which are commonly used are the Mandrel Mill and the Mannesman Plug Mill process. In both seamless pipe processes the pipes start out as a solid round rod called a billet, which is heated in a furnace. Afterwards the heated billets are rotationally driven through the pinch zone of two rollers which are also rotating. The rotary motions of the billet and rollers leads to a reduction of the stress needed for the piercer between the rollers to penetrate the rod and to shape it into a pipe. The first piercing step produces the primary pipe shape. Afterwards the pipe is brought to a finish in a number of steps, dependent on which process is used. The exact steps of the Mandrell Mill process can be seen in Figure D.1 and the Mannesman Plug Mill process is given in Figure D.2.



Figure D.1: The seamless Mandrill Mill pipe fabrication process[35]



Figure D.2: The seamless Mannesman Plug Mill pipe fabrication process[35]

Longitudinal welding using a submerged arc

A Longitudinally welded pipe using a submerged arc is also known as a U-O-E pipe, after its manufacturing steps. Individual steel plates are first bent into a U shape, then pressed into a tubular shape (O) and finally after longitudinal welding, the pipe is expanded (E) to ensure its circular shape. The longitudinal weld is produced using submerged arc welding (SAW).

Longitudinal welding by electrical resistance

Welding by electrical resistance is a continuous process where a pipe is formed from a coiled steel plate. The plate is uncoiled, sheared to the appropriate length, flattened and then processed to a tubular shape by a sequence of rollers. The longitudinal seam is then welded by electrical resistance welding.

Helical welding

A coil of hot coiled plate is uncoiled, straightened, flattened and then wound up helical to form a pipe. The internal helical seam is then welded using inert gas welding or SAW at first, and as the seam rotates to the top position, the external weld is made. The process is continuous.

E

1ST END PLET PF-01-A





F

PIPELINE LENGTH APPROXIMATION

As seen before from the numerical computations on the pipeline in sagbend formation, the length of the suspended pipeline is the key unknown parameter in the calculations beforehand. It is therefore useful to approximate the expected pipeline length (and the corresponding pipeline configuration) before the actual calculations. It is known that the length of the suspended pipeline is related to the top tension T_0 , the flexural rigidity EI, the self weight W_s and the top angle θ_0 . According to Buckingham's Pi-Theorem [11], a dimensionless function can be derived that is defined as follows:

$$\frac{W_s L}{T_0} = f\left(\frac{EIW_s^2}{T_0^3}, \theta_0\right) \tag{E1}$$

Equation E1 has been determined by solving the boundary value problem for the governing equations Equation 2.19 and Equation 2.21 and the previously mentioned boundary conditions. The system has been solved extensively within the range of:

$$\begin{array}{rrrr} 0 < & \pi_1 & \leq 500 \\ \\ \frac{76}{180} \pi < & \pi_2 & \leq \frac{90}{180} \pi \end{array}$$

Where π_1 is assumed to be equal to $\frac{EIW_s^2}{T_0^3}$ and $\pi_2 = \theta_0$. Using these results the approximate formula for π_L , which is equal to $\frac{T}{W_s L}$, is obtained via mathematical fitting. This is shown in Equation E2 and Equation E3.

$$\pi_L = -0.4085\pi_1^{0.4} + 5,16\pi_1^{0.5} - 3.895\pi_1^{0.6} - 0.773\pi_2^{-0.9793} + 1.5435$$
(E2)

For,

$$\left(0 < \pi_1 \le 1, \frac{76}{180}\pi < \pi_2 \le \frac{90}{180}\pi\right)$$

And,

$$\pi_L = -0.124\pi_1^{-1.011} - 0.773\pi_2^{0.9793} + 2.526 \tag{F3}$$

For,

$$\left(1 < \pi_1 \le 500, \frac{76}{180}\pi < \pi_2 \le \frac{90}{180}\pi\right),$$

Once the value for π_L is known, the length of the pipeline can be approximated by computing:

$$L_{approx} = \frac{\pi_L T_0}{W_s}$$

By varying the top tension T_0 within a range expected for laying at certain waterdepths, a range of pipeline lengths L_{approx} together with the corresponding coordinates of the pipeline can be computed. Hereby one must realize that the z-coordinate at s = L is equal to the waterdepth during the lay operations. By cross referencing the desired Top tension and waterdepth for a pipelay operation, the value of the pipeline length *L* can be faster estimated via the approximation equation.

G

METOCEAN DATA KEATHLEY CANYON BLOCK 875 & 874

Table 4-5: 1year return current profile

Depth	Speed	100%	Speed 70%		
[m]	[m/s]	[knots]	[m/s]	[knots]	
0	1.11	2.16	0.78	1.51	
25	1.11	2.16	0.78	1.51	
50	1.10	2.14	0.77	1.50	
75	1.01	1.96	0.71	1.37	
100	0.86	1.67	0.60	1.17	
125	0.77	1.50	0.54	1.05	
150	0.71	1.38	0.50	0.97	
175	0.65	1.26	0.46	0.88	
200	0.59	1.15	0.41	0.80	
225	0.53	1.03	0.37	0.72	
250	0.49	0.95	0.34	0.67	
275	0.45	0.87	0.32	0.61	
300	0.41	0.80	0.29	0.56	
325	0.39	0.76	0.27	0.53	
350	0.36	0.70	0.25	0.49	
375	0.34	0.66	0.24	0.46	
400	0.32	0.62	0.22	0.44	
425	0.30	0.58	0.21	0.41	
450	0.28	0.54	0.20	0.38	
475	0.27	0.52	0.19	0.37	
500	0.25	0.49	0.18	0.34	
525	0.23	0.45	0.16	0.31	
550	0.22	0.43	0.15	0.30	
575	0.21	0.41	0.15	0.29	
600	0.20	0.39	0.14	0.27	
800	0.20	0.39	0.14	0.27	
2000	0.20	0.39	0.14	0.27	



[51]

ANADARKO PETROLEUM CORPORATION METOCEAN CRITERIA FOR KEATHLEY CANYON BLOCK 875 & 874





2.7 Current Profile Characterization

Figure 2-41 Characteristic Current Profiles 1 to 36

Fugro GEOS/C56309/6067/R10

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Figure 2-42 Characteristic Current Profiles 37 to 72

Fugro GEOS/C56309/6067/R10

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Figure 2-43 Characteristic Current Profiles 73 to 108

Fugro GEOS/C56309/6067/R10

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Figure 2-44 Characteristic Current Profiles 109 to 144



Figure 2-45 Characteristic Current Profiles 145 to 180

Fugro GEOS/C56309/6067/R10



Figure 2-46 Characteristic Current Profiles 181 to 216



Figure 2-47 Characteristic Current Profiles 217 to 252

Fugro GEOS/C56309/6067/R10



Figure 2-48 Characteristic Current Profiles 253 to 288



Figure 2-49 Characteristic Current Profiles 289 to 324

Fugro GEOS/C56309/6067/R10



Figure 2-50 Characteristic Current Profiles 325 to 360

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Figure 2-52 Characteristic Current Profiles 397 to 432



Figure 2-53 Characteristic Current Profiles 433 to 468

Fugro GEOS/C56309/6067/R10



Figure 2-54 Characteristic Current Profiles 469 to 504



Figure 2-55 Characteristic Current Profiles 505 to 540

Fugro GEOS/C56309/6067/R10



Figure 2-56 Characteristic Current Profiles 541 to 549

Fugro GEOS/C56309/6067/R10

Η

INSTALLATION METHODS

For the installation of pipelines there are several methods that can be used. The principal ones used in the offshore industry are given below:

- S-lay
- J-lay
- Reel-lay
- Towing of pipeline bundles

H.1. S-LAY

Individual pipe sections are stored inside the pipelay vessel and are transported with an intricate system of pipe transfer cranes and conveyors to the line-up carriages. Once a pipe section has come to the line-up carriage, it will pass through a number of stations before entering the water. The first process is the beveling of the pipe ends: this is a process where an angle is formed between the edge of the end of a pipe and a plane perpendicular to the surface. After the beveling process, the individual pipe is lined up with the rest of the pipeline. The welding process can now commence. Typically, there are five welding stations through which the pipeline will pass. Once the welding stations have been passed, it is time for the pipe to undergo Non Destructive Testing (NDT). Typically this is one station and here the pipe welds are screened for any possible flaws. In the final station(s), field joint coating is applied to the section between the the welded pipes, also know as the pipe joint. After the pipeline has passed all these stations, it is lowered into the water via the so called stinger of the vessel. The stinger supports the bend of the pipe, also known as the overbend. Depending on the pipe diameter and the water depth, the stinger can be adjusted in radius (which influences the curvature of the pipe) as well as in departure angle. It contains a set of rollers which supports the pipe while it is being guided to the seabed. The pipeline situated in the stinger is known to be in the overbend. Following the pipeline downwards, there comes a point where the curvature of the pipe is switched from one direction to another. This point is known as the inflection point. After the inflection point, the pipeline has reached the underbend or also called, sagbend. Here, the pipeline has opposing curvature when compared to the overbend. At the end of the sagbend the pipe is situated on the seabed, where it can be connected to subsea modules.



Figure H.1: S-lay vessel layout

There are two extreme stinger configurations, and everything between these two. The first one is for shallow water, where the stinger is relatively flat (has a small departure angle). The stinger will have a large radius, typically 200-300 meters, and the tension in the pipe is determined by the sagbend. For the second configuration the stinger is set up for deep water operations. It is highly curved and the departure angle is large. Ergo, the curvature radius must be small, in the order of 70-100 meters. The tension in the pipe is in this case determined by a combination of the sag bend and the vertical load.

An important part of the S-lay principle is the tensioner. Depending on the size of the vessel, there can be one or more present in the line-up carriage. The principle function of the tensioner is to maintain tension in the pipeline in order to avoid buckling. It incrementally lets through pipe sections (typically 12-24 m) and it compensates for ship motions.

H.2. J-LAY

The difference between J-lay and S-lay can already be deduced by its name: Where during the S-lay the pipeline goes through an overbend at the stinger and an underbend near the bottom giving the pipeline an S-shape, for the J-lay procedure the pipeline leaves the firing line vertically and finally reaches the bottom after a sagbend (the pipeline is laid in a J-shape). The J-lay process is more concentrated in its overall procedure when compared to S-lay. The pipes are beveled at the beveling station usually located on deck. Afterwards the pipe transfer cranes transports an individual pipe to the welding tower. Here the pipes are lined-up, welded together and afterwards lowered further downwards in the welding tower. In the tower, NDT is done and field joint coating is applied. Once the coating is applied, the pipeline is lowered either at an angle or vertically downwards, where it will eventually come into the sagbend and afterwards lie horizontally on the seabed. The tensioner(s) for this lay procedure have the same functions as for the S-lay, but are know in a vertical configuration in the J-lay welding tower. This type of procedure does not have a stinger with which it can change the curvature of the pipe or the departure angle. It is only possible to change the departure angle for the J-lay operation, by tilting the entire tower.



Figure H.2: J-lay vessel layout

H.3. TOWING OF PIPELINE BUNDLES

For this installation method, the pipelines are built on land in long pipe stalk sections. These stalks are launched in the water and remain buoyant via buoyancy modules or via a carrier pipe sheathed around the bundle. These stalks are then towed to the designated site using one of the following towing methods:

Surface tow: For this type of tow, a tug tows the pipe on top of the water. Here either buoyancy modules or the carrier pipes ensures that the pipeline is kept on the water's surface.

Mid-depth tow: Here less buoyant modules are used compared to the surface tow. The forward speed of the tug boat is used to keep te pipeline at a submerged level. Once the forward motion of the tug ceases, the pipeline will settle on the sea floor.

Off-bottom tow: Here a combination of buoyancy modules and chains for added weight are used, working against each other to keep the pipe stalks just off the seabed. When the destination is reached, the buoyancy modules are removed and the pipe settles to the sea floor.

Bottom tow: For the last method, no buoyancy modules are used. The pipe stalks are simply being dragged along the sea bed and the tug pull operation ceases when the pipe has arrived at the desired location. This type of tow is reserved solely for shallow-water installation where the sea floor is soft and flat.

H.4. REEL-LAY

With reel-lay, the pipelay is decomposed into the manufacturing of long pipeline stalks onshore and the actual pipelay offshore. Onshore, pipeline stalks are made in the fabrication yard. These stalks can then be either spooled on a reel aboard the vessel directly, or they can be spooled onto a detachable spool onshore and afterwards transported to the lay vessel. On the reel-lay vessel, the pipeline is guided through the so called aligner wheel, located in the pipelay tower. The pipeline with residual curvature from being spooled onto the reel drum, is now bent over the aligner wheel, obtaining the curvature radius of the aligner wheel. In theory, all curvature is subsequently removed at the straightener section of the pipelay tower. Here the pipeline is reverse bent to remove any residual curvature in the pipeline. The straightened pipeline is then guided through the tensioner section, where tension is applied to the pipeline. Finally the pipe is lowered into the water through the firing line. The reel construction can either be followed by S-lay (see Figure H.4) or by J-lay (see Figure H.3) and usually the angle of the lay tower relative to horizontal can be changed.



Figure H.4: Reel-lay/S-lay vessel layout

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