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Trawl Gear interaction with Subsea Pipelines

Master Thesis

Marine and Subsea Technology

Berhane Yohannes

Spring 2012

ABSTRACT

This master thesis is written at the University of Stavanger in collaboration with IKM Ocean Design.

The presence of both the fishing industry together with the oil and gas industry offshore results in development of methodologies and techniques on how to exercise offshore activities. It is known that problems can develop when trawl gear interacts with subsea structures like subsea pipelines, manifolds, wellheads, cables and others. Problems resulting from the interaction of trawl gear with subsea structures include safety of vessels, damage to subsea structures and fishing equipment as well as poor communication between the two industries.

One of the most severe design cases for an offshore pipeline system is when there is interaction of fishing gear with a pipeline. Therefore it is important to further understand the behavior of the trawl equipment. Fishing gear weight and velocity as well as pipeline conditions like wall thickness, diameter, coating and flexural rigidity are basic parameters that need to be considered in order to understand the damage to the pipeline and fishing gear during the interference. Realistic description of load level and time history for interaction with a pipeline configuration on the sea bed including free span and pipeline stiffness shall be taken into account during the analysis of the interaction.

The main object of this thesis is to determine pullover loads from commonly and recently used trawl gears on submarine pipelines by using FE assessments. Resulting loads are compared with DNV code recommendations. We contribute towards reducing the conservatism in the design curves through this as we shed light on the expected loads from relatively new trawl gears such as with roller type clump weights. Parameters as trawl gear type, pipeline and seabed soil are investigated in order to understand the significance of the variables and hence derive appropriate design curves based on the significant parameters for DNV-RP-F111 comparisons.

All simulations in this thesis are performed by means of the computer software SIMLA.

Keywords: Trawl gear, Pipeline, SIMLA, Finite element analysis.

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Lastly, I offer my regards and blessings to my beloved wife Saba Tsahaye and children, for their love, patience, encouragement and support for all aspects until the completion of the project.

Berhane Yohannes.

Stavanger, June 14, 2012.

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Nomenclature

List of symbols

A_s	Cross sectional area of steel pipe
c	Steel pipeline centroid
C_D	Drag coefficient
C_F	Coefficient of pullover force
C_h	Span height correction factor
C_I	Inertia coefficient
C_L	Lift coefficient
C_T	Coefficient of pullover duration
D_i	Internal diameter
D_o	Outside diameter
E	Young's modulus of the steel pipe
F	Residual axial tension
f	Friction factor between the pipeline and seabed bottom
F_{endcap}	End cap force
F_f	Friction force
F_h	Horizontal force (drag and inertia forces)
f_n	Natural frequency of span
F_O	Post buckle axial force
F_p	Maximum pullover load on pipe in horizontal direction
f_v	Vortex shedding frequency
F_V	Vertical lift force
F_z	Maximum pullover load on pipe in vertical direction
$\overline{H}$	Dimensionless height
H_{sp}	Span height, measured as seabed to pipeline gap
I	Moment of inertia
k_b	Trawl gear bending board stiffness
k_i	Trawl gear in plane stiffness

k_w	Warp line stiffness
M	Bending moment
m_a	Hydrodynamic added mass and mass of entrained water
m_t	Board steel mass
P	Operating pressure
P_{ecr}	Elastic critical pressure
P_i	Internal pressure
P_o	External pressure
R	Mean radius
S_t	Strouhal number
t	Pipe wall thickness
T_p	Pullover duration
V	Trawling velocity
V_r	Reduced velocity
W	Submerged weight of pipe
θ	Operating temperature rise
α	Coefficient thermal expansion
δ_p	Displacement of the pipe at the point of Interaction
Δp	Pressure difference between the internal and ambient seawater pressures
σ_{eq}	Equivalent stress
ε	Pipeline strain
γ_{st}	Factor of safety
ρ	Density of water
μ_a	Coefficient of axial soil friction
μ_l	Coefficient of lateral soil friction
σ_h	Hoop stress
σ_l	Longitudinal stress
ν	Poisson’s ratio

Abbreviations

API	American Petroleum Institute
DIN	Deutsches Institut für Normung (German Institute for Standardization)
DNV	Det Norske Veritas
FEA	Finite Element Analysis
NPD	Norwegian Petroleum Directorate
OD	Outside diameter
RP	Recommended Practice
SMYS	Specified Minimum Yield Strength

Chapter 1: INTRODUCTION

1.0 Background

Bottom trawling is of concern to subsea structures and pipelines as both offshore petroleum and fishing industries are often operating in the same areas. Subsea structures attract fish and populations of fish are likely to attract fisherman, hence their interaction is not avoidable. Burial of submarine pipelines reduces the risk but burial is not economically feasible where current velocities are low and wave action insignificant at deeper parts of the Norwegian continental shelf at which bottom trawling takes place. Only the trawling (and possibly anchoring) could justify special protection of a pipeline. Therefore pipeline global response when subjected to frequent crossing of commonly used type of trawl gears has to be investigated.

It is interesting to note that the NPD Regulations now require that all subsea installations on the Norwegian sector of the North Sea be designed so that fishing gear will not be damaged. This requirement may not apply for fishing exclusion zones, which could be imposed on the grounds of low fishing activity in the area and or proximity to permanent platforms [15].

During recent years it has been documented that the trend for trawl gear design and weight has increased. Particularly the use of clump weights often increases the efficiency and is expected to be popular and common in the future. New submarine pipelines need to be designed according to recently used trawl gears and previously installed pipelines subjected to interference have to be reevaluated.

Forces which are imposed on pipeline systems from fishing activities can be classified as interference loads. These loads include:

a. Impact force or trawl impact: this is the initial impact from fishing trawl which may damage the coating or cause local dents in the pipeline.

b. Pull over: this is the second phase and happens as the trawl is pulled over the pipeline. Pullover loads usually give a more global response of the pipeline.

c. Hooking: the trawl board is stuck under the pipe, the trawler is forced to stop, back up, and attempt to free the gear by winching in the warp line. It should be noted that during hooking the ship could pull down in case the ship does not stop.

1.1 Project scope

The scope of this project encompasses the following:

- Determination of pull over loads on a pipeline from commonly and recently Clump weights using the Finite Element Analysis software SIMLA.
- Sensitivity study of variables, like velocity of trawl gear, warp line (water depth), weight of trawl gear, pipeline diameter, soil condition and span heights, to understand their significance and derive design curves.
- Comparison of computed pull over loads, for most significant parameters, with DNV-RP-F111.
- Comparisons of cases (with and without temperature and pressure loads) to understand effect of axial compressive force prior to trawling.

1.2 Thesis Organization

The following are to be undertaken in this thesis work:

Chapter 2 discusses the different types of fishing gears which are commonly as well as recently used in the North Sea and the Norwegian Sea. Bottom trawl gears such as otter trawl, pair trawl rigging and beam trawling are the main focus in this chapter.

Chapter 3 deals with the theoretical background for subsea pipeline design analysis. Stresses in pipeline, hydrostatic collapse, pipeline stress-strain relationship, hydrodynamic loads and pipeline free spans are discussed.

Chapter 4 covers pipeline upheaval and lateral buckling. Sources and corrective actions of both pipeline upheaval and lateral buckling are discussed.

Chapter 5 discusses DNV pullover loads and duration following trawl gears data for trawl boards, trawl beam and clump weights.

Chapter 6 addresses trawl gear, seabed soil and pipeline parameters. In here special attention is provided to the most important parameters like trawling velocity, warp line length, mass of Clump weights and span height.

Chapter 7 provides pullover analysis and SIMLA model for interaction of Clump weights with subsea pipelines.

Chapter 8 provides results of pullover loads from commonly as well as recently used Clump weights on submarine pipeline. Results for Horizontal displacements are also included for pipelines with and without axial compressive force.

Chapter 9 provides the conclusions and recommendations from the study.

Chapter 2: REVIEW OF FISHING GEAR

Many different types of fishing gear are used in the commercial fishing industry especially in the North Sea, and further North on the Norwegian continental shelf. Commonly used trawl gears are as follows:

2.0 Bottom otter trawl

This type of bottom trawl gear is commonly used in Norwegian waters where the trawl bag or net is kept open by trawl boards. Trawl boards, see figure 2-1, are typically made of steel and are more or less rectangular. The boards keep the trawl bag open by hydrodynamic drag forces. It is known that the larger area swept by the trawl bag, the more efficient the trawl gear will be, giving a larger catch for the same distance travelled by the trawler [6].

Bottom trawling encompasses a wide range of gear designs and methods of operations. Towing speed ranges from 2 knots (1 m/s) to 6 knots (3 m/s) and fishing might be conducted at depths from 10 to 2500 meters. The vessels operating bottom trawls might have towing bollard pull ranging from 200 to 70000 kg [16].

Figure 2-1: Typical bottom trawl boards [16].

Bottom otter trawls are of different types and designs such as single trawl rigging and double bottom trawl rigging (twin trawl with clump weight):

a. Single trawl rigging

Common components are a pair of otter boards (trawl doors), sweeps/bridles and one or more trawl nets as shown in figure 2-2. At both sides the trawl bag is connected to the trawl boards through sweep lines. The trawl boards are further connected to the surface vessel by means of warp lines and the net is kept open by use of trawl boards.

The largest trawl board used in the North Sea and the Norwegian Sea has increased from about 1500 kg in the late 70's and 80's to 4000 kg in 2005. Currently trawl boards with a mass up to 6000 kg are used in the Barents Sea [5].

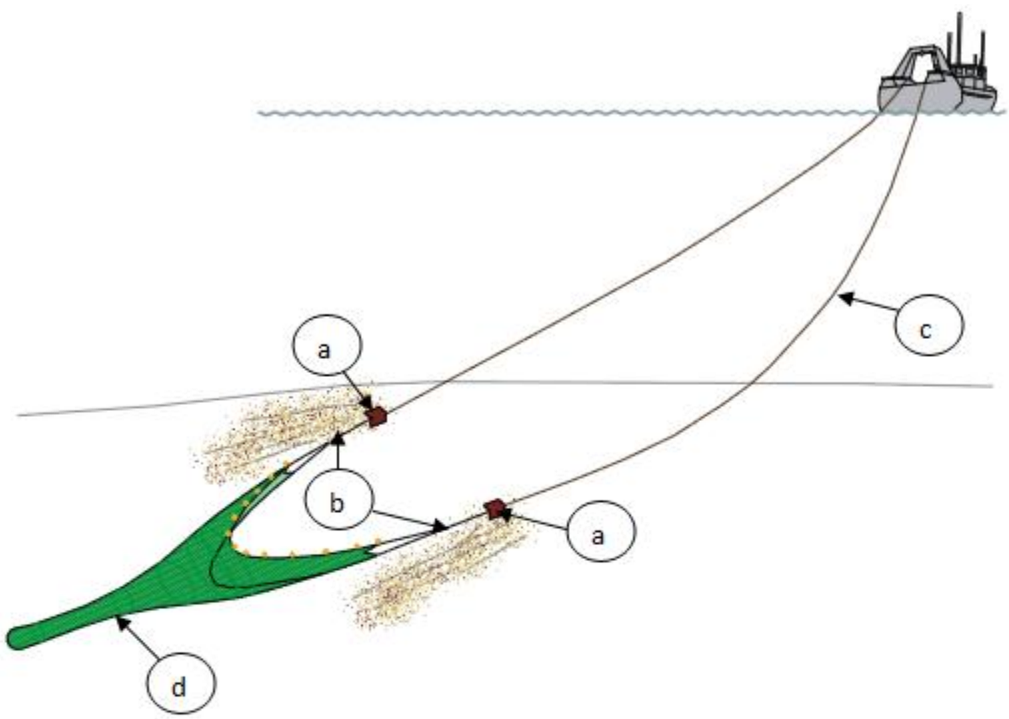


Figure 2-2: Bottom otter trawling – single trawl rigging [16].

In figure 2-2: **(**a) Trawl board

(b) Sweep line

(c) Warp line

(d) Trawl net

b. Double trawl rigging (Twin trawl with clump weight)

Recent development is to increase the size and weight of trawl doors. It is common to use two trawl bags, known as twin trawl or double trawl rigging, to increase the swept area and the performance of the trawler, see Figures 2-3 and 2-4. In this arrangement a weight (i.e. clump weight) is used to achieve bottom contact of the front part of the inner sweeps/bridles located in the centre between the two trawl nets. The weights differ in shape and rigging, and their effect on the bottom will vary. Two roller clump designs are shown in Figure 2-5. The doors and weights are connected to the trawl wings by sweeps or bridles (wire/chain/ropes).

Typical Clump weights, which are used in the Barents Sea and outside Greenland, have a mass up to 9-10 tonnes based on 2005 data [5].

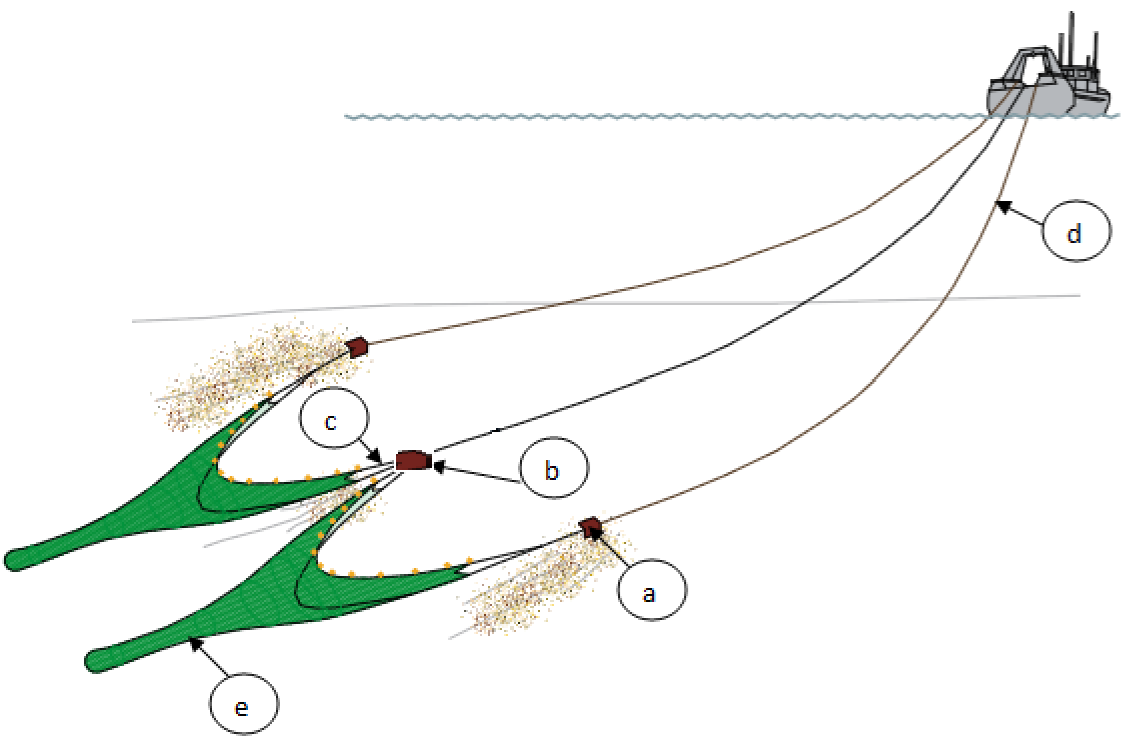


Figure 2.-3: Twin trawling with clump weight [16].

In figure 2-3: **(**a) Trawl board (d) Warp line

(b) Clump weight (e) Trawl net

(c) Sweep line

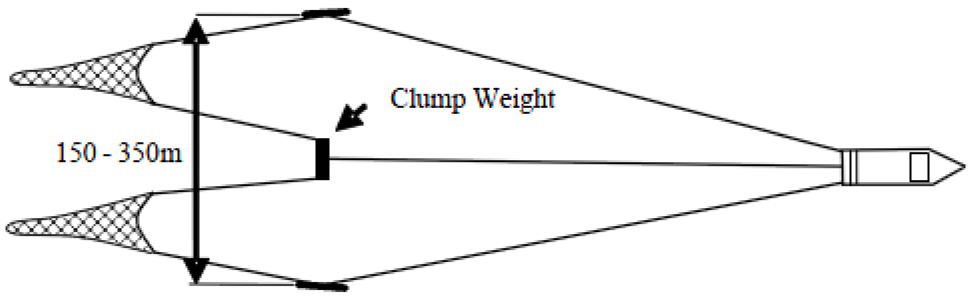


Figure 2-4: Typical twin trawling with clump-weight [5].

Figure 2-5: Typical roller clump weights used while double trawling [16].

2.1 Bottom pair trawl rigging

This trawl has no otter boards as bottom otter trawl, see Figure 2-6. Instead, the foremost contact points of a bottom pair trawl are often weights attached to the joining of the towing warp and the sweep [16].

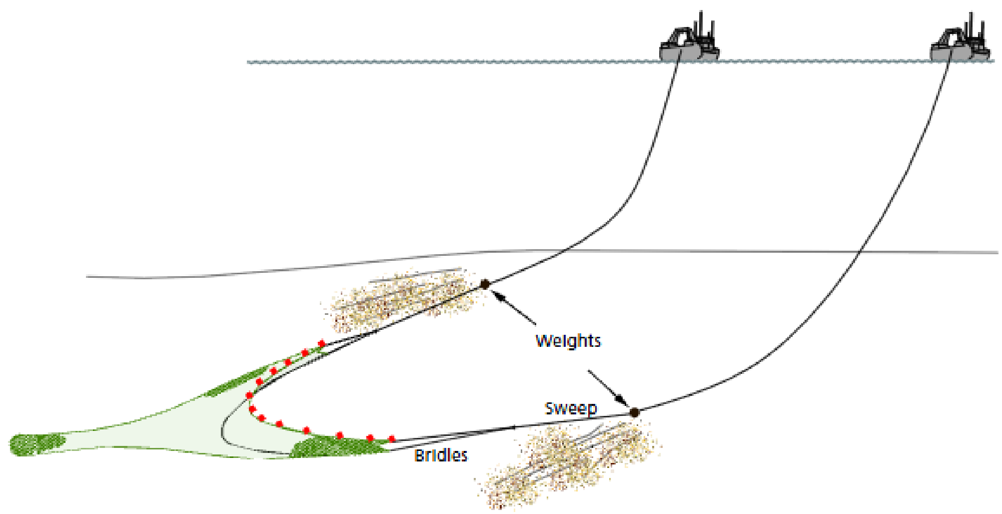


Figure 2-6: Bottom pair trawl rigging [16].

2.2 Beam trawling

For a beam trawl the mouth of the net is held open by a solid metal beam. Solid metal plates (shoes) are welded to the ends of the beam so that it can slide over the seabed as shown on Figures 2-7 and 2-8. Often two parallel beam trawls are towed from outriggers by a single vessel.
The concept of opening a trawl with a boom or spar has existed since the 1400s. It became more important as a fishing method in the 1960s as a replacement for otter trawls where chains had been added between the two otter boards to enhance flatfish catches. Since then the beam trawls have increased in weight, number of chains used and size of the beam. Since 1988, beam width has been restricted to 12 meters in European Union waters [16].

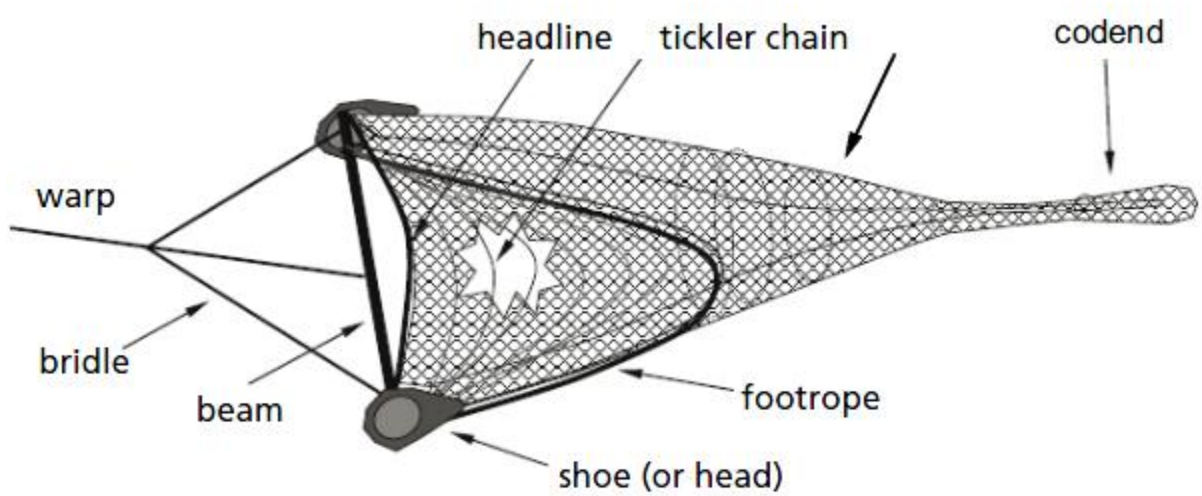


Figure 2-7: Beam trawl [16].

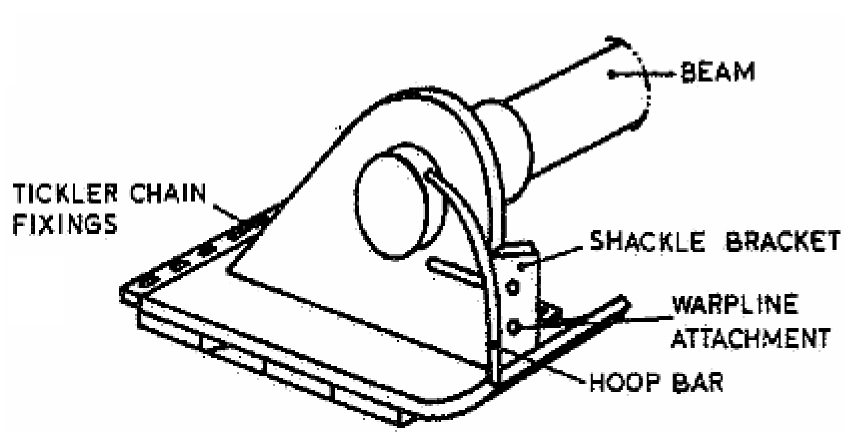


Figure 2-8: Typical outline of a beam trawl shoe [5].

Chapter 3: OVERVIEW OF SUBSEA PIPELINE DESIGN ANALYSIS

3.0 General

Pipeline stress analysis is carried out to find out if the pipeline stresses are acceptable in compliance with code needs and client needs throughout pipeline installation, testing and operation phases. The analysis carried out to ensure the stresses experienced are acceptable includes, trawl gear triggered stress, hoop stress, longitudinal stress, equivalent stress, span analysis and vortex shedding, stability analysis, expansion analysis and hooking analysis.

The fundamental theoretical background approach for the analysis is briefly outlined below:

3.1 Pipeline stress analysis

3.1.1 Hoop stress

Hoop stress is defined as the stress in a pipe wall acting circumferentially in a plane perpendicular to the longitudinal axis of the pipe and produced by the external pressure and the pressure of the fluid in the pipe.

The hoop stress (σ_h) or stresses in the pipe wall (steel) can be determined using the following equations:

- Stresses in the pipe wall (steel) due to the internal (P_i) and external pressure (P_o) based on equilibrium of forces is given by:

$$\sigma_h = \frac{p_i D_i - p_o D_o}{2t}$$

The following approximations are made:

- Stresses in the pipe wall (steel) from Barlow formula is given by:

$$\sigma_h = \frac{p_i D_o}{2t}$$

- Stresses in the pipe wall (steel) from Det Norske Veritas (DNV) is given by [2]:

$$\sigma_h = (p_i - p_o)\,\frac{D_O - t}{2 \cdot t}$$

Where:

σ_h = Hoop stress

P_i = Internal pressure

P_o = External pressure

t = Wall thickness (steel thickness)

D_i = Internal diameter

D_O = External diameter

Generally it is required from standards/codes of practice that hoop stress should not exceed a certain fraction of the Specified Minimum Yield Stress (SMYS) [1].

Basic derivation of hoop stress form equilibrium of forces is as follows:

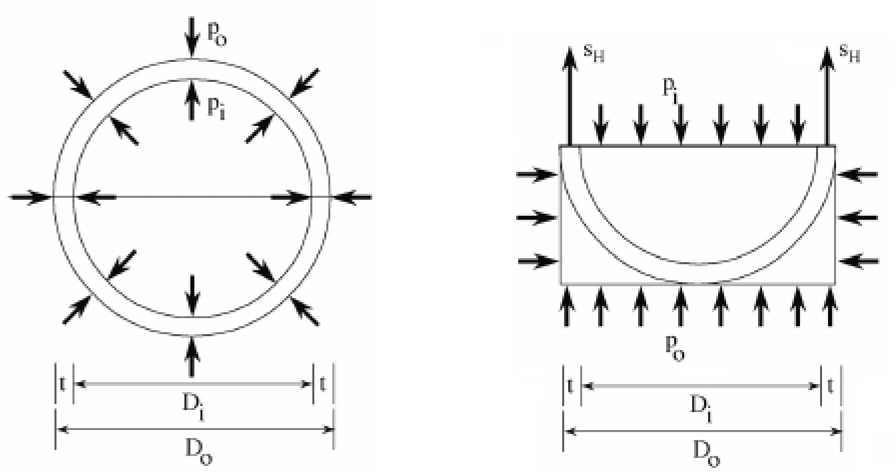


Figure 3-1: Circumferential stress in a pipeline pressurized internally (P_i) and externally (P_o) [13].

Vertical equilibrium of unit length gives (see figure 3-1):

$P_o D_O + 2S_h t = P_i D_i$, where $S_h = \sigma_h$

Rearranging the above equation gives:

$$\sigma_h = \frac{p_i D_i - p_o D_o}{2t}$$

The above equation gives the mean circumferential stress exactly, whatever the diameter-to-thickness (D/t) ratio [13].

3.1.2 Longitudinal stress

The longitudinal stress (σ_l) refers to the axial stress experienced in the pipe wall. The stresses arise primarily from two effects that are Poisson's effect and temperature. Besides they can arise due to bending stress, residual stress and end cup force induced stress [10].

A) Poisson's effect or Hoop stress (σ_h)

Considering idealized pipeline as thin-walled tube with mean radius (R), wall thickness (t), Elastic modus (E) and Poisson's ratio (ν). The longitudinal stress due to Poisson's effect (hoop stress) can be derived from stress-strain relationship for linear elastic isotropic material as follows [13]:

$$\varepsilon_l = \underbrace{\frac{\sigma_l}{E}}_{\text{longitudinal}} - \underbrace{\frac{\nu\sigma_h}{E}}_{\text{hoop}}$$

For complete axial constraint, ε_l = 0. Hence longitudinal stress (σ_l) is computed as follows:

$$\sigma_l = \frac{\nu p R}{t}$$

Where:
p = Operating pressure
ν = Poisson's ratio of steel
t = Wall thickness
R = Mean radius

B) Temperature effect (Thermal stress)

Thermal stresses are created due to the temperature difference between installation and operation. If the temperature of a pipeline is increased and if the pipeline is free to expand in all directions, it expands both circumferentially and axially. Circumferential expansion is usually completely unconstrained, but longitudinal expansion is constrained by seabed friction and attachments. It follows that if expansion is prevented, a longitudinal compressive stress will be induced in the pipe [13].

For constrained pipelines, longitudinal stress (compressive) due to thermal is computed as follows [13]:

$$\sigma_l = -E\alpha\theta$$

Where:
θ = Operating temperature rise (temperature difference between operating and installation)
α = Thermal coefficient for steel expansion
E = Young's modulus of steel

C) Residual stress

The longitudinal stress due to the residual stress is computed as follows [10]:

$$\sigma_l = \frac{F}{A_s}$$

Where:
F = Residual axial tension
A_s = Cross sectional area of pipe

D) Bending stress

The longitudinal stress due to bending stress is computed as follows [10]:

$$\sigma_l = \frac{Mc}{I}$$

Where:

M = Bending moment

c = Pipeline centroid, ($c = \pm D_0/2$)

I = Moment of inertia, ($I = \pi/4(D_O{}^2 - D_i{}^2)$)

E) Stress due to end cap force

End cup force occurs at any curvature along the pipeline and the longitudinal tensile stress due to end cup force for unrestrained condition is computed as follows [10]:

$$\sigma_l = \frac{F_{endcap}}{A_s}$$

Where:

F_{endcap} = End cap force, ($F_{endcap} = \pi/4\ Di^2\ (P_{internal} - P_{hydrostatic})$)

A_s = Cross sectional area of pipe

F) Total longitudinal stress

The total longitudinal stress during installation and operation phase can be determined by the following equations [10]:

- Longitudinal stress during installation phase (It can be due to temperature and Poisson's effect only)

$$\sigma_l = \frac{\nu p R}{t} - E\alpha\theta$$

- Longitudinal stress during operation phase (it can occur due to combination of temperature, pressure, residual stress, bending stress and end cup force induced stress)

$$\sigma_l = \frac{\nu p R}{t} - E\alpha\theta \pm \frac{F}{A_S} \pm \frac{Mc}{I} + \frac{F_{endcap}}{A_S}$$

3.1.3 Equivalent stresses

Combined stresses due to longitudinal and hoop stress can be determined from the relation given by Von Mises criterion where shear stress is neglected [1].

$$\sigma_{eq} = \sqrt{\sigma_h^2 + \sigma_l^2 - \sigma_h \sigma_l}$$

Generally it is required from standards/codes of practice that equivalent stress (σ_{eq}) stress should not exceed a certain fraction of the Specified Minimum Yield Stress (SMYS) during installation phase as well as operation phase.

3.2 Hydrostatic Collapse

During installation a pipeline is often empty for handling reasons. Therefore a substantial external pressure tends to make the pipeline cross section ovalize and collapse. Deepwater pipelines can be subjected to high external (hydrostatic) pressures and significant bending and this can lead to collapse of the pipeline and propagation along significant lengths. Bending/Hydrostatic collapse is the main determinant of wall thickness in very deepwater pipelines.

A perfectly round pipeline loaded by a steadily increasing internal pressure would remain circular until the pressure reached the elastic critical pressure (p_{ecr}), given by [13]:

$$p_{ecr} = \frac{E}{4(R/t)^3(1-\nu^2)}$$

Where:

R = The mean radius (measured to halfway through the wall thickness)

t = The wall thickness

E = The elastic modulus

ν = Poisson's ratio

In reality most pipelines are not perfectly circular; hence design codes often modify the above equation to account for initial pipe ovality and material plasticity. Besides care should be taken to include ovalisation induced during construction (e.g. reeling).

3.3 Stress-Strain relationship for pipelines

Figure 3-2 shows typical stress-strain relationships for pipelines. The behavior of the deformation during increasing stress is characterized by a linear and a plastic region. The transition between linear and plastic behavior is termed as the yield stress.

For pipelines some plastic behavior is usually allowed. API defines σ_Y as the stresses that will give us 0.5 % strain (0.005). The ultimate strain the pipe can handle, whereupon it breaks is found at 0.18-0.20%. The stress-strain curve will usually follow the experimentally formula below [7]:
The parameters σ_o and *k* are experimentally determined.

$$\varepsilon = \frac{\sigma}{E}\left[1+\left(\frac{3}{7}\cdot\frac{\sigma}{\sigma_0}\right)^k\right] \text{ (Ramberg-Osgood Formula)}$$

Where:

ε = Pipeline strain

E = Elastic modulus

σ = Pipeline stress

σ_o and k = Parameters determined experimentally

The quality of steel pipes can be obtained from formulas given in American Petroleum Institute (API) Specification. Pipeline steel of type X70 means for example yield stress at 70 ksi (kilo pounds/square inch)

Recently grade X70 is widely used for high pressure pipelines in many countries. This is because reduction in material cost can be achieved by reducing the wall thickness for internal pressure containment or external in case of deep waters. However, design, manufacture and construction with high grade materials require new fabrication approaches and welding techniques and these can expose submarine pipeline projects to increased levels of technical and commercial risks.

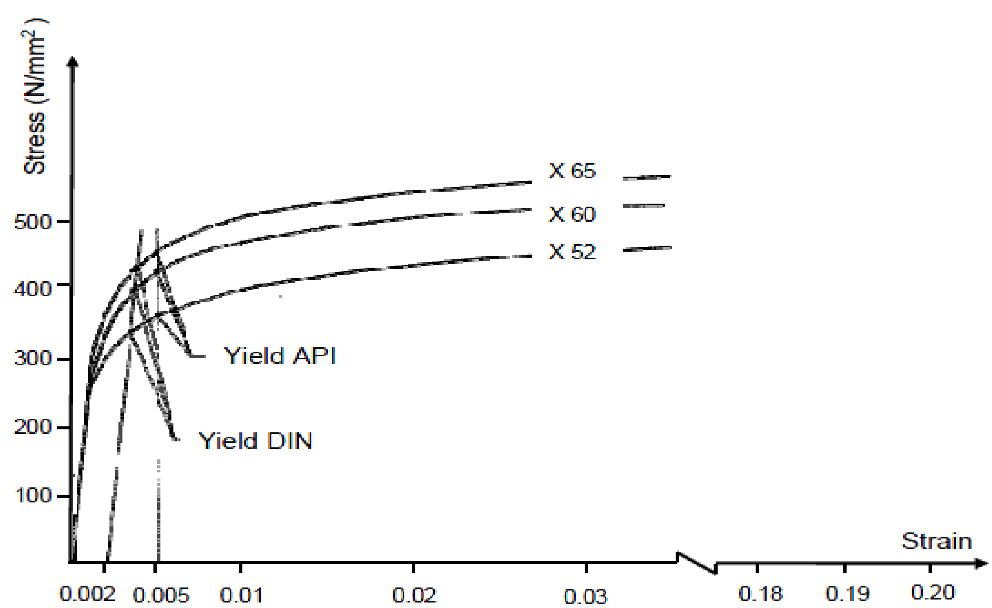


Figure 3-2: Stress strain relationship for pipelines according to API and DIN [7].

3.4 Hydrodynamic loads and on bottom stability

Hydrodynamic loads are defined as flow-induced loads caused by the relative motion between the pipe and the surrounding water. Hydrodynamic loads include waves, current, relative pipe motions and indirect forces e.g. caused by vessel motions during pipe laying [2].
A pipeline on the sea bottom experiences a vertical lift force and horizontal drag and inertia forces due to the particle flow past the pipeline induced by waves and currents. And as a requirement the pipeline shall not move horizontally or vertically in waves and currents during operations.

During installation, a safety factor of 1.1 is generally recommended by DNV-RP-F109 [17] in order to avoid floatation of the pipeline in water. The requirements are to be fulfilled for a 10-year sea state condition for the actual period (summer-criteria or all year-criteria). This means, there is a 10 % probability of exceedance per year for the actual period. The criteria for pipe laying during the summer period are typically defined as the 10-year summer-storm criteria.

During operations DNV-RP-F109 [17] recommends that the stability has to be ensured for the 100-year criteria for waves (1% probability of excellence per year) together with 10-year criteria for currents or 100-year current criteria together with a 10-year wave.

The methodology for stability design is dependent on which design code is to be used. The installation depth becomes important due to the decreasing values of the water wave particle velocities with increasing water depth. Note that the particle velocity for waves in deep water is very small, whereas ocean currents might be considerable. Figure 3-3 shows Hydrodynamic forces and other forces on pipelines placed on sea bottom [7]:

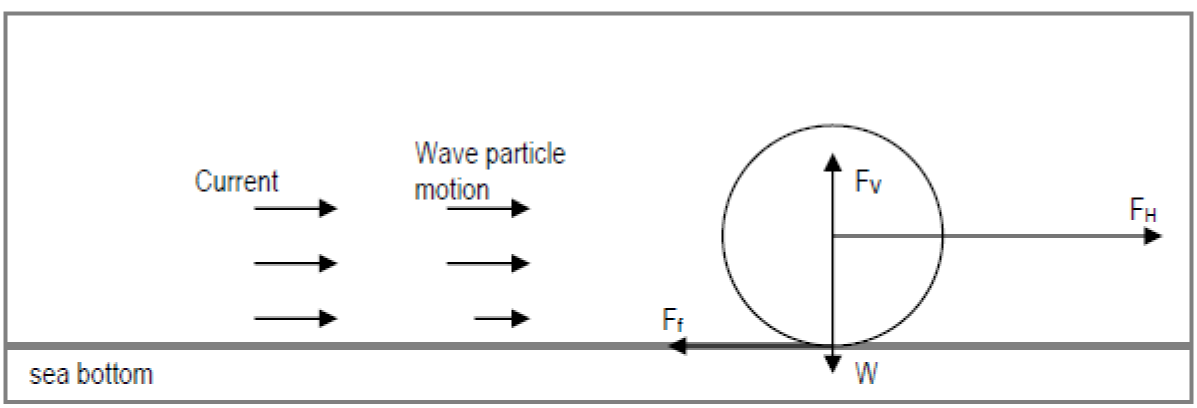


Figure 3-3: Forces on submarine pipeline [7].

In Figure 3-3:

F_v = Vertical force (lift force)

F_h = Horizontal force (drag and inertia forces)

F_f = Friction force

W = Gravity force (submerged weight of the pipe)

- **Horizontal pipeline stability is secured when:**

$$F_f > F_H = \gamma_{st}(F_D + F_I)$$

Where:
γ_{st} = Factor of safety, normally not taken less than 1.1
ρ = Water density, (sea water typically 1025 kg/m3)
D = Outer diameter of the pipeline (including thickness of coating)
C_D = Drag coefficient
V = *Water particle velocity of current plus wave*
C_I = *Inertia coefficient*

$$F_D = \frac{1}{2} \cdot \rho \cdot D \cdot C_D \cdot |V| \cdot \mathrm{V} \text{ (Drag term)}$$

$$F_I = \frac{\pi}{4} \cdot \rho \cdot D^2 \cdot C_I \cdot \dot{V} \text{ (Inertia term)}$$

And:

$$F_f = f \cdot (W - F_V)$$

Where:
f = Friction factor between the pipeline and sea bottom

$$F_V = \frac{1}{2} \cdot \rho \cdot D \cdot C_L \cdot |V| \cdot V$$

Substituting this equation to the above equation gives:

$$f \cdot (W - F_V) > \gamma_{st}(F_D + F_I)$$

Substituting equation for lift force to the above equation gives:

$$f \cdot (W - \left(\frac{1}{2} \cdot \rho \cdot D \cdot C_L \cdot |V| \cdot V \right) > \frac{\gamma_{st}}{2} \cdot \rho \cdot D \cdot C_D \cdot |V| \cdot V$$

Where:
C_L = Lift coefficient

Further manipulation the above equation gives the required pipeline weight as a function of the lift, drag and friction force as expressed below [7]:

$$W > \left(\frac{\gamma_{st}}{2 \cdot f} \cdot C_D + \frac{1}{2} \cdot C_L \right) \cdot \rho \cdot D \cdot |V| \cdot V$$

- **Vertical pipeline stability is secured when:**

$$W > \gamma_{st} F_L$$

Substituting for lift force (F_V) gives [7]:

$$W > \frac{\gamma_{st}}{2} \cdot \rho \cdot D \cdot C_L \cdot |V| \cdot V$$

Since hydrodynamic forces may reduce the lateral pipe-soil resistance due to lift effects, DNV RP F110 [4] recommends that the hydrodynamic forces not to be included in the fishing gear pull-over analysis. Therefore hydrodynamic forces are not included in the thesis during FE analysis.

3.5 Pipeline free spans

Free spans of pipelines can mainly occur because of scour or eroded bed, rocky or cohesive bed, pipeline crossing and bathymetric features. Free spans are important to understand and address as they can create problems related to vortex induced vibrations, static stresses (self weight and environmental loads) and global buckling.

In this section free span problems related to Vortex Induced Vibration (VIV) will be discussed

3.5.1 Vortex Induced Vibration (VIV)

VIV can result if the vortex shedding frequency coincides with a multiple of the natural frequency of the span. And this can lead to fatigue of welded pipe joints.

The Vortex shedding frequency is given by the Strouhal Number, given by [10]:

$$St = \frac{f_v D}{U}$$

Where:

f_v = Vortex shedding frequency

D = Pipe diameter

U = Flow velocity

St = Strouhal Number (region of interest for most pipeline applications St $\approx$ 0.2)

When the Vortex shedding frequency matches the natural frequency of the span, a complex interaction occurs. Therefore understanding the dynamic behavior of the span is important. VIV behavior is usually linked with different flow regimes, characterized by the reduced Velocity, given by [13]:

$$V_R = \frac{U}{f_n D}$$

Where:

D = Pipe outside diameter

U = Flow velocity

f_n = *N= Natural frequency of span*

And the natural frequency (N= f_n) is given by:

$$N = \frac{C}{L^2}\sqrt{\frac{F}{m}}\sqrt{1-\frac{P}{P_E}}$$

Where:

L = Length of the span

F = Flexural rigidity (F= EI)

m = Mass per unit length (including added mass to account for the surrounding water).

C = *C*onstant that depends on the end conditions, 3.5 for fixed at both ends.

P = Effective axial force (compression positive)

P_E = Euler buckling force for the same span length L, F and end condition*s,* $4\pi^2 F / L^2$ for fixed ends*.*

Generally Pipeline VIV behavior can be summarized into three regions [13].

. If 1.0 < VR < 2.2 then Symmetric (plus alternate) vortex shedding causing in-line oscillations occurs.

. If 2.2 < VR < 3.5 then Alternate vortex shedding causing in-line oscillations occurs.

. If 4.8 < VR < 12 then Alternate vortex shedding causing cross-flow oscillations occurs.

Generally industries used to restrict the length of the free span to 40m in order to avoid large vibrations and as rough rule of thumb, if the length to diameter ratio (L/D) is < 40, then VIV will not be a problem, however, DNV-RP-F105 [3] gives an advanced engineering methodology for calculating VIV induced stresses so as longer free spans might be accepted.

Even though mitigation methods for VIV come at some cost, it is possible to mitigate VIV by strakes, fins, shields, artificial seaweed and supports. The most common is use of rock dump to support free spans.

Chapter 4: UPHEAVAL AND LATERAL BUCKLING

4.1 General upheaval buckling

The term “Upheaval buckling” is used when a buried pipeline arch upward out of the seabed, forming a raised loop that may project several meters. Buckles have to be taken seriously, because they can overstress the pipe wall, occasionally lead to a rupture, and lead to other difficulties such as excessive hydrodynamic loads or easy hooking by fishing trawls and anchors if they projects up into the sea [13].

Upheaval buckling is a common design issue for buried pipelines when out-of-straightness of the pipeline combined with high axial compressive forces induced by extreme operating conditions causes the pipeline to buckle upwards. In order to prevent upheaval buckling, the pipeline has to be buried deep enough such that the soil cover is sufficient to provide adequate uplift resistance. Figure 4-1(a) shows a typical schematic of upheaval buckling while Figure 4-1(b) shows a pipeline in the field which has undergone upheaval buckling [14].

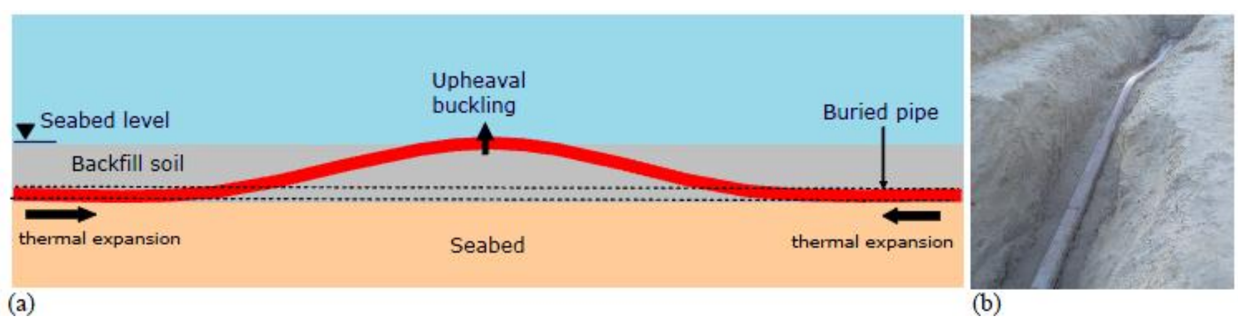


Figure 4-1: Upheaval buckling of a buried pipeline [14].

4.1.1 Driving force for upheaval buckling

Upheaval buckling is driven by the longitudinal compressive force in the pipe wall and the fluid contents [13].Refer section 3.1.2 (F) that the total longitudinal stress for complete axial constraint is given by:

$$\sigma_l = \frac{v p_i R}{t} - E\alpha\theta$$

Where:

p_i = p = Operating pressure (Internal pressure)

v = Poisson's ratio of steel

t = Wall thickness

R = Mean radius

And refer section 3.1.1 Stresses in the pipe wall is given by Barlow formula:

$$\sigma_h = \frac{p_i D_o}{2t}$$

Replacing D_O with twice the mean radius R, stresses in the pipe wall is given as:

$$\sigma_h = \frac{p_i R}{t}$$

The cross-section area of a pipe wall is $2\pi Rt$, and then the longitudinal force in the pipe wall can be computed as:

$$2\pi Rt\sigma_l = 2v\pi R^2 p_i - 2\pi\, RtE\alpha\theta$$

An additional component of the longitudinal force is given by the pipe contents pressure. The cross section of the contents is πR^2. The longitudinal stress in the contents is –p if counting tension positive. Therefore, the longitudinal force in the contents will be [13]:

$$-\pi R^2 p_i$$

Adding the two above equations that are the longitudinal force in the pipe wall and fluid contents, the total longitudinal force (Upheaval buckling driving force) is given by:

$$-(1-2v)\pi R^2 p_i - 2\pi \, RtE\alpha\theta$$

In most cases p and θ are positive. Besides both terms of the above equation; that involves temperature and pressure; are negative and hence compressive. Referring the equation, it is possible to suggest that the pressure alone can cause upheaval buckling.

4.1.2 Upheaval buckling preventing measures

The following listed strategies are mainly applied to prevent upheaval buckling,

a. Reduce the driving force, either by reducing the operating temperature and pressure or by reducing the pipeline wall thickness to the minimum possible value. Alternatively, the driving force can be reduced by laying the line in a zigzag, introducing cooling loops that allow the fluid to cool by heat transfer to the sea, or incorporating expansion loops at intervals along the pipeline [13].

b. Make the pipeline profile smoother. This can be done by selecting the route so as to avoid rough areas, which also helps to reduce the length and number of spans. The profile can be further smoothed by careful trenching, particularly with a ‘smart’ plough that trenches more deeply on high points of the profile [13].

4.1.3 Upheaval buckling corrective actions

Even though not all buckles need to be corrected, the following corrective actions can be taken based on the situation and consequences,

a. If the pipeline is not overstressed, stabilize the pipeline in its new position, by placing rock mattresses around the pipeline. They must have enough weight to prevent further movement and must themselves be stable [13].

b. Cut and remove the buckled section of the pipe and replace it with a new spool piece connected by hyperbaric welding, surface tie-in, or mechanical connection. It will obviously be necessary to make sure that the buckling will not repeat itself, by provision of additional cover or by incorporating an expansion spool [13].

4. 2 Lateral buckling

Lateral buckling is like upheaval buckling except that constraint is provided by friction instead of weight. The movement is sideways and sometimes called as "snaking of the pipeline", see figure 4-2.

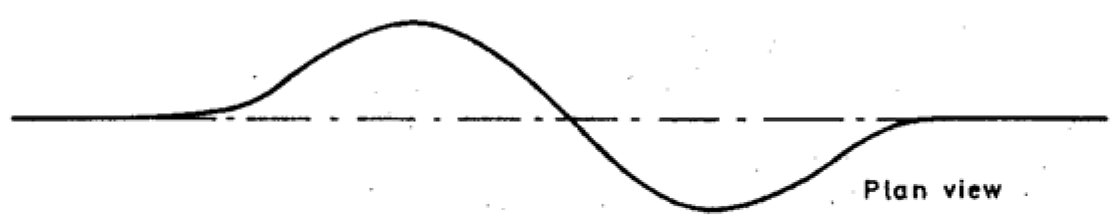


Figure 4-2: Lateral buckling of a pipeline [9].

Assuming that the pipeline has no severe component of out-of-straightness in the vertical plane, then a pipeline laid directly onto the seabed without cover will buckle sideways on the seabed rather than upwards. A pipeline on the seabed can displace to either side of its original position. The pipeline can buckle into either a symmetric or asymmetric mode, where the symmetry is referred to an axis drawn through the centre of the buckle and perpendicular to the original centreline of the pipeline. The symmetric and asymmetric modes are illustrated in Figure 4-3(a) and 4-3(b).

In the symmetric mode a large main half-wave forms at the centre of the buckle, while in the asymmetric mode two main half-waves form on either side of the centre of the buckle. The actual mode which will be adopted by the pipeline will depend on the pipeline out-of-straightness in the horizontal plane, and on any other seabed features which could influence the lateral movement of the pipeline.

The asymmetric and symmetric modes consist of one or two central half-waves surrounded by a decaying sequence of half-waves moving away from the centre of the buckle. The sequence of half-waves arises because the distributed soil resistance forces cannot provide the concentrated lateral forces at the ends of each half-wave which are required for equilibrium. The amplitude of each half wave decreases rapidly with increasing distance from the centre of the buckle [11].

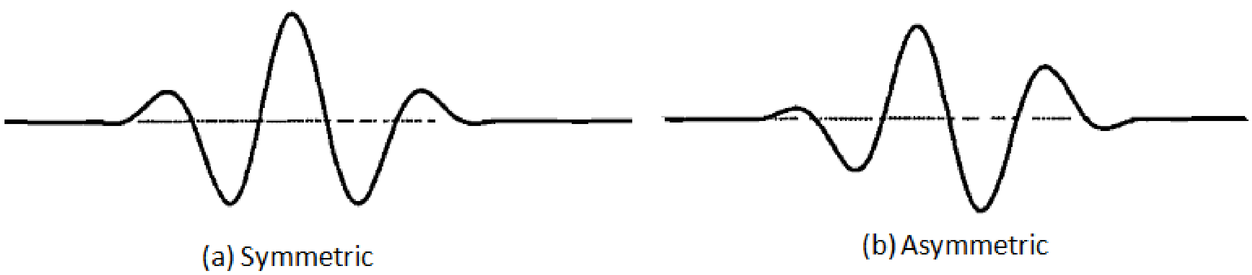


Figure 4-3: Symmetric and asymmetric buckle modes [11].

The total length of pipe over the half-waves of the buckle is greater than the length of the initially straight pipe over the same section. The formation of a buckle therefore involves the movement of pipe into the buckle from the straight pipeline sections on either side of the buckle, and leads to a modification of the axial force within the pipeline. The axial feed in movement for a single, isolated buckle in an infinitely long pipeline is illustrated below in Figure 4-4 [11].

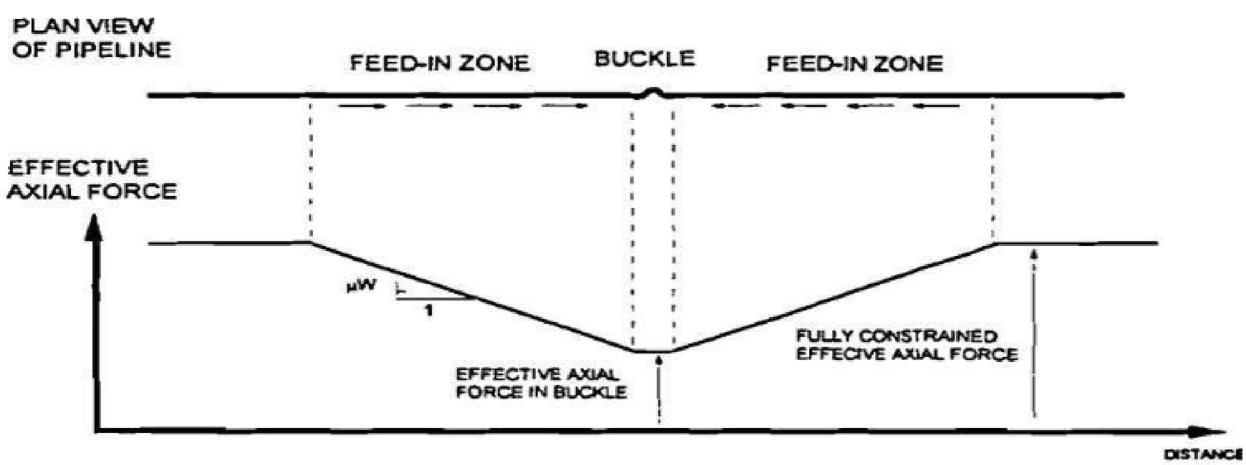


Figure 4-4: Feed – In to a Single Buckle in an Infinite Pipeline [11].

4.2.1 Driving force for lateral buckling

See figure 4-5 for common lateral buckle mode shapes, where the definition of buckle length L is the same as used by Hobbs [9]. All of the idealised modes in Figure 4-5 assume some form of concentrated lateral force at the end of the outermost half-waves for equilibrium. The error introduced by this assumption is greatest for mode 1 (which is the same mode as an upheaval buckle) but becomes insignificant for higher modes [11].

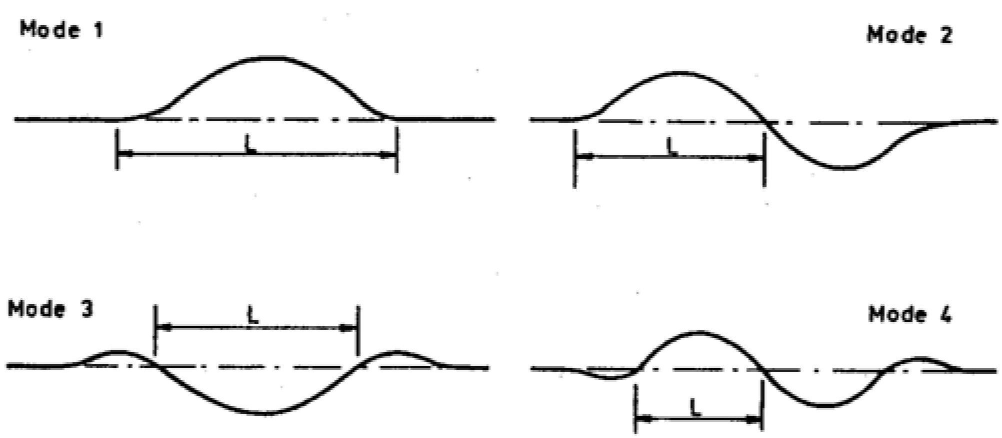


Figure 4-5: Theoretical buckle modes [9].

Same as section 4.1.1, for a thin-walled pipeline which is fully constrained from axial or lateral movement, the effective axial force in the pipeline (which includes the contribution due to the internal pressure) is given by:

$$F_0 = (1-2\nu)\frac{\pi}{4}D^2\Delta p + \pi D t E \alpha \theta$$

Where:

D = Pipe diameter

t = Pipe wall thickness

Δp = Pressure difference between the internal and ambient seawater pressures

θ = Temperature difference between the operating and installation temperatures.

The relationship between effective axial force (fully restrained axial force) and buckle length is given by [9]:

$$F_0 = F + k_3 \mu_a W L \left[\sqrt{\left(1 + k_2 \frac{E \cdot A \cdot \mu_l^2 \cdot W L^5}{\mu_a (EI)^2} \right)} - 1 \right]$$

Where:

F_o = Post-buckle axial force

μ_a = Coefficient of axial friction

μ_l = Coefficient of lateral friction

W = Submerged weight of the pipeline

A = Steel cross-sectional area

I = Second moment of area

L = Buckle length corresponding to F_o

E = Young's modulus

Where, F is the compressive effective axial force within the buckle given by:

$$F = k_1 \frac{EI}{L^2}$$

The maximum amplitude of the buckle can then be determined from [9]:

$$y = k_4 \frac{\mu_l W}{EI} L^4$$

And the maximum bending moment is given by:

$$M = k_5 \mu_l W L^2$$

The five constants k_1, k_2, k_3, k_4 and k_5 are dependent on the mode of buckling and are listed on the following table 4-1 [9].

Mode	k_1	k_2	k_3	k_4	k_5
1	80.76	6.391×10^{-5}	0.500	2.407×10^{-3}	0.06938
2	$4\pi^2$	1.743×10^{-4}	1.000	5.532×10^{-3}	0.1088
3	34.06	1.668×10^{-4}	1.294	1.032×10^{-2}	0.1434
4	28.20	2.144×10^{-4}	1.608	1.047×10^{-2}	0.1483

Table 4-1: Hobbs' Lateral Buckling Constants

4.2.2 Management of lateral buckling

Lateral movements are often harmless, because the lateral movement occurs over a substantial distance, the bending stresses are small, and the buckle does not localize into a sharp kink. In some instances, however, it may be larger, and longitudinal movements of the pipeline towards the buckle may lead to a localization in which all the movement is concentrated in one buckle. At the point where the lateral movement is largest, the pipe may form a localized kink in which the strain is large enough for the wall to rupture. A good solution to manage lateral buckling is just by deliberately creating small bends at regular intervals. Each bend initiates a lateral buckle, but because there are many buckles, the displacement at the largest buckle is not excessive [13].

Methods of mitigating lateral buckling include, product cooling, rock dumping, anchoring, or mats and laying the pipeline in a snaked configuration.

Chapter 5 DNV PULLOVER FORCES AND TRAWLING DATA

5.0 General

As DNV-RP-F111 defines pull-over as the second phase after an impact where the trawl board, beam trawl or clump weight is pulled over the pipeline. This phase can last from about 1 second to some 10 seconds [5].
As clearly mentioned in the project scope, only pull over loads for commonly and recently used Clump weights in the North Sea and the Norwegian Sea will be investigated. Generally pullover loads are much higher than impact loads; hence it is reasonable to limit the project scope to analysis of pullover loads.
It has been common industrial practice in the North Sea and the Norwegian Sea to leave pipelines exposed when the diameter is more than 16". Note that DNV-RP-F111 is applicable to rigid pipelines with outer steel diameters larger than 10". Hence rigid steel pipeline with outside diameter 0.7868m (about 30") is considered in the analysis in order to enable us to compare the results with DNV's recommended practice.

5.1 Pullover loads for trawl boards

Pullover loads namely horizontal and vertical forces from trawl boards shall be applied as a single point load to the pipeline under consideration.

- The maximum horizontal force applied to the pipe, F_p, is given by [5]:

$$F_p = C_F \cdot V(m_t k_w)^{1/2}$$

Where:

Fp = Pull over force (Horizontal) for one trawl board

k_w = Warp line stiffness = $\frac{3.5*10^7}{L_W}[N/m]$, (for one single 32-38mm diameter wire)

V = Trawling velocity

m_t = Board steel mass

L_w = Length of the warp line (typically 2.5 to 3.5 times the water depth)

The coefficient C_F is calculated as follows:

$C_F = 8.0\left(1 - e^{-0.8\overline{H}}\right)$ for polyvalent and rectangular boards

$C_F = 5.8\left(1 - e^{-1.1\overline{H}}\right)$ for v-shaped boards

And $\overline{H}$ is a dimensionless height:

$$\overline{H} = \frac{H_{sp} + D_o / 2 + 0.2}{B}$$

Where:

H_{sp} = Span height

D_o = Pipe outer diameter

B = Half-height of the trawl board

- The maximum vertical force acting in the downward direction, F_z, is given by [5]:

$$F_z = F_p (0.2 + 0.8 \cdot e^{-2.5 \cdot \overline{H}}) \text{ for polyvalent and rectangular boards}$$

$$F_z = \tfrac{1}{2} F_p \quad \text{for V-shaped boards}$$

Where:

Fp = Pull over force (Vertical) for one trawl board

e = Mathematical constant ($e \approx$ 2.718)

5. 2 Pullover loads for beam trawls

- The maximum horizontal force applied to the pipe, F_p, is given by [5]:

$$F_p = C_F \cdot V\left[(m_t + m_a) \cdot k_w\right]^{1/2}$$

Where:

Fp = Total pull over force (Horizontal) from both beam shoes

m_t = Steel mass of beam with shoes

m_a = Hydrodynamic added mass and mass of entrained water

- The maximum vertical force acting in the downward direction, F_z, is given by [5]:

$$F_z = \tfrac{1}{2} F_p$$

Where:

F_z = Total pull over force (Vertical) from both beam shoes

5.3 Pullover loads from recently used trawling (Clump weight)

As specified on the project scope, roller type clump weight is used as shown below on the figures 5-1(a) and 5-1(b). Commonly and recently used Clump weights with masses of 4.5 tonnes and 9 tonnes are considered for this thesis. In the future the weight of fishing gear is expected to increase as fishing methods and designs are changing. Hence it is important to reflect the increase in weight and design of trawl gears in new pipeline designs and reassess the integrity of already installed pipelines.

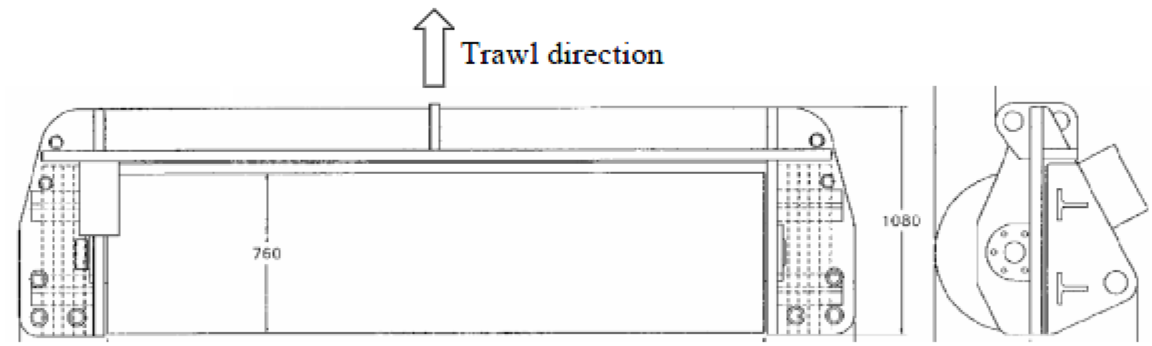


Figure 5-1(a): Typical clump weight (Roller type) [5].

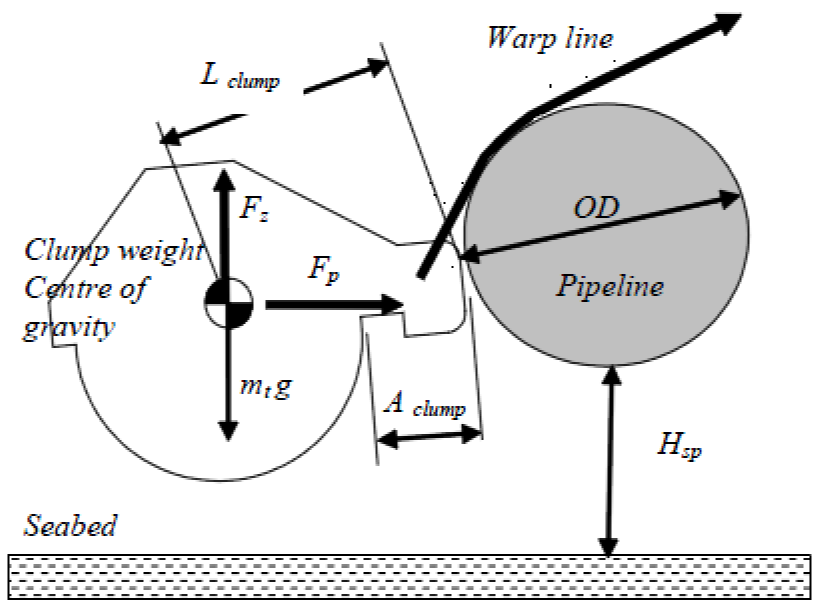


Figure 5-1(b): Clump weight interaction with pipeline [5].

In figure 5-1(b): $OD = D_o$ (outer diameter of pipeline)

The maximum horizontal force from the clump weight applied to the pipe, F_p, is given by [5]:

$$F_p = 3.9 \cdot m_t \cdot g \cdot \left(1 - e^{-1.8 \cdot h'}\right) \cdot \left(\frac{D_o}{L_{clump}}\right)^{-0.65}$$

Where $h^{'}$ is a dimensionless height:

$$h^{'} = \frac{H_{sp} + D_o / 2}{L_{clump}}$$

And where:

D_o = Pipe outer diameter including coating

L_{clump} = Distance from the reaction point to the centre of gravity of the clump weight

(L_{clump} = 0.7m for drum diameter of 0.76m)

m_t = Steel mass of clump

g = gravitational acceleration

- The maximum vertical force from the clump weight, F_z, is given by [5]:

$$F_z = 0.3F_p - 0.4 \cdot m_t \cdot g, \text{ downward force (i.e. negative sign)}$$

$$F_z = 0.1F_p - 1.1 \cdot m_t \cdot g, \text{ upward (vertical) force}$$

Take one that gives the most critical load combination during analysis input.

5.4 Pullover forces time history

5.4.1 Trawl boards and beam trawls

The total pull-over time, T_p, is given by [5]:

$$T_p = 2 \cdot C_F \sqrt{\frac{m_t}{k_w}} + \frac{\delta_p}{V} \quad , \text{ for trawl boards}$$

$$T_p = 1.5 \cdot C_F \sqrt{\frac{m_t}{k_w}} + \frac{\delta_p}{V} \quad , \text{ for beam trawls}$$

Where:

δ_p = Displacement of the pipe at the point of interaction

The value of $\frac{\delta_p}{V}$ is unknown prior to analysis. Therefore, according to DNV-RP-F111 it is assumed that:

$$\frac{\delta_p}{V} = \frac{2 \cdot C_F \left(\frac{m_t}{k_w}\right)^{0.5}}{10} \quad , \text{ for trawl boards}$$

$$\frac{\delta_p}{V} = \frac{1.5 \cdot C_F \left(\frac{m_t}{k_w}\right)^{0.5}}{10} \quad , \text{ for beam trawls}$$

Figures 5-2 and 5-3 show the force-time history of the horizontal (F_p) and vertical (F_z) forces applied to the pipeline for trawl boards and beam trawls.

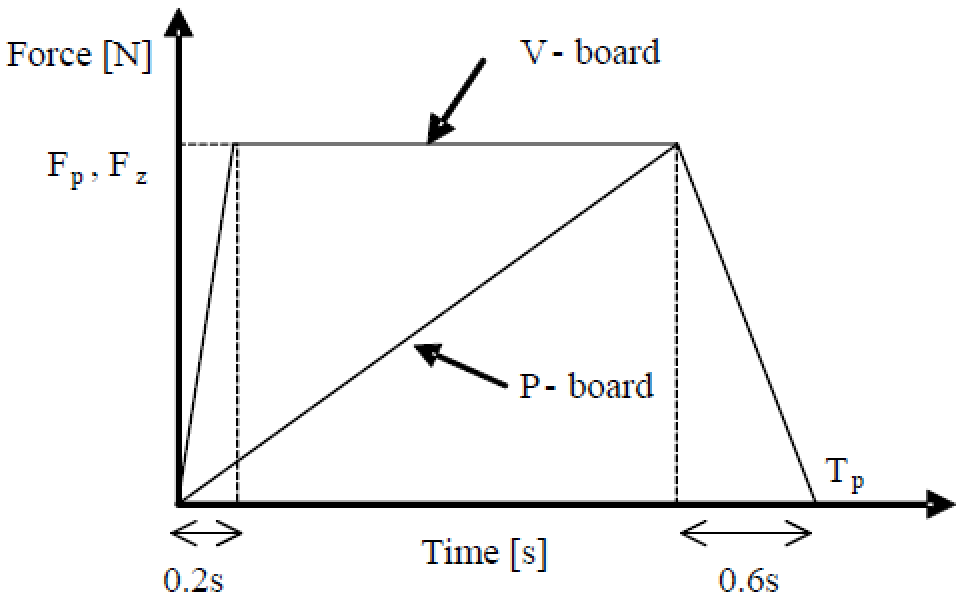


Figure 5-2: Force-time history for trawl boards pullover force on pipeline [5].

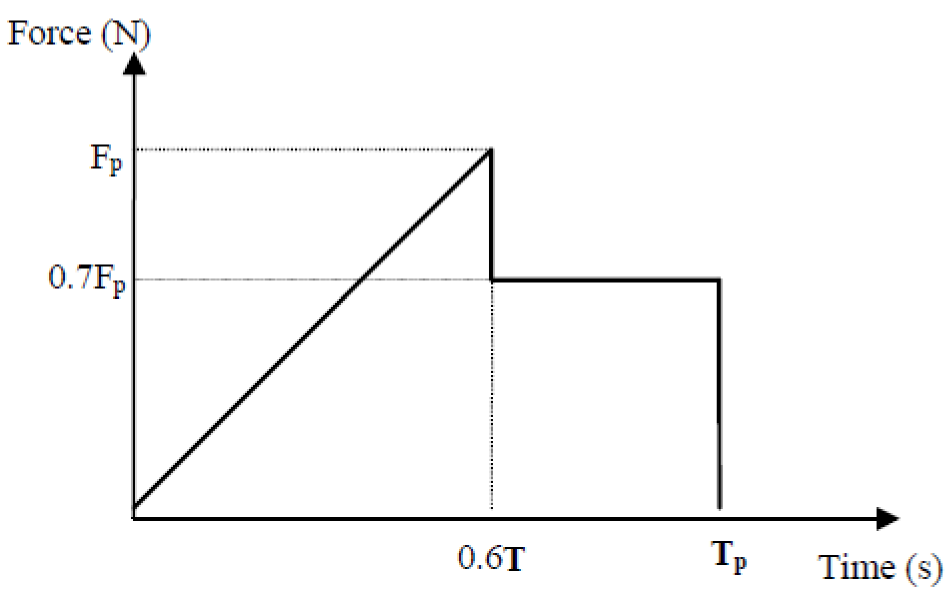


Figure 5-3: Force-time history for beam trawls pullover force on pipeline [5].

5.4.2 Clump weight

The pull-over duration of the roller type clump weight is given by [5]:

$$T_p = \frac{F_p}{(k_w \cdot V)} + \frac{\delta_p}{V}$$

The value of $\frac{\delta_p}{V}$ is unknown prior to analysis. Therefore, according to DNV-RP-F111 [5] it is assumed that:

$$\frac{\delta_p}{V} = 0.1\left(\frac{F_p}{(k_w \cdot V)}\right)$$

Figures 5-4 shows the force-time history of the pullover force for roller type clump weights. This applies for both horizontal (F_p) and vertical (F_z) forces.

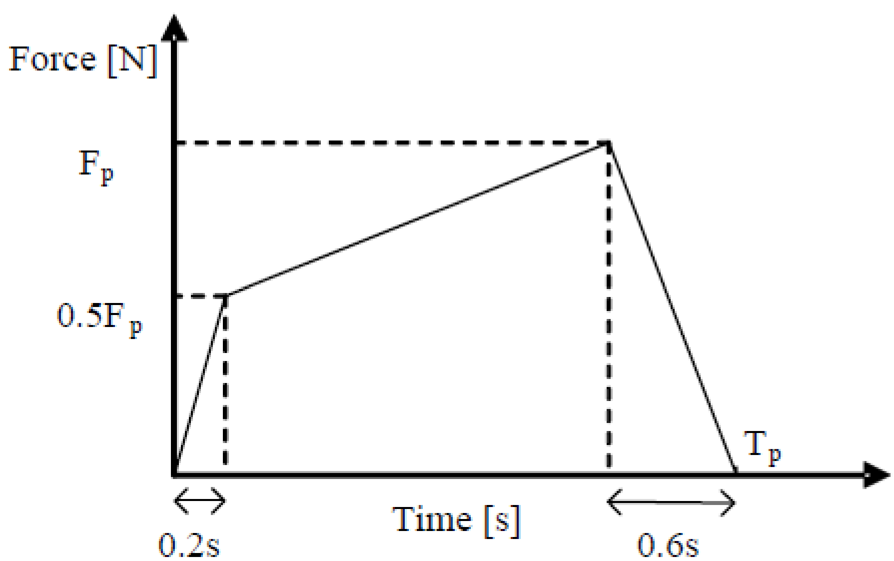


Figure 5-4: Force-time history for roller clump weight pullover force on pipeline [5].

5.5 Trawl gears data

Following DNV-RP-F111, this section gives appropriate data for the largest trawl boards, beam trawls and clump weights in use in the North Sea and the Norwegian Sea. Refer table 5-1 for the above mentioned trawling equipment.

Parameter	Unit	Trawl Gears Type			
		Polyvalent & Rectangular	Industrial V-board	Beam Trawl	Clump Weight
Steel Mass, m_t	kg	4500	5000	5500	9000
Dimension Lxh	m	4.5 x 3.5	4.9 x 3.8	17.0 3)	2)
Effective impact velocity	m/s	$2.8C_h$ 1)	$1.8C_h$ 1)	3.4	2.8
In plane stiffness, k_i	MN/m	500	500	-	4200
Bending board stiffness, k_b	MN/m	10	10	-	-
Hydrodynamic added mass, m_a	kg	$2.14m_t$	$2.90m_t$	1500	3140
Pull-Over Duration Coeff. C_T	-	2.0	2.0	1.5	-

Table 5-1: Trawl gears data [5].

1) The factor C_h (span height correction factor) is given in figure 5-5.

2) Typical dimension of the largest roller clump weight of 9T are L= 4 m wide (i.e. length of roller) by 0.76m dia. cross section.

3) Beam Trawl length (i.e. distance between outside of each shoe)

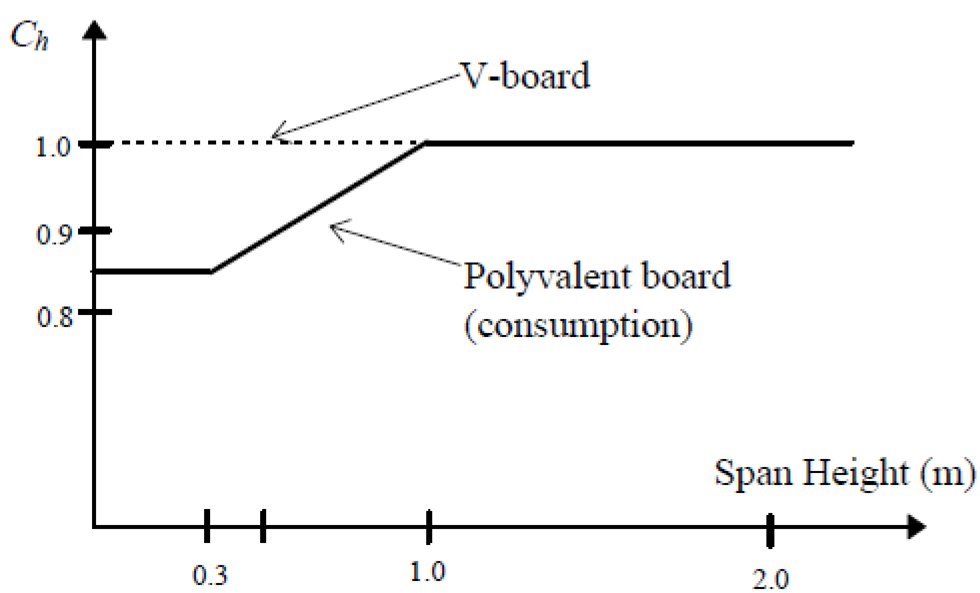


Figure 5-5: C_h coefficient for effect of span height on impact velocity [5].

Chapter 6: PARAMETERS FOR TRAWL GEAR, SEABED SOIL AND PIPELINE

6.0 Introduction

In this chapter, the effect of various parameters regarding trawl equipment, seabed soil and the pipeline will be studied**.** In order to come up with appropriate pullover estimates, the most significant parameters must be identified. Variables for trawl gear, pipeline and seabed soil will be studied and sensitivity analysis will be done during FE analysis by changing one variable as a time and keeping all other parameters at fixed values.
The relevant parameters can be categorized as follows as related to trawling equipment, pipeline and seabed soil:

Trawl gear parameters include:

- Shape and size of trawl gear
- Trawling velocity
- Trawl gear impact frequency
- Warp line length (or warp line stiffness)

Pipeline parameters include:

- Span height
- Pipeline initial condition
- Pipeline flexibility

Seabed soil parameters include:

- Soil friction
- Sea bed stiffness
- Seabed unevenness

6.1 Trawl gear parameters

6.1.1 Shape and size of trawl gear

Shape and size of trawl gear have direct impact on the amount of pullover forces on the pipeline. For nice rounded shapes the transition is smooth and the resulting pullover force is relatively small. Similarly for collision with large clump weights, large pullover forces are expected.

6.1.2 Trawling velocity

This parameter can be directly associated with pullover duration. For low velocity values, duration becomes longer. Resulted durations can then be associated with pullover forces.

6.1.3 Trawl gear impact frequency

This is the expected frequency of trawl gear crossing over the pipeline. A large frequency of trawling can result in higher probability of interaction. As the number of interactions increase, the effect of pullover forces accumulates and gives high pullover force.

6.1.4 Warp line length (Stiffness)

Warp line length or stiffness has effect on pullover duration similar to trawling velocity. A lower stiffness gives longer time to mobilize the necessary force in the warp line to pull the clump weight over the pipeline.

6.2 Pipeline parameters

6.2.1 Span height

Free spans can be caused by seabed unevenness, change of seabed topology (e.g. scouring, sand waves), artificial supports or rock beams and strudel scours [3].
In this project, trawl interference analyses will be performed for three different span heights namely 0 m, 1 m and 2 m.

6.2.2 Pipeline initial conditions

These include pipeline initial configurations and pipeline content conditions. In reality subsea pipelines on bottom geometry have some lateral imperfections. These lateral imperfections are due to the pipe-lay vessel sway motion during installation process and uneven seabed. In this thesis work, the pipeline is assumed to be straight and laid on even seabed. Hence, there is no on bottom geometry imperfection introduced in the finite element analysis model. Besides, a pipeline; with zero operational pressure and ambient temperature; has been assumed prior to trawl gear interaction.

6.2.3 Pipeline flexibility

This is governed by pipeline diameter, wall-thickness, span length and supporting (fixation) condition. With reference to DNV-RP-F111 code, pull-over loads are valid when the flexibility of the potential free span is low and the pipeline diameter is between 10"and 40".
As stated in section 5.0, a rigid steel pipeline; 200m length, 0.7868m diameter, 0.0192m wall thickness; fixed at both ends is modeled in the FE analysis.

6.3 Seabed soil parameters

6.3.1 Soil friction

In the event of trawling, sea bed friction resulting from soil-pipe interaction can have a major influence on pullover loads when it is in full contact with the seabed as it is develops a lateral restraint.
Besides the soil friction, also for free span pipelines fixed at both ends , has some influence on the pullover loads during interaction as some portion of the load is taken or damped by the soil friction far from the interaction point. But this soil friction effects on free spanning pipelines are negligible and hence the soil friction for free span pipelines is not a significant parameter to consider.
Soil friction model follows Coulomb's law of orthotropic friction model. The orthotropic friction model is based on two coefficients of soil friction, axial friction (friction coefficient in X-direction) and lateral friction (friction coefficient in Y-direction). In the FE analysis SIMLA model friction coefficient of 0.5 and 1.9 are considered in X and Y directions respectively.

6.3.2 Seabed stiffness

Soil stiffness is applicable only to pipelines resting on seabed. No soil stiffness is assumed for pipeline free spans. Hence seabed stiffness is considered during the analysis of pipeline with span heights of 0 m. Seabed stiffness for span heights 1 m and 2 m will not be applicable.

6.3.3 Seabed unevenness

It is obvious that a pipeline can undergo buckling prior being exposed to trawl load when laid on uneven seabed. As mentioned in presenting the boundary conditions for the analysis in section 7.1, a flat sea bed is assumed for this project.

Chapter 7: PULLOVER ANALYSIS AND SIMLA MODEL

7.0 Pullover analysis

During the pullover phase the pipeline may be subjected to relatively large horizontal (lateral) and vertical forces. These forces may be estimated through actual model test results or numerical simulations (i.e. by modeling trawl gear interference as a dynamic load using non-linear finite element analysis). In this thesis simulation models for a pipeline system subjected to pullover loading from commonly as well as recently used Clump weights will be carried out by using the software SIMLA.

The most significant parameters must be identified in order to come up with good pullover force estimates. These parameters can be identified by sensitivity analysis in the FE analysis.

Refer section 6 for the most important parameters used during the analysis.

FE assessment is used to verify the most significant parameter and DNV recommendations. Refer figure 7-1 for an overview of the analysis process for trawl gear interaction with subsea pipelines.

As in DNV-RP-F111 [5], all relevant non-linear effects shall be taken into account during the analysis of trawl gear pullover. For example, displacements including geometrical stiffness and non-linear material behavior shall be accounted for.

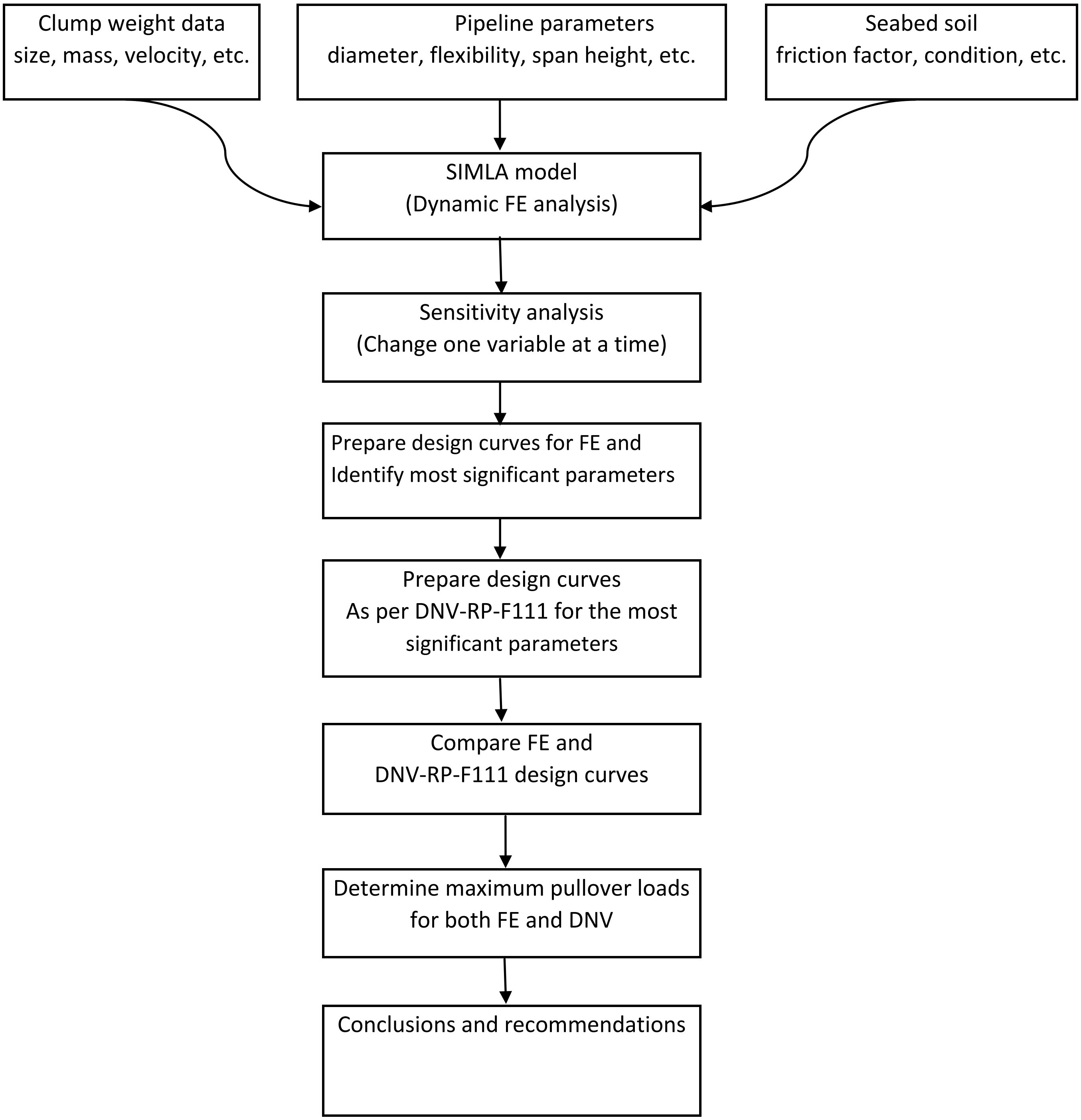


Figure 7-1: Overview of the analysis process as used in this report for trawl gear interaction with subsea pipelines.

7.1 Limitations of the analysis

During pullover analysis, the following assumptions (boundary conditions) are made:

a. No Initial random on bottom geometry imperfection (out of straightness) from laying.

b. Only external interference from trawl pullover is considered. That is, no buckling due to functional loads (temperature and pressure), or prior interference. In other words the pipeline is considered to have negligible compressive forces due to thermal and internal pressure effects.

c. Analyses are based on zero operational pressure and ambient temperature.

d. No hydrodynamic forces (environmental loads) are considered.

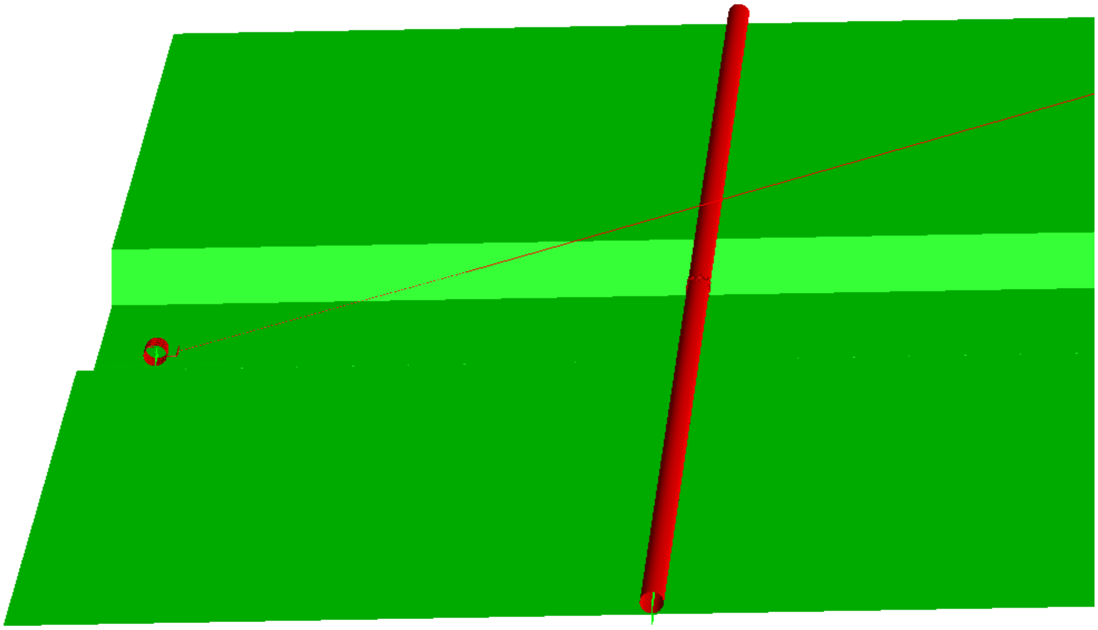
e. Rigid and fixed pipeline sections at relatively intermediate water depths are considered. i.e. water depths of 220m and 330m.

f. Pipeline is modeled with sufficient length to ensure that end effects have no influence. i.e. the end is anchored far away for there to be no forces on the anchor due to pullover. In other words contribution from effect of soil friction faraway is negligible.

7.2 SIMLA model

A 200m straight rigid steel pipeline is modeled with 0, 1 and 2m span heights. The pipeline has 0.7868m diameter and 0.0192m wall thickness and density of 7850kg/m^3. The clump weight is also modeled as a pipeline with 0.76m diameter and 0.01m wall thickness. The clump weight has a length of 2.5m. Three different clump weights are considered during the analysis; namely clump weights with masses of 4500kg, 6750kg and 9000kg; to cover commonly and recently used clump weights in the Norwegian Sea and also to investigate the significance of increase in weight by 50 % and 100% of the commonly used clump weights (i.e. clump weight with mass of 4500kg).

Refer to figures 7-2a, 7-2b and 7-2c for the behavior of the free span pipeline during three different scenarios that are: before, during and after collisions. In this model, a tangential pullover is considered, which means clump weight crosses the pipeline at 90 degrees.



Figurer 7-2a: Clump weight and pipeline simulation before collision

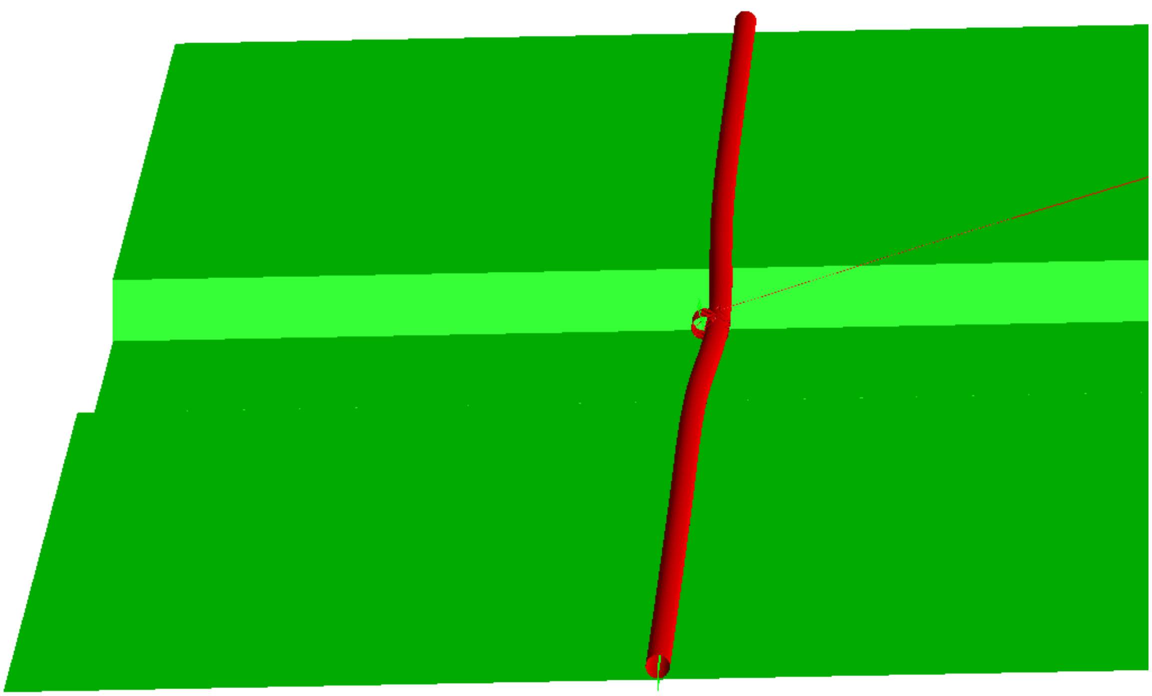


Figure7-2b: Clump weight and pipeline simulation during collision

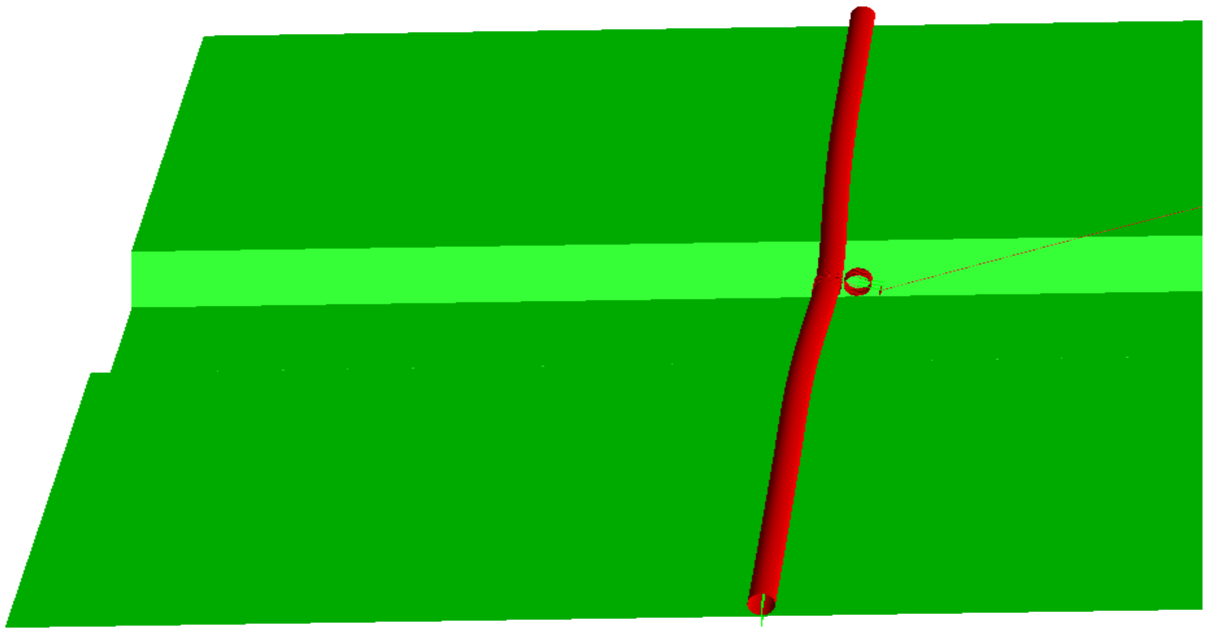


Figure7-2c: Clump weight and pipeline simulation after collision

Chapter 8: RESULTS

All results from analysis using commonly and recently used clump weights and DNV-RP-F111 are contained within this chapter. As mentioned in chapter 6, the most significant parameters must be identified. In this section, such above parameters are identified and their corresponding results are used for DNV-RP-F111 comparisons.

Results from the analysis are also supported by comments.

The output from SIMLA includes horizontal pullover forces, vertical pullover forces and horizontal displacements.

The vertical pullover forces in this chapter are defined to be positive upwards.

Vertical pipeline displacements were found to be small in all simulations. Hence they are not included among the results, while horizontal displacements resulting from interaction of clump weights and the subsea pipeline and a table that summarizes maximum pullover forces due to change in parameters are presented in appendices. Besides it should be noted that screenshots of some simulations for the pipeline with, and without the presence of axial compressive force are presented in Appendix C.

In this chapter interference analyses are performed for different span heights namely 0, 1m and 2m. Similarly, analyses are performed for different clump weight masses, warp line lengths and trawling velocities.

Finally additional results are added in order to understand the pipeline conditions when the pipeline is in operating condition prior to the interference with the trawl gear.

8.1 Span height of 2m and with different trawling velocities.

Refer figure 8-1 for displacement versus time plot prepared for velocities input to the SIMLA model. Three cases are shown in figures 8-2a and 8-2b with different trawling velocities. The velocities considered are 1.65m/s, 2.3 m/s and 3.0 m/s, while other parameters are the same for all cases. Velocities are increased by 40% and 80% from base velocity of 1.65m/s in order to see the effects of the maximum pullover force. For low velocity values, the durations become longer, as discussed in the theory part of section 6.1. Moreover, 40% and 80% increases in velocity result in 20% and 55% increase in the maximum horizontal pullover force of clump weights respectively.

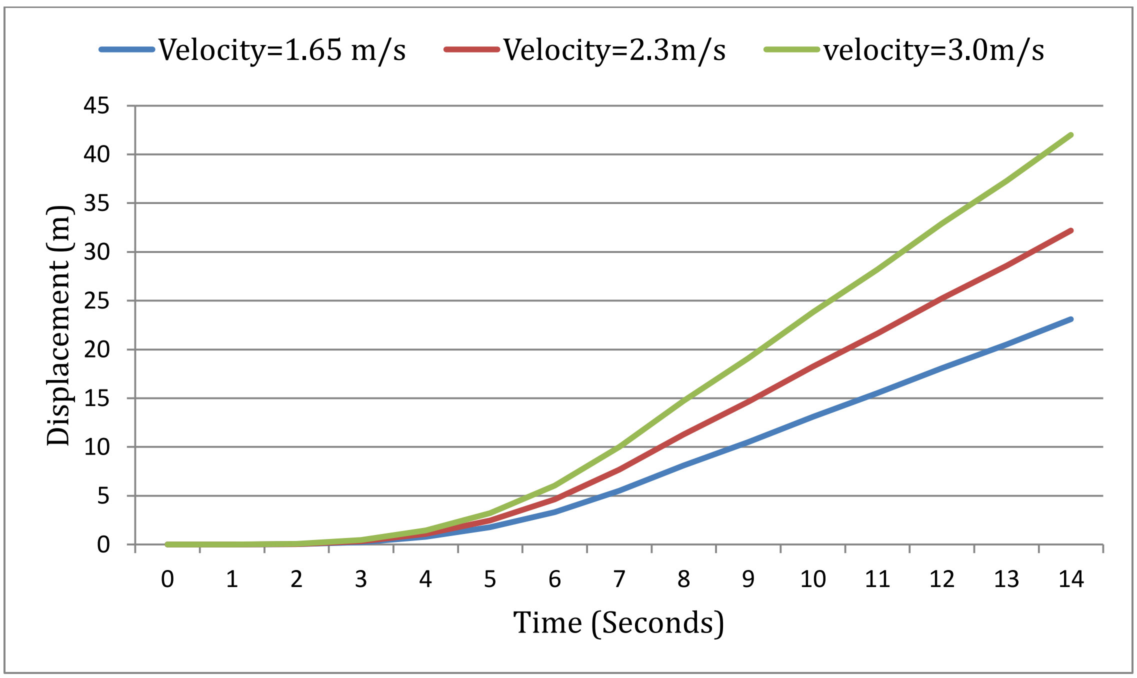


Figure 8-1: Displacement vs. Time

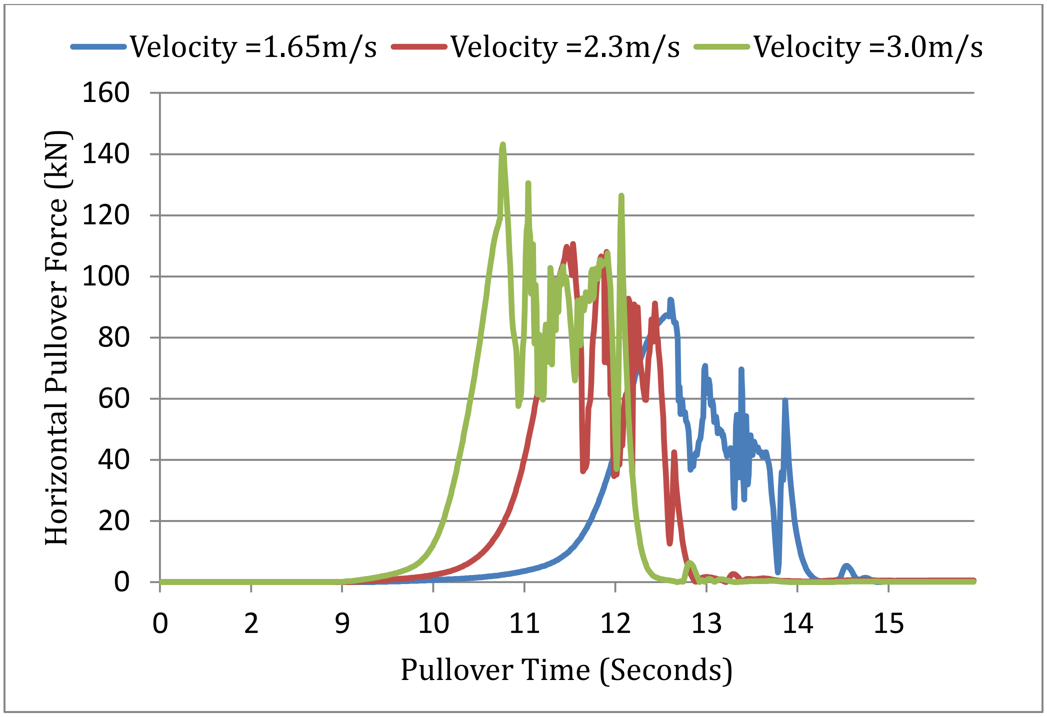


Figure 8-2a: Horizontal pullover force for different trawling velocities (4500kg Clump weight)

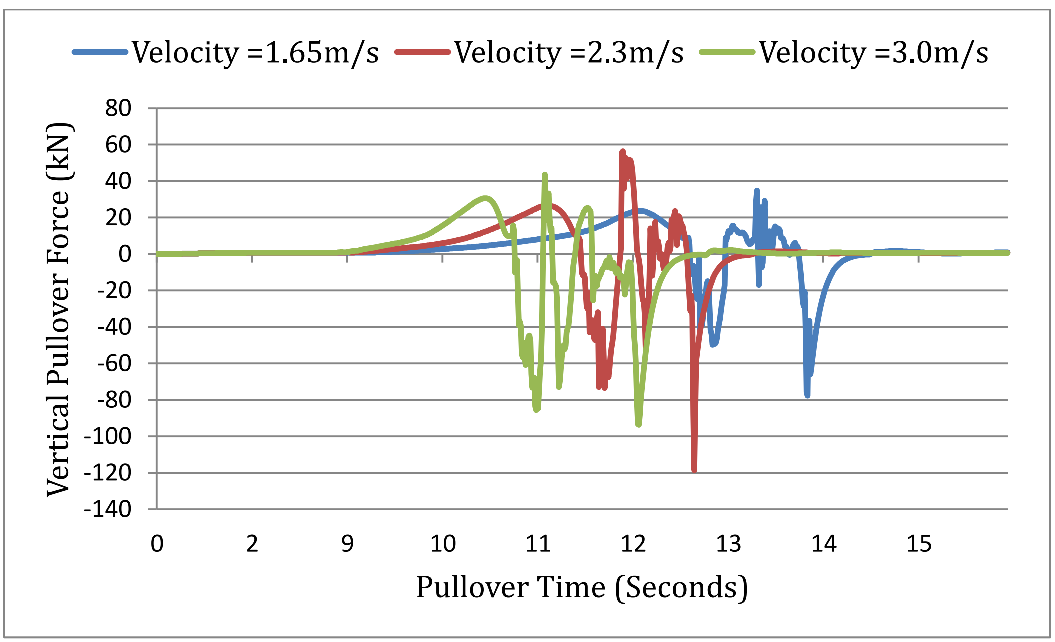


Figure 8-2b: Vertical pullover force for different trawling velocities (4500kg Clump weight)

8.2 Span Height of 2 m and with different Warp line lengths

As shown in figures 8-3a and 8-3b, warp line lengths are different when other parameters like clump weight mass, span and trawling velocity are same. Comparing the values for the maximum horizontal pullover force, it is obtained that 50% increase in warp line length increases the maximum horizontal pullover force by 53%. This is similar to the velocity parameter, a longer warp line length (lower stiffness) gives longer time to mobilize the force in the warp line to pull the clump weight over the pipeline.

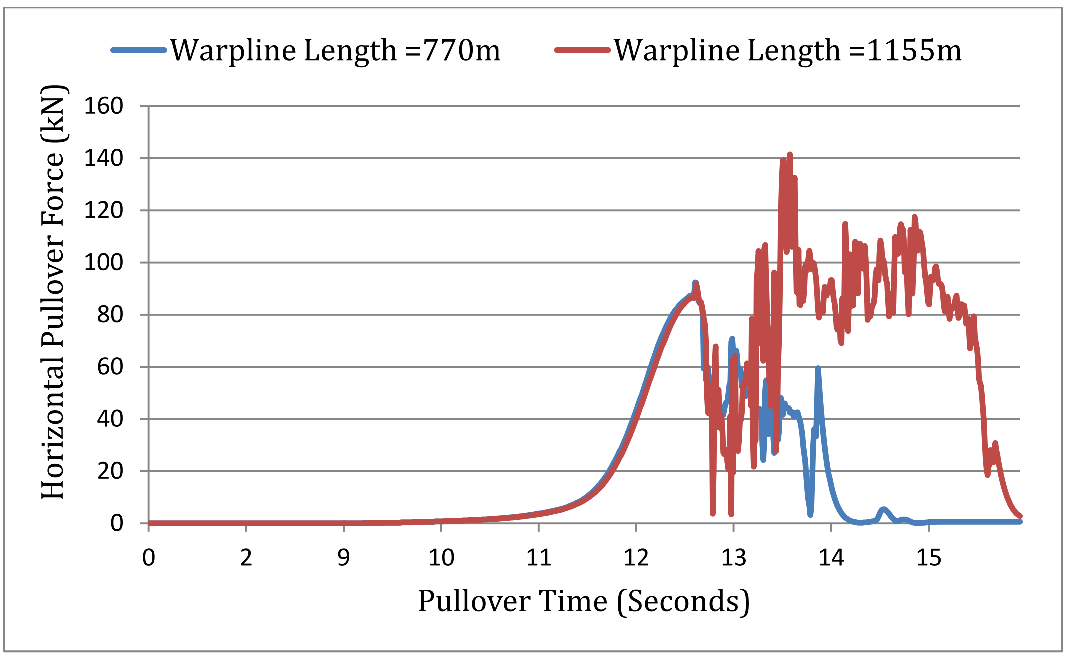


Figure 8-3a: Horizontal Pullover force for different warp line lengths (4500kg Clump weight)

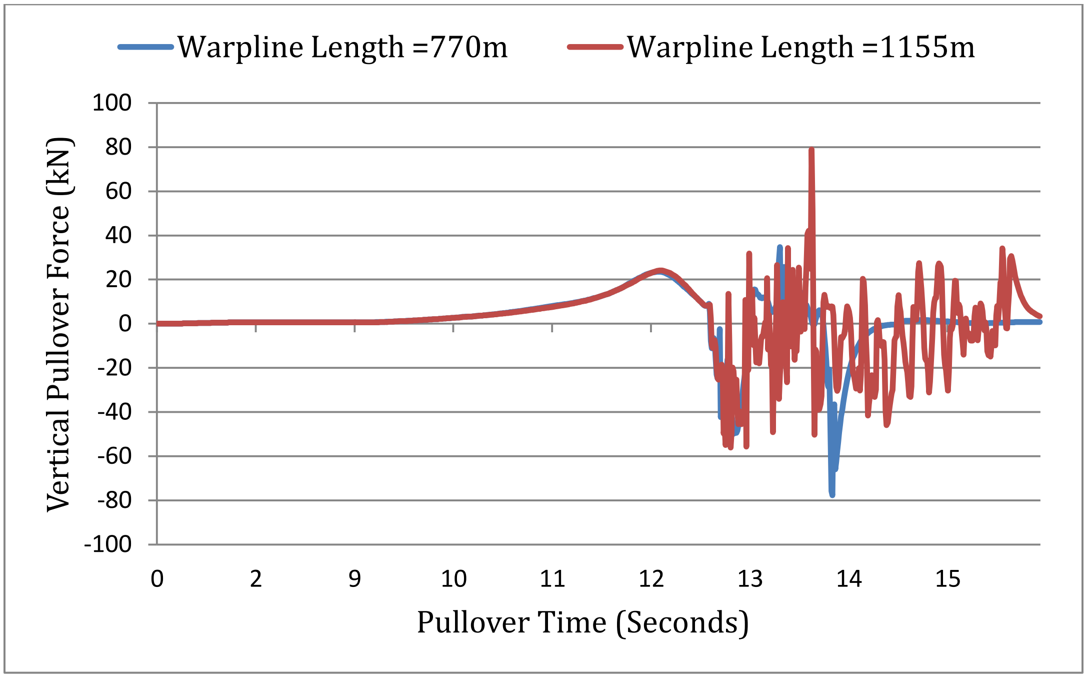


Figure 8-3b: Vertical Pullover force for different warp line lengths (4500kg Clump weight)

8.3 Span Height of 0 m and with different Clump weight masses

In figures 8-4a and 8-4b pullover force for different clump weight masses are shown. Basically commonly (4500kg) and recently (9000kg) used clump weights are shown. Span height of 0 m and trawling velocity of 3m/s are considered during the simulation. All other parameters are identical except clump weight masses. As shown in the figures below the horizontal pullover forces greatly depends on the masses of the clump weight. With 50 % and 100% increase in the masses of the clump weight, the maximum horizontal pullover forces increase by 12 % and 34% respectively with pipeline geometry resting on ground (i.e. with 0 m span height). These show that the pullover of the clump weight is a quasi-static response where the clump weight pulls the pipeline by rotation, sliding or a combination of both.

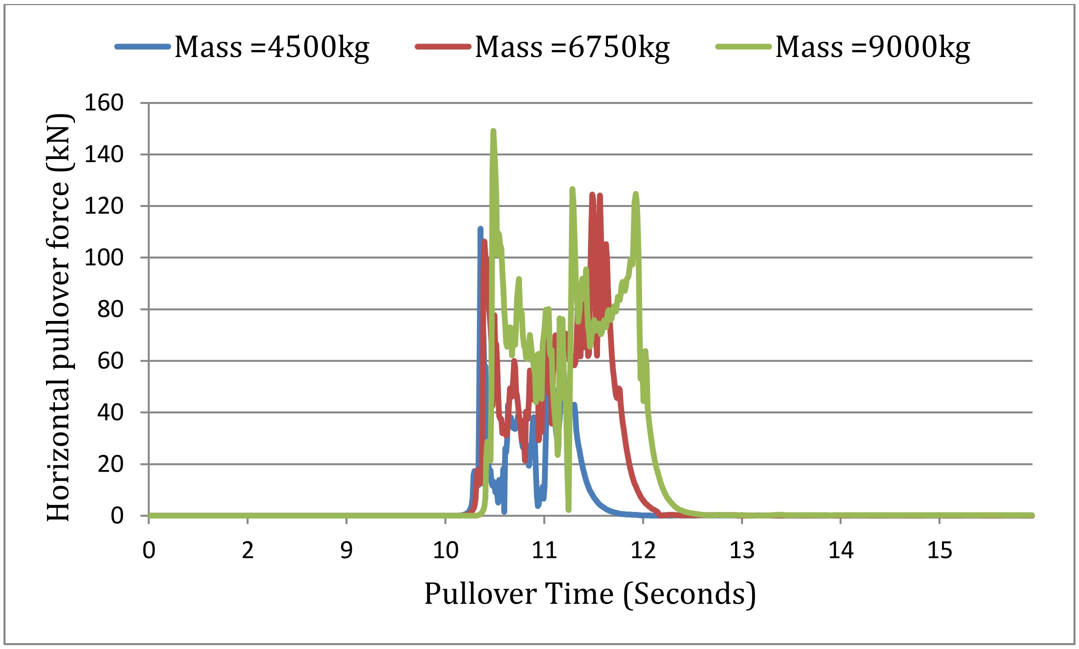


Figure8-4a: Horizontal Pullover force for different Clump weight masses (span height 0 m)

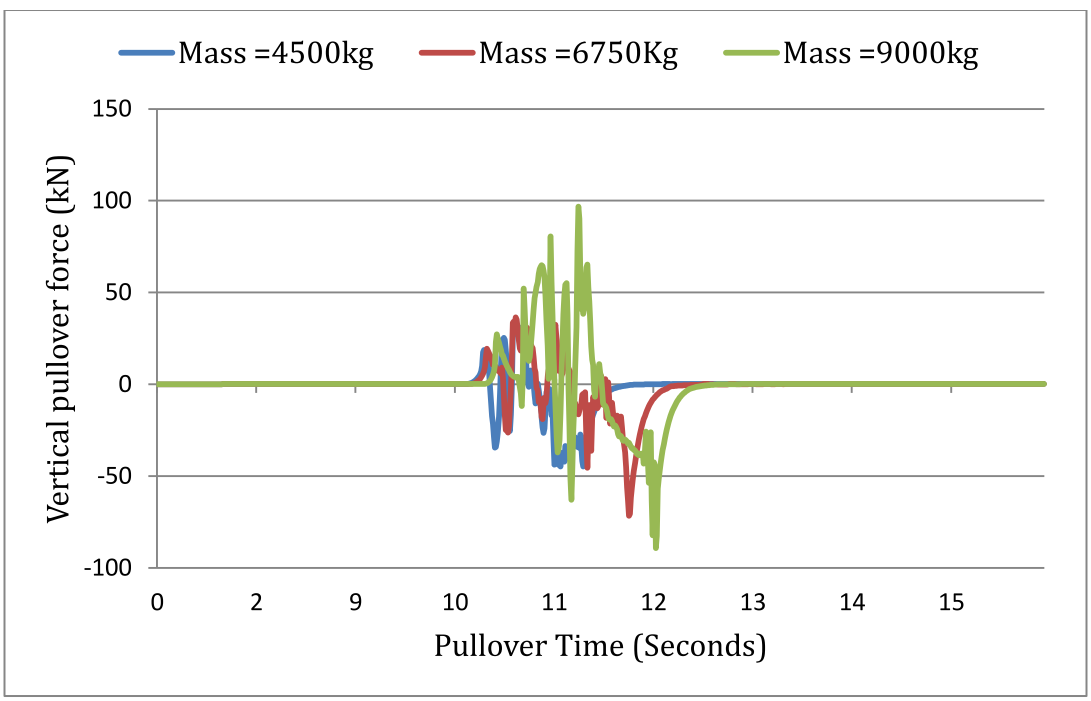


Figure 8-4b: Vertical Pullover force for different Clump weight masses (span height 0 m)

8.4 Span Height of 2 m and with different Clump weight masses

Similar to section 8.3, pullover forces for different clump weight masses are shown in figures 8-5a and 8-5b. In here only the span height is modified from 0 m to 2 m in order to see the effects of clump weight masses in combination with span heights. All other parameters are identical, with span height of 2 m, except clump weight masses. As shown in the figures below the horizontal pullover forces depends on the masses of the clump weight and span height. With 50 % and 100% increases in the mass of the clump weight, the maximum horizontal pullover forces increases by 27% and 67% respectively for span height of 2m. These show that the maximum horizontal pullover forces for span height of 2m are almost doubled compared with span height of 0m for same mass of clump weight.

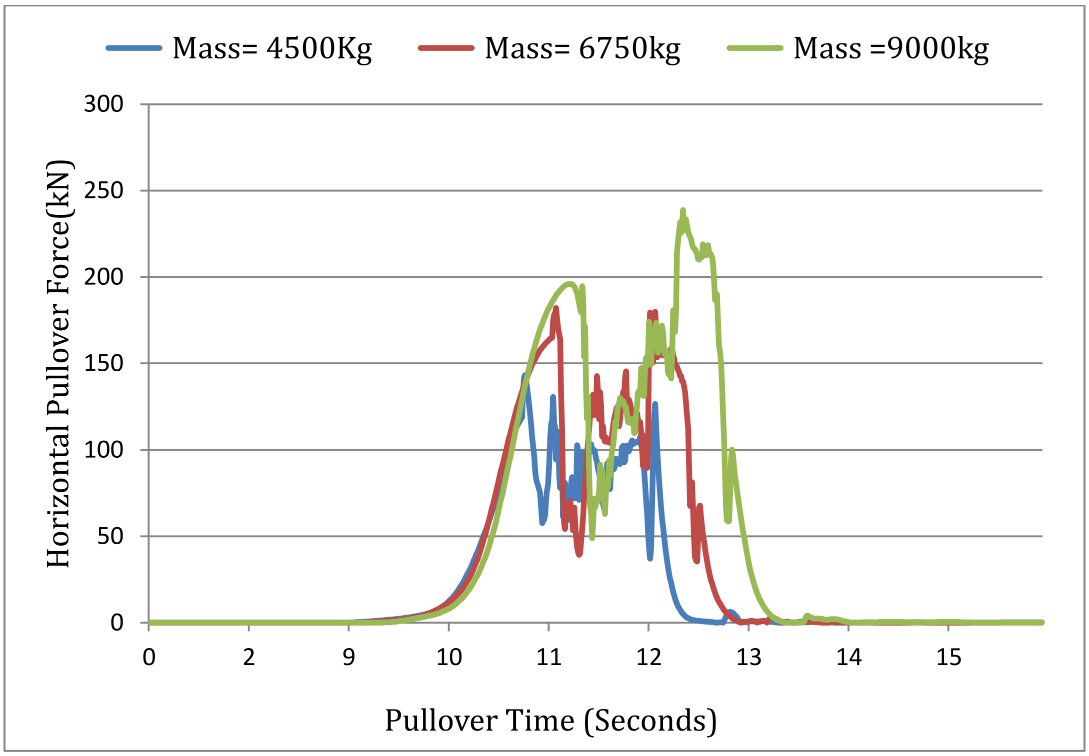


Figure8-5a: Horizontal Pullover force for different Clump weight masses (span height 2 m)

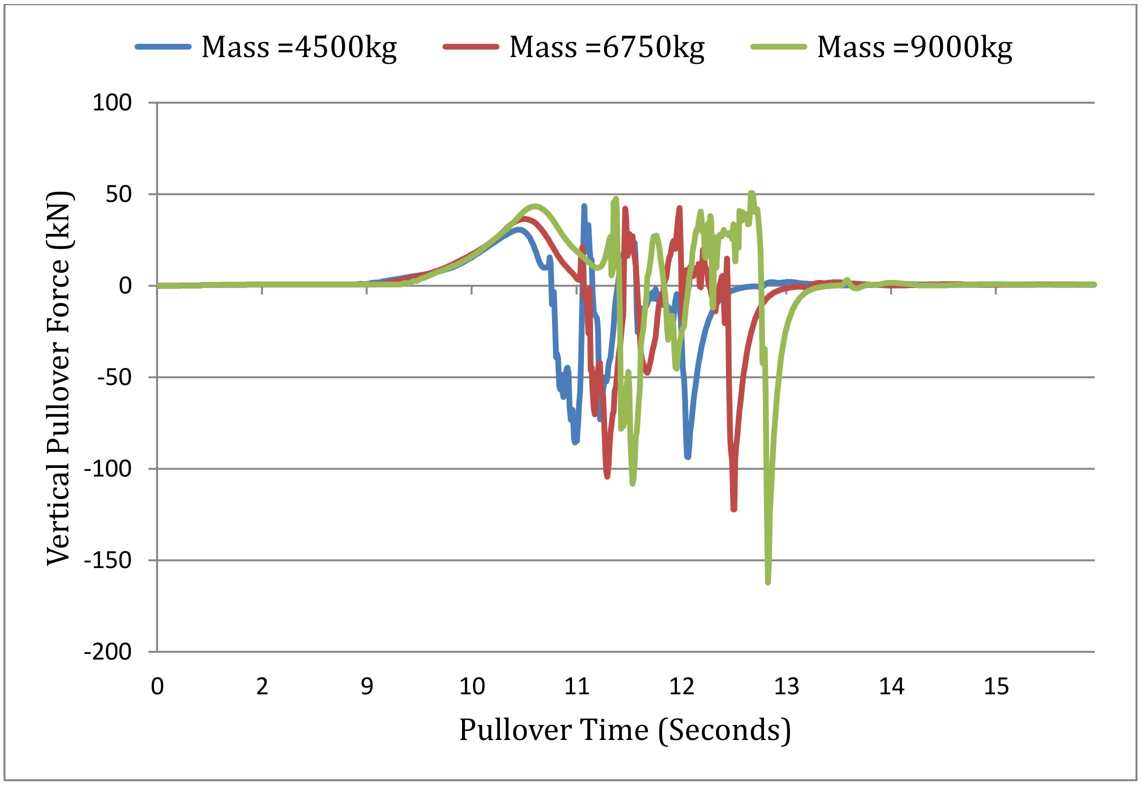


Figure 8-5b: Vertical Pullover force for different Clump weight masses (span height 2 m)

8.5 Clump weight mass of 4500kg and with different span height.

A comparison of the pullover forces for different pipeline span heights are shown in figures 8-6a and 8-6b when crossed by a clump weight mass of 4500kg. As shown, the horizontal pullover forces increase with span height. When comparing span height of 0 m with span heights of 1m and 2m, the maximum horizontal pullover forces increased by 7% and 29% for the same clump weight mass and trawling velocity.

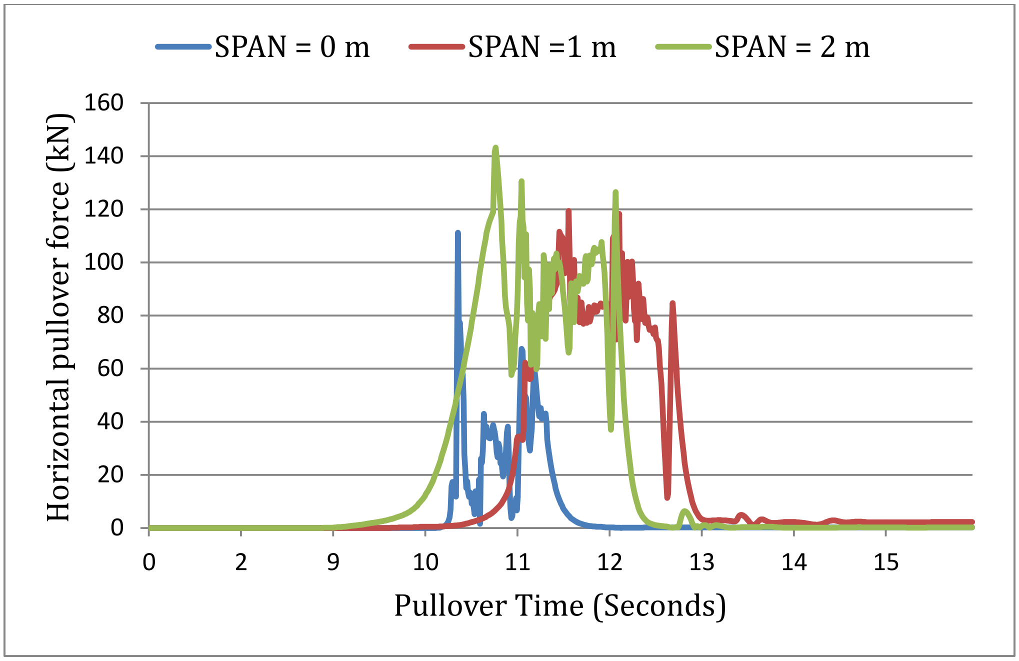


Figure8-6a: Horizontal Pullover force for different span heights (4500kg Clump weight)

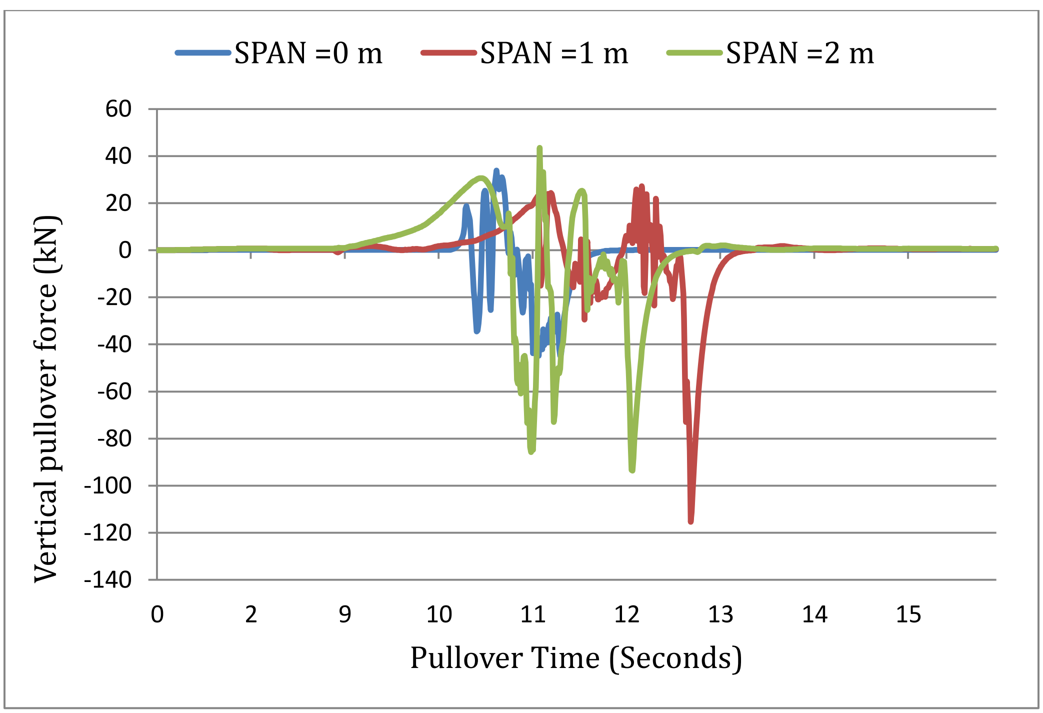


Figure 8-6b: Vertical Pullover force for different span heights (4500kg Clump weight)

8.6 Pullover loading from DNV

The pullover forces for span heights of 0 m and 1 m were determined for clump weights with different masses. The clump weight masses are 4500kg, 6750kg and 9000kg. These masses were selected to cover the common and recently used clump weights in the North Sea and the Norwegian Sea.
The maximum horizontal and vertical pullover forces were determined from the formula and data presented in section 5. Trawling velocity of 3m/s was used during the determination of pullover forces in order to compare the values with results obtained from the FE analysis.
Refer tables 8-1 and 8-2 for the pullover forces for span heights of 0 m and 2 m.

Clump weight mass	Horizontal pullover force	Downward pullover force
4500 kg	101.54 kN	38.41 kN
6750 kg	152.32 kN	57.61 kN
9000 kg	203.09 kN	76.81 kN

Table 8-1: Pullover forces according to DNV-RP-F111 for span height of 0 m

Clump weight mass	Horizontal pullover force	Downward pullover force
4500 kg	159.23 kN	32.64 kN
6750 kg	238.85 kN	48.95 kN
9000 kg	318.46 kN	65.27 kN

Table 8-2: Pullover forces according to DNV-RP-F111 for span height of 2 m

8.6.1 DNV-RP-F111 versus SIMLA model with span height of 0 m

Referring figures 8-7a, 8-7b, 8-8a, 8-8b, 8-9a and 8-9b for span height of 0 m, it was found that the maximum horizontal pullover loads from DNV-RP-F111 are about 20% and 35% more than that obtained from the FE analysis for clump weight masses of 6750kg and 9000kg respectively, while it is 9% less for 4500kg clump weight. However, the maximum vertical pullover forces from the FE analysis are 1.2 times those of the vertical pullover forces from DNV-RP-F111, although still lower than the horizontal pullover forces.

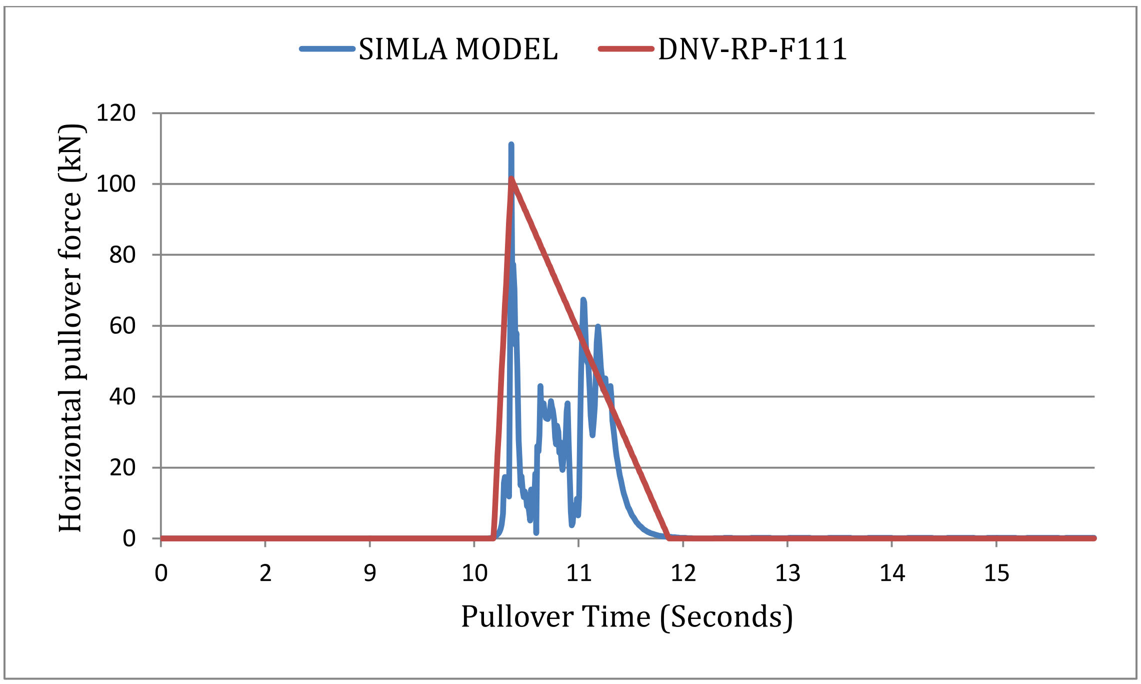


Figure 8-7a: Horizontal Pullover force, DNV-RP-F111 vs. SIMLA model for 4500kg clump weight with span height of 0 m.

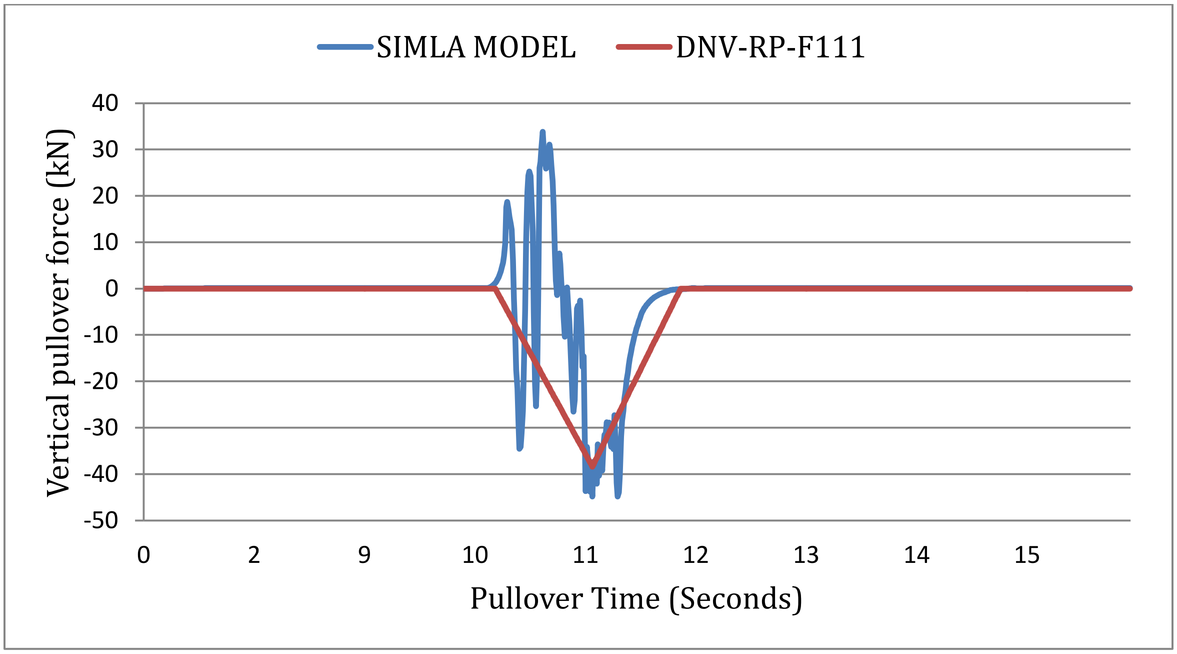


Figure 8-7b: Vertical Pullover force, DNV-RP-F111 vs. SIMLA model for 4500kg clump weight with span height of 0 m.

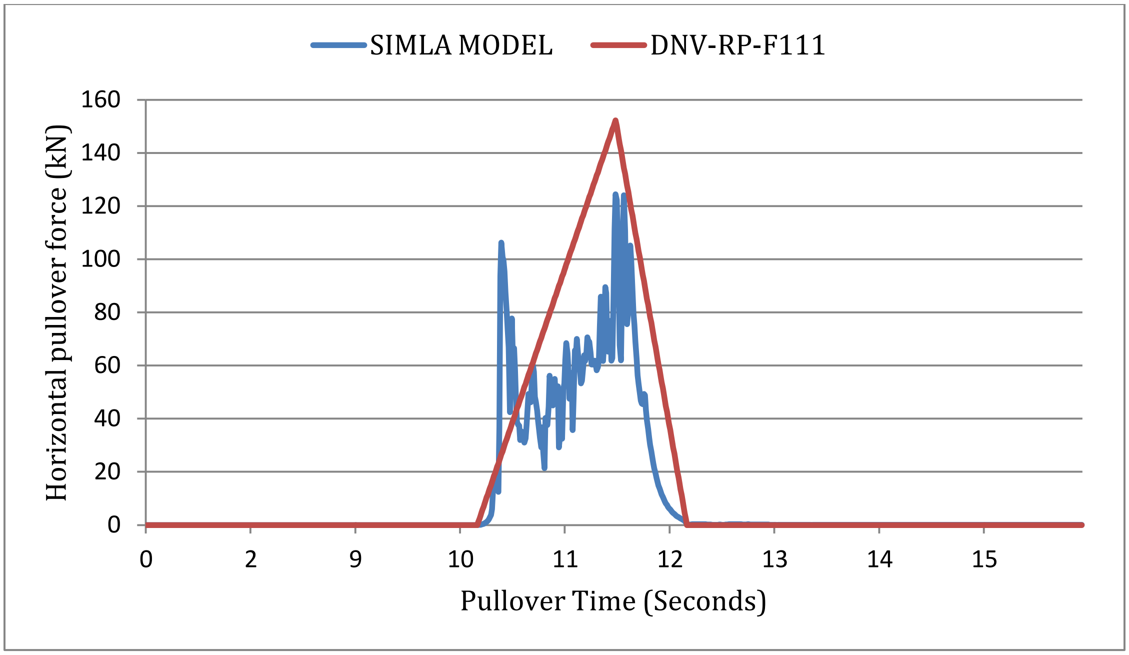


Figure 8-8a: Horizontal Pullover force, DNV-RP-F111 vs. SIMLA model for 6750kg clump weight with span height of 0 m.

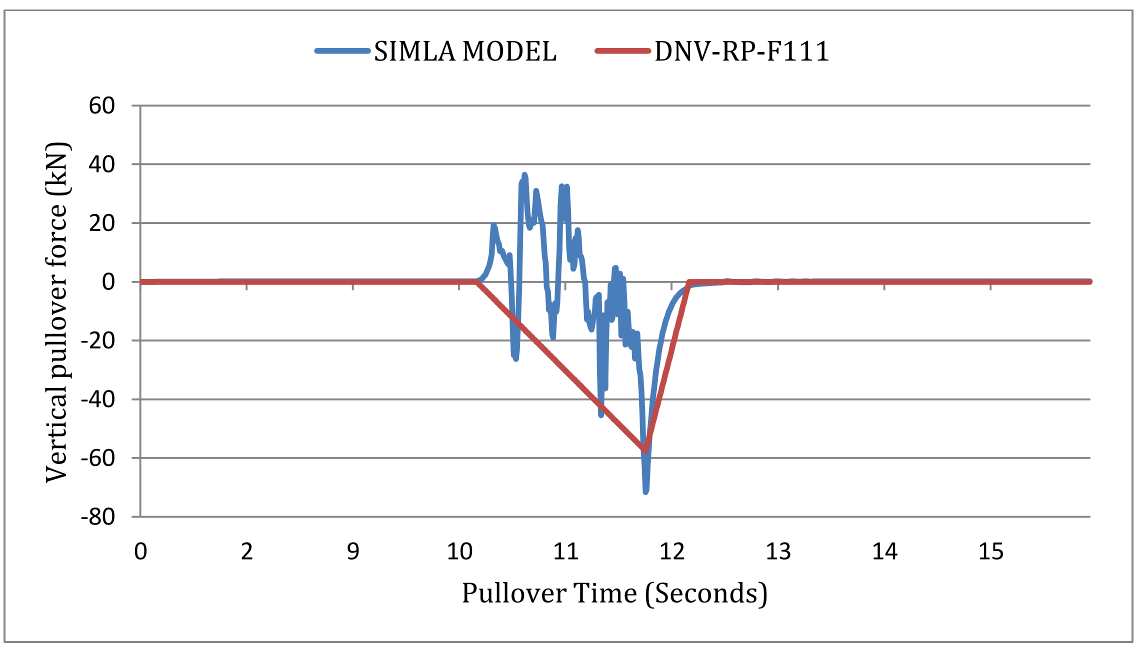


Figure 8-8b: Vertical Pullover force, DNV-RP-F111 vs. SIMLA model for 6750kg clump weight with span height of 0 m.

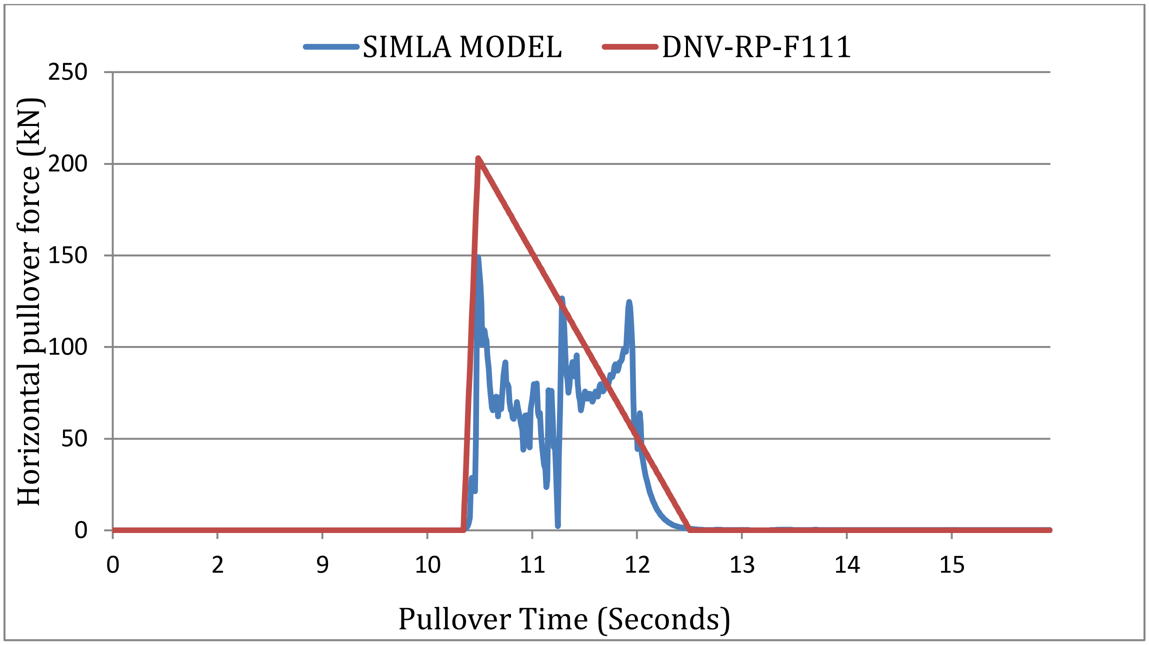


Figure 8-9a: Horizontal Pullover force, DNV-RP-F111 vs. SIMLA model for 9000kg clump weight with span height of 0 m.

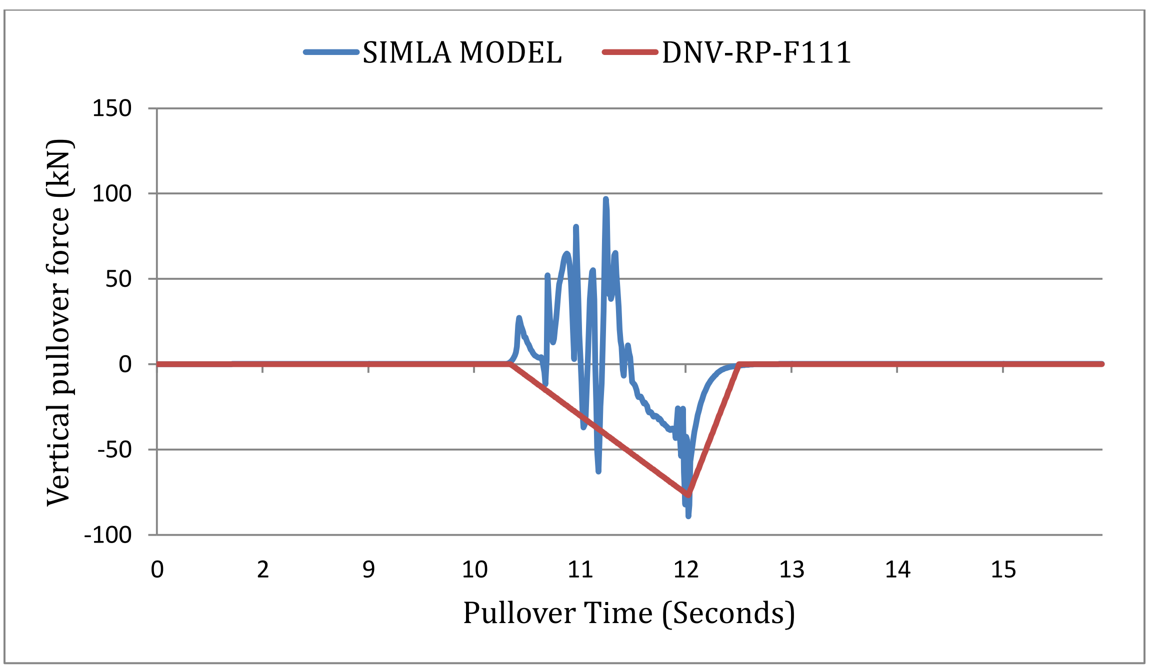


Figure8-9b: Vertical Pullover force, DNV-RP-F111 vs. SIMLA model for 9000kg clump weight with span height of 0 m.

8.6.2 DNV-RP-F111 versus SIMLA model with span height of 2 m.

For span height of 2 m, refer figures 8-10a, 8-10b, 8-11a and 8-11b. The maximum horizontal pullover loads from DNV-RP-F111 are about 10% more than that of maximum horizontal pullover forces obtained by the FE analysis for clump weight mass of 4500kg. Similarly, DNV estimates maximum Horizontal pullover force about 30% larger than that of the FE analysis for clump weight masses of 6750kg and 9000kg. However, the maximum vertical pullover forces from the FE analysis are 2.5 times those of the vertical pullover forces from DNV-RP-F111, although still lower than the horizontal pullover forces.

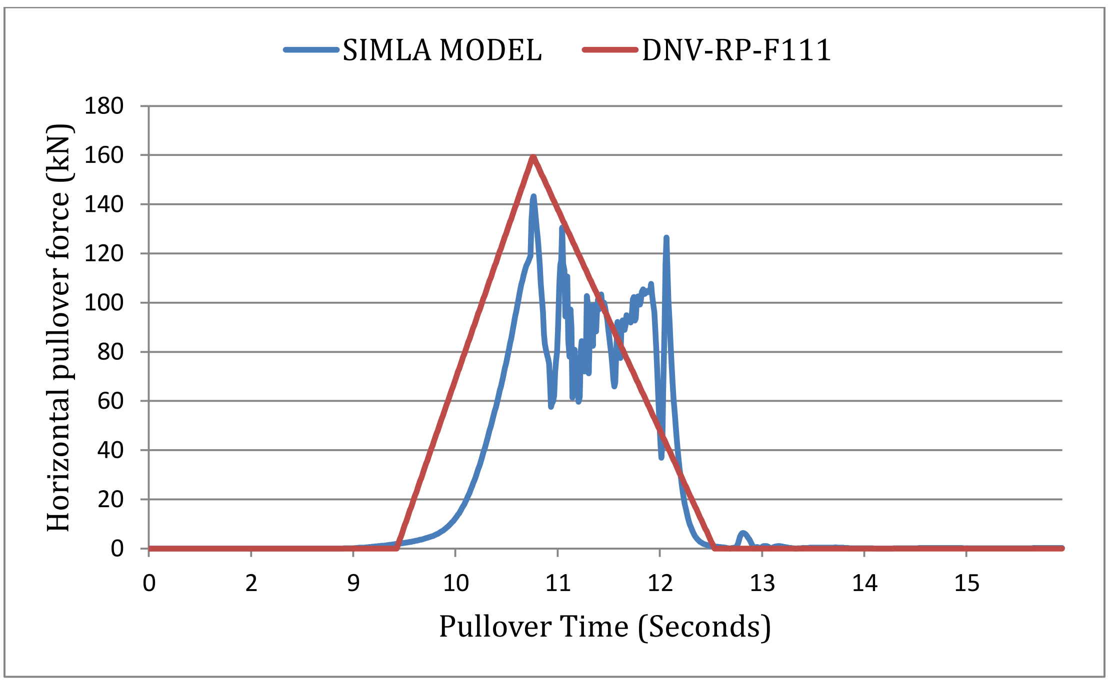


Figure 8-10a: Horizontal Pullover force, DNV-RP-F111 vs. SIMLA model for 4500kg clump weight with span height of 2m.

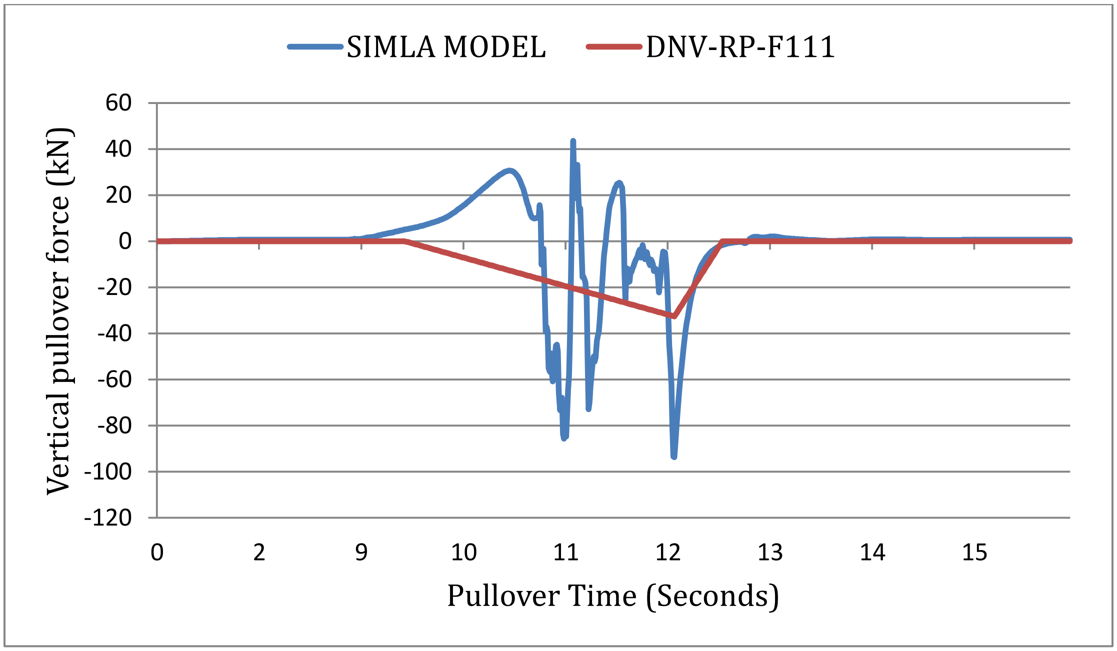


Figure 8-10b: Vertical Pullover force, DNV-RP-F111 vs. SIMLA model for 4500kg clump weight with span height of 2m.

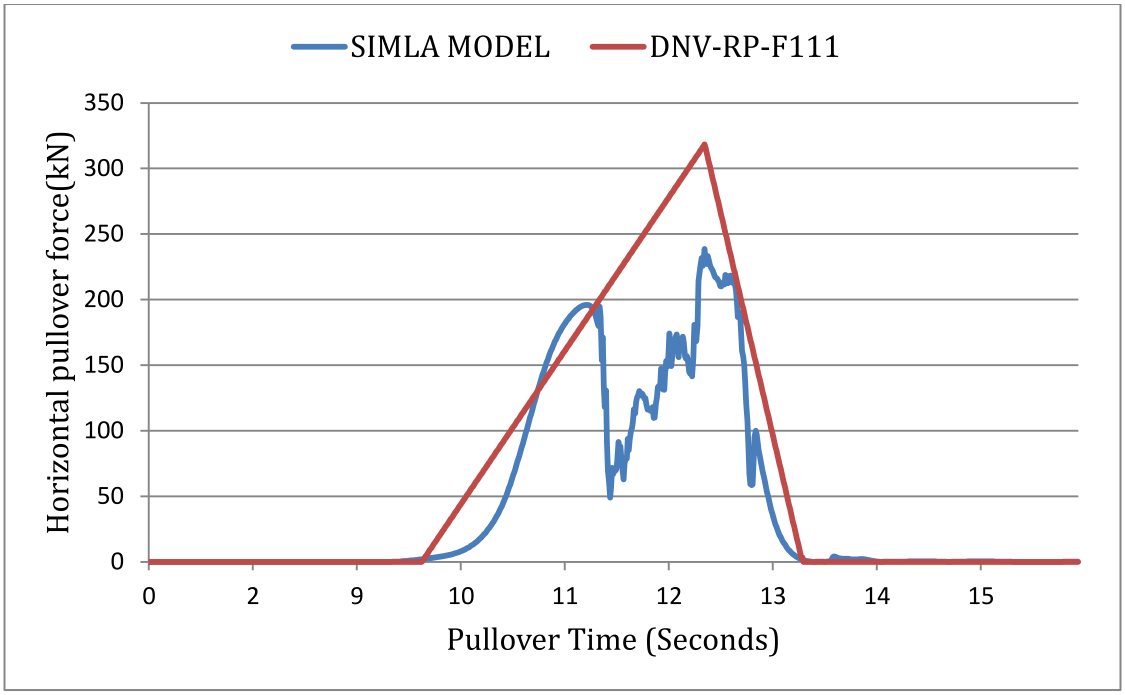


Figure 8-11a: Horizontal Pullover force, DNV-RP-F111 vs. SIMLA model for 9000kg clump weight with span height of 2m.

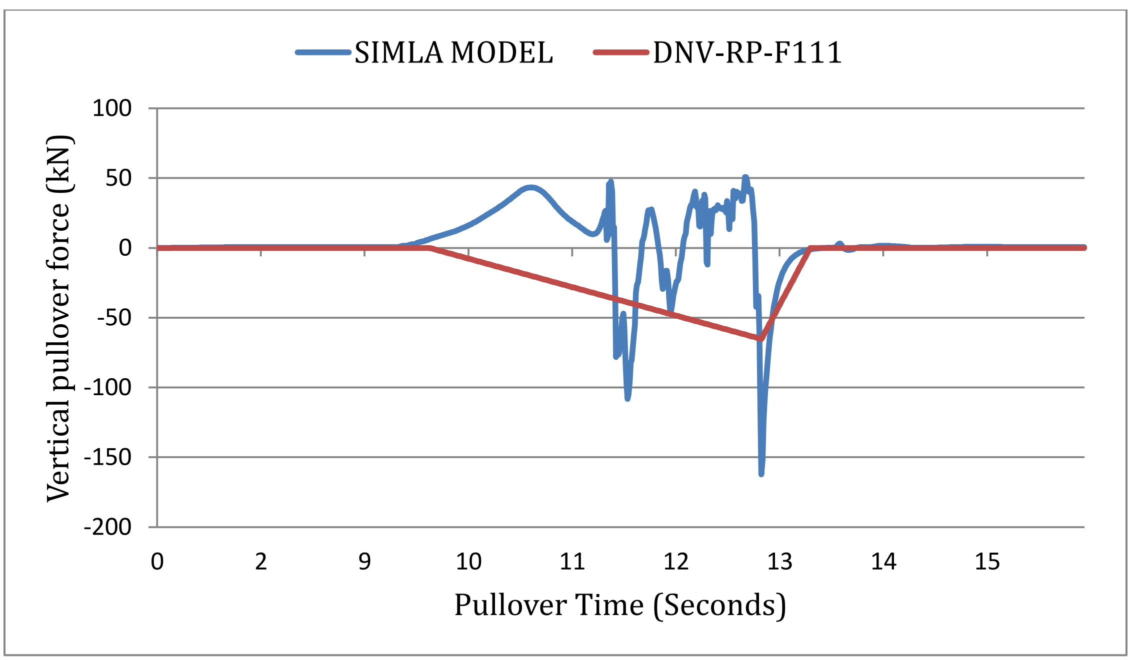


Figure 8-11b: Vertical Pullover force, DNV-RP-F111 vs. SIMLA model for 9000kg clump weight with span height of 2m.

8.7 Span height of 0 m with and without axial compressive force

In these cases simulations of the pipeline with span height of 0 m are performed. The pipeline is in operation condition that is at its full design pressure and ambient temperature, and with operating content prior to pullover. Refer table 8-3 below for the subsea pipeline operational data taken for the analysis. Due to the operating conditions, the pipeline initiates buckling prior to trawling. Therefore, as shown in figure 8-12, the maximum horizontal displacement is found to be six times more than that of a pipeline which is not under operating condition during collision with the clump weight. Referring figures 8-13a and 8-13b, there are no considerable differences with regard to the pullover force obtained from the FE analysis for both cases.

Description	Unit	Value
Content density	kg/ m^3	900
Design pressure	MPa	15
Operating temperature	^oC	100
Ambient temperature	^oC	5

Table 8-3: Pipeline operational data.

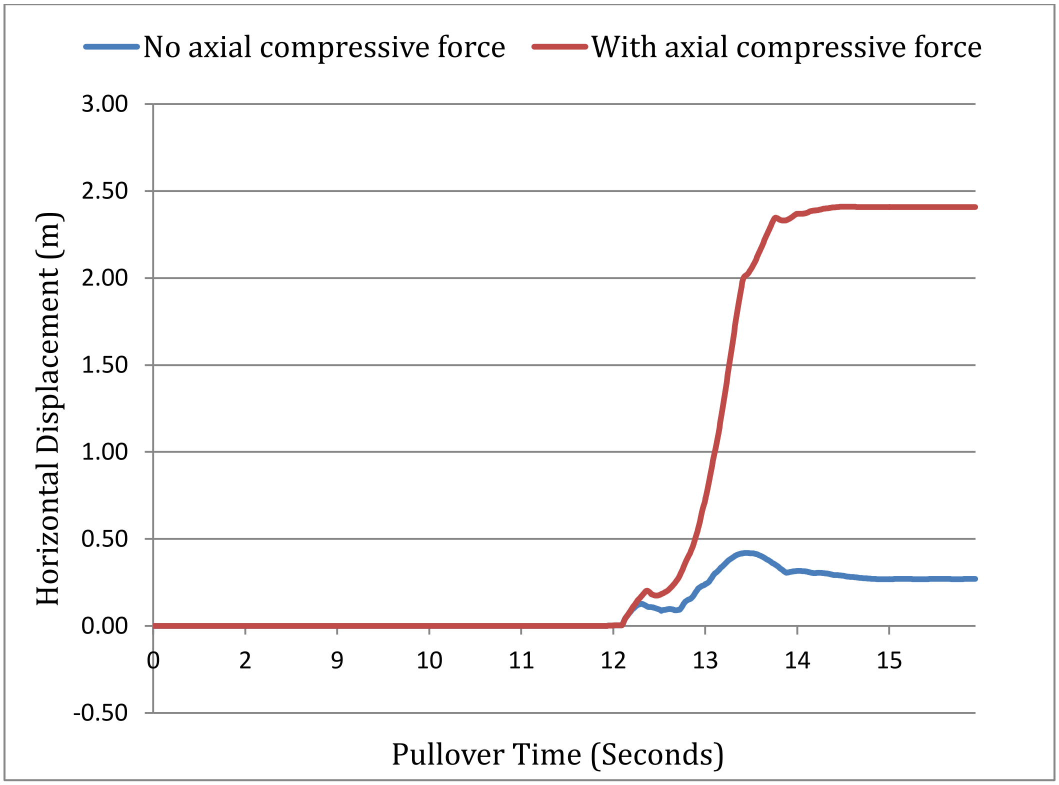


Figure 8-12: Horizontal displacement, Pipeline with axial force vs. with no axial force for 4500kg clump weight with span height of 0 m.

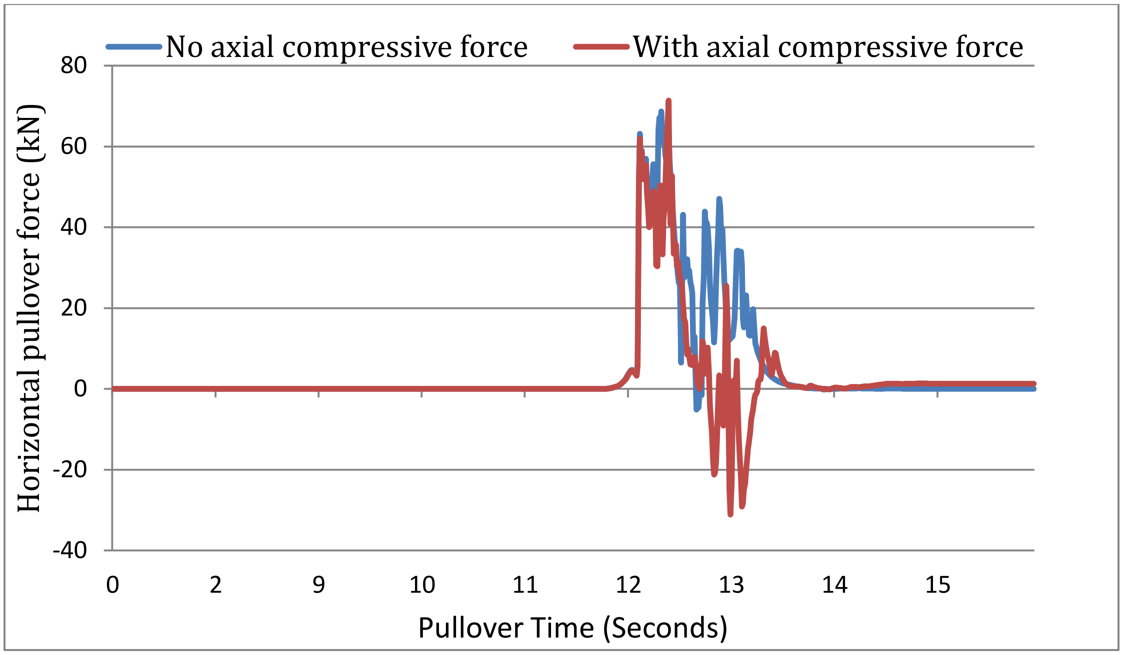


Figure 8-13a: Horizontal pullover force, Pipeline with axial force vs. with no axial force for 4500kg clump weight with span height of 0 m.

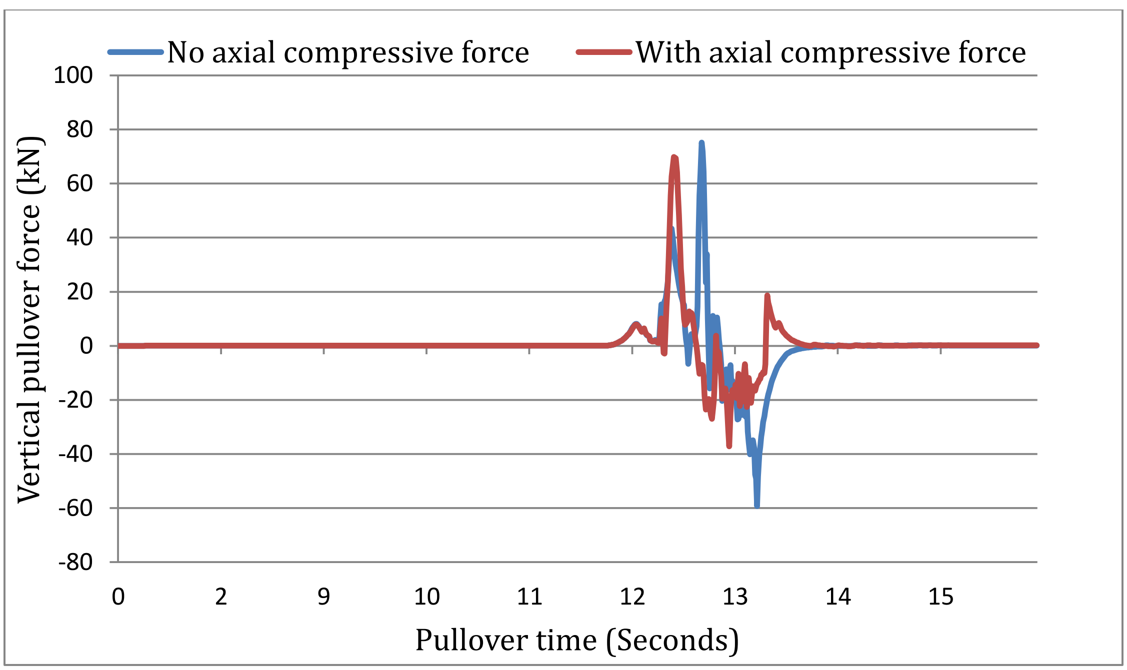


Figure 8-13b: Vertical pullover force, Pipeline with axial force vs. with no axial force for 4500kg clump weight with span height of 0 m.

8.8 Span height of 2 m with and without axial compressive force

When the pipeline is on free span, the horizontal displacement with axial compressive force is doubled compared with no axial compressive force as shown in figure 8-14 below.
The increase in the horizontal displacement is not as exaggerated as for the pipeline with span height of 0 m. This is due to the fact that some displacements are taken place downwards prior the horizontal displacements.
Similar to section 8.7, there are no big differences with regard to pullover forces from the two above cases as shown in figures 8-15a and 8-15b.

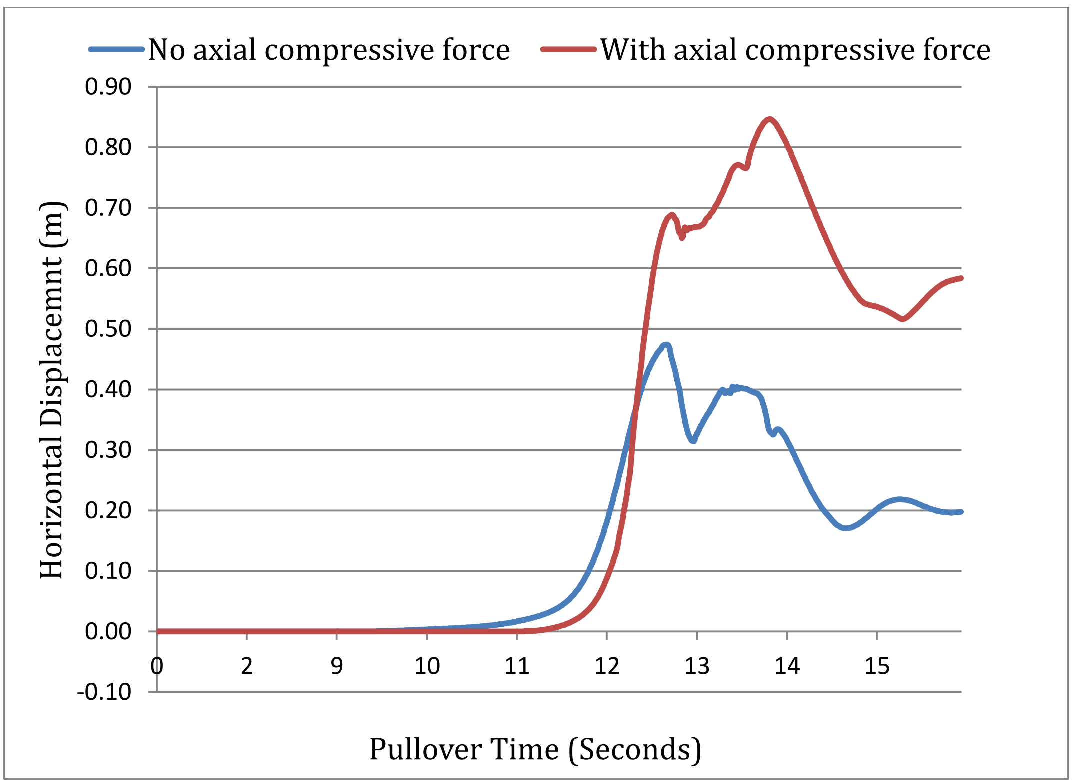


Figure 8-14: Horizontal displacement, Pipeline with axial force vs. with no axial force for 4500kg clump weight with span height of 2 m.

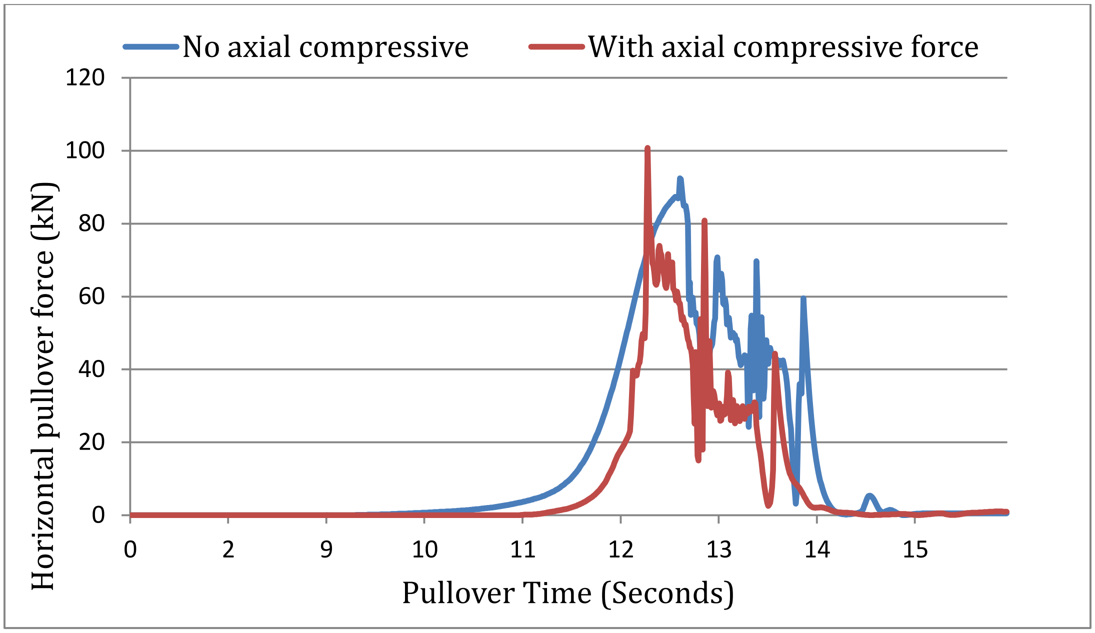


Figure 8-15a: Horizontal pullover force, Pipeline with axial force vs. with no axial force for 4500kg clump weight with span height of 2 m.

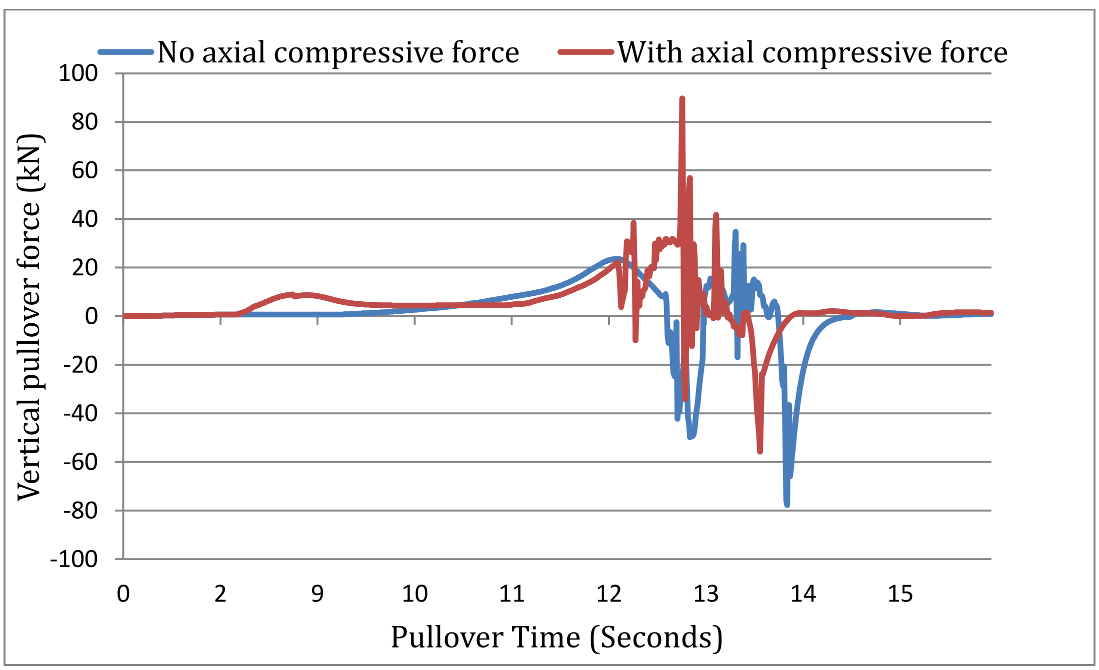


Figure 8-15b: Vertical pullover force, Pipeline with axial force vs. with no axial force for 4500kg clump weight with span height of 2 m.

Chapter 9: CONCLUSIONS AND RECOMMENDATIONS

9.1 Most significant parameters

As obtained from the simulations using commonly and recently used clump weights, the mass parameter in conjunction with the span height plays an important role towards the pullover loads during trawl gear interaction with subsea pipelines. As trawling with clump weights increases the efficiency of trawlers, it is expected to see more and more trawlers which will use clump weights with bigger masses in the future. Clump weights normally give higher pullover loads and with the increase in weight of the clump weight the increase in resulting pullover loads as roughly linear. Meaning when the mass of the clump weight is doubled for pipeline in a span, it is expected also that the pullover loads to be doubled. This proportion of the increase is already shown in the results obtained from the simulations. Hence the loading resulting from trawling gear interaction with subsea pipelines should be reflected in the design of new subsea pipelines as well as whether pipelines should be buried or not. Moreover, reassessment has to be done for pipelines which are already installed for integrity in case if exposed to recently used clump weights.

9.2 DNV-RP-F111 versus SIMLA Model

In this section maximum pullover loads from DNV-RP-F111 are compared against results from the FE analysis software SIMLA for span heights of 0 m and 2 m. During comparison reference are done for only two parameters namely span height and mass. These two parameters are found to be most significant towards pullover forces; therefore, they are used for comparison purposes.

Span height 0 m.

The DNV-RP-F111 code predicts a maximum Horizontal pullover load which is about 30 % higher than the maximum values obtained from clump weight simulations. Therefore, it can be concluded that the DNV code is not fully in line with clump weight simulations for span heights of 0 m. The DNV code is conservative (over predict) as compared with the simulation adopted.
In case of maximum vertical pullover loads, DNV-RP-F111 predicts much lower values than that of clump weight simulations. The vertical pullover loads from simulations are 1.2 times the values found by the DNV code predictions. Hence DNV-RP-F111 estimates values far below and is not in accordance with the simulations; whereas the directions of the vertical loads are same for both cases.

Span height 2 m.

The maximum horizontal pullover force obtained has no significant difference between clump weight simulations and DNV-RP-F111 predictions.
DNV code predicts about 20% more than that of FE simulations, hence it can be said that DNV code is not so conservative as in case of 0 m span.
The maximum vertical pullover force is, however, not in agreement with the DNV-RP-F111 code. It is about 2.5 times larger than the DNV predictions. Hence DNV-RP-F111 estimates are not in accordance with the clump weight simulations with regard to vertical pullover loads; whereas the directions of the vertical loads are same for both cases.

9.3 Effect of axial compressive force together with trawling by Clump weights

Due to the fact that subsea pipelines release compressive axial force or buckles when it is under operating conditions (i.e. due to temperature and pressure loads), the horizontal displacement is more than that of subsea pipelines without axial force prior trawling. It is also critical when the pipeline is not in a free span. Hence attention shall be given to those subsea pipelines lying near subsea structures. Even though it is not included or analyzed here, It is possible to understand from the above results that the most severe condition could be subsea pipelines with a release of compressive force simultaneously with trawling, specifically a pipeline with a span height of 0 m.

9.4 Further Works

In order to draw more accurate conclusions, limitations to the analysis should be as minimal as possible for future works. In such hydrodynamic loads should be included, besides pipelines with different dimensions should be investigated. It is also important to introduce different hit (attack) angles during the simulation of the interaction between clump weight and subsea pipeline. It is also recommended for the future to include the actual sea bed unevenness plus simultaneous occurrences of trawling and release of effective compressive axial force during simulation, as this can have a major impact during the analysis of trawl gear interaction with subsea pipelines with span height 0 m.

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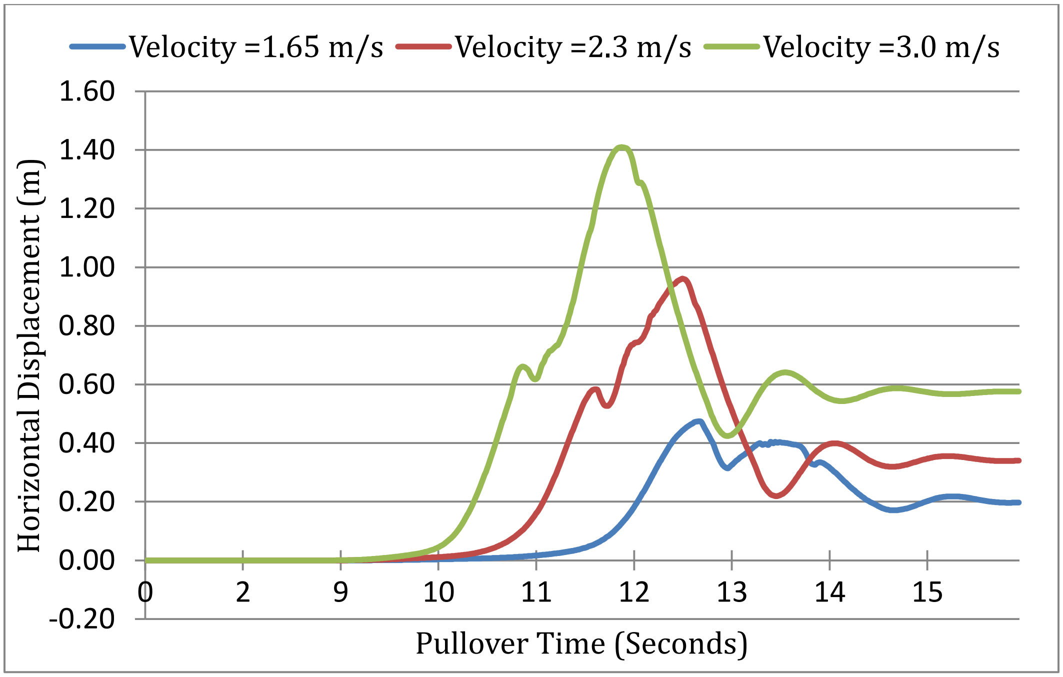
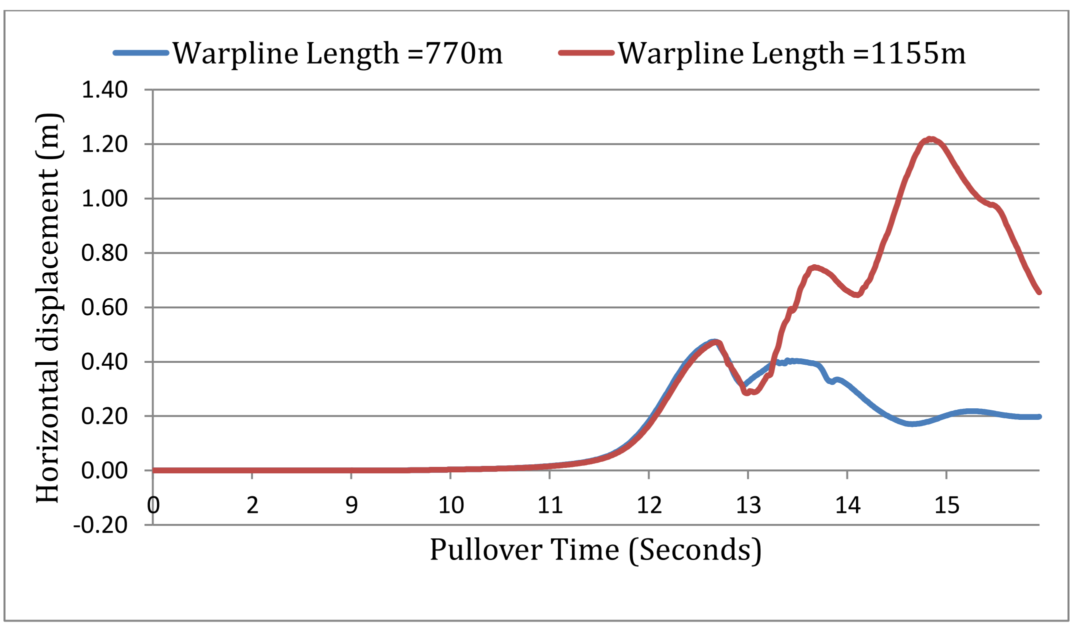
Appendix A: RESULT FIGURES

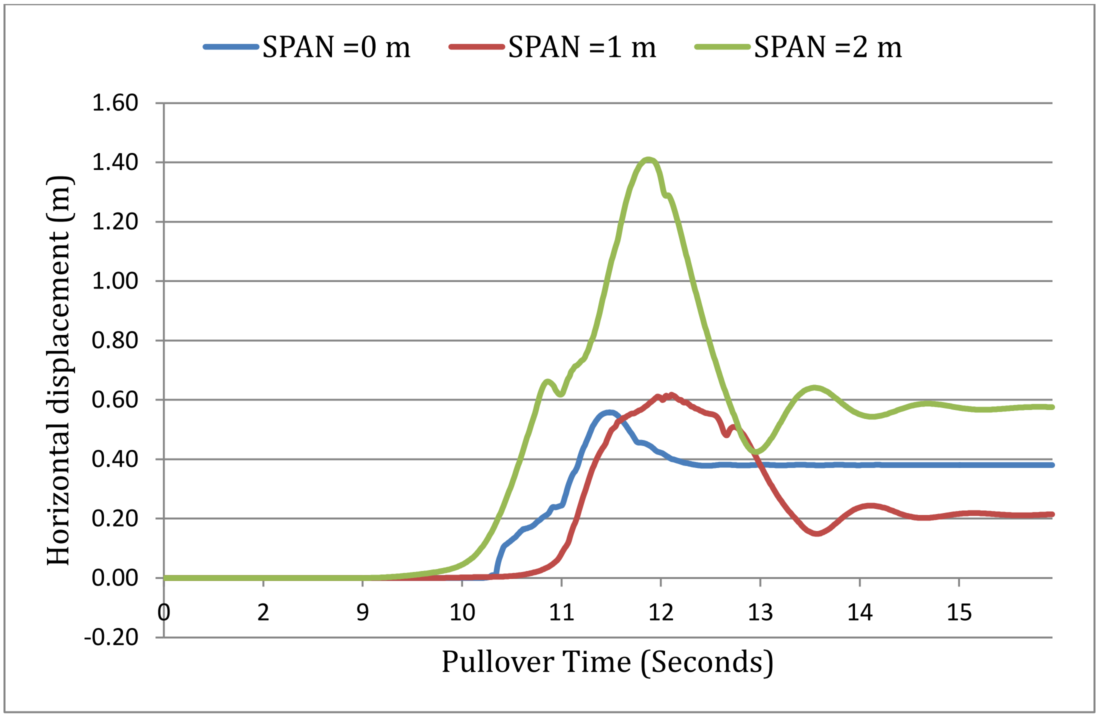
A.1 Horizontal displacements due to trawling for all selected parameters



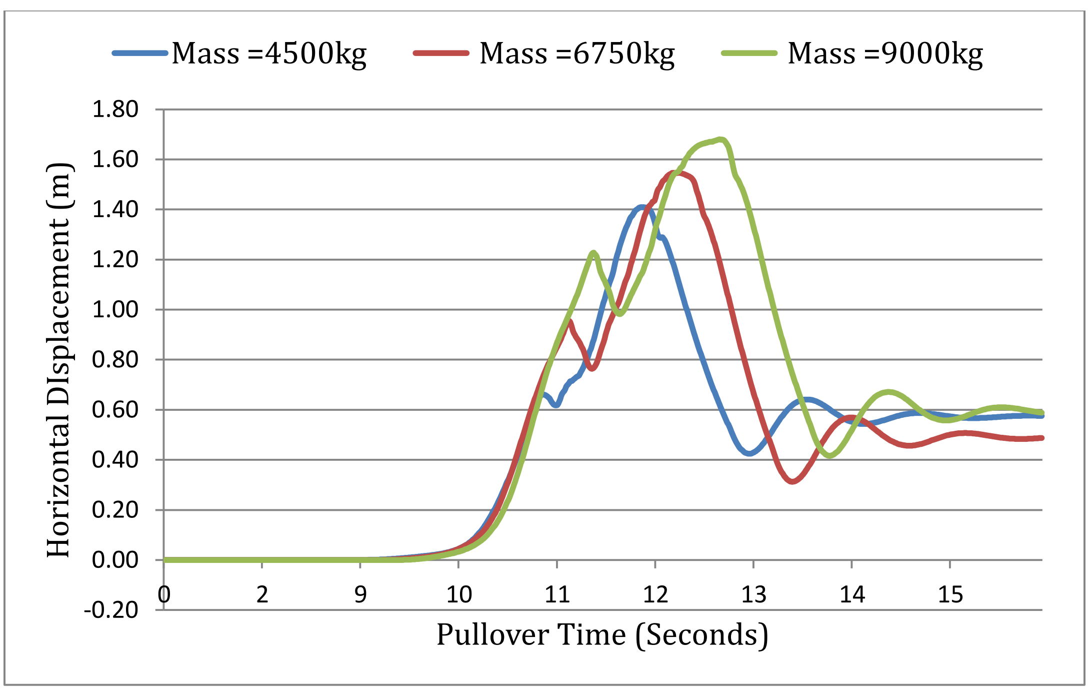
A.1-1: Horizontal Displacement for different trawling velocities (4500kg Clump weight)



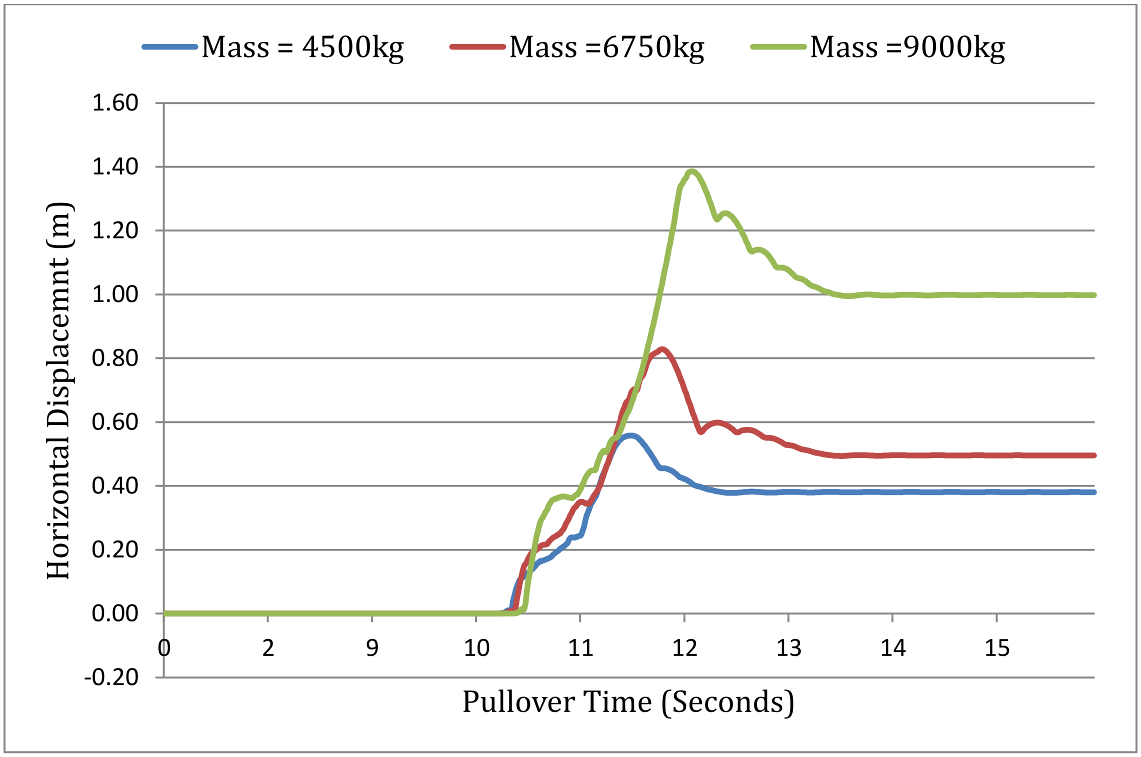
A.1-2: Horizontal Displacement for different warp line lengths (4500kg Clump weight)



A.1-3: Horizontal Displacement for different span heights (4500kg Clump weight)



A.1-4: Horizontal Displacement for different Clump weight masses (Span height 2m)



A.1- 5: Horizontal Displacement for different Clump weight masses (Span height 0 m)

Appendix B: RESULT TABLES

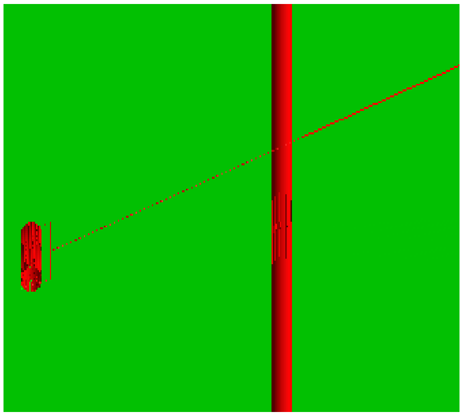
B.1 Summary of maximum pullover forces from clump weight simulations for all selected parameters.

Parameters				Clump weight FEA Results		
Span height (m)	Mass (kg)	Trawling Velocity (m/s)	Warp line (m) (3.5 x water depth)	Maximum Horizontal Pullover Force (kN)	Maximum Vertical Pullover Force (kN)	Remarks
2	4500	1.65 2.3 3	770	92.42 110.62 143.22	-77.79 -118.62 -93.69	All cases identical except for trawl velocity
2	4500	1.65	770 1155	92.42 141.42	-77.79 79.04	All cases identical except for warp line lengths
0 1 2	4500	3	770	111.17 119.4 143.22	-44.85 -115.37 -93.69	All cases identical except for span height
2	4500 6750 9000	3	770	143.22 182.1 238.61	-93.69 -122.14 -162.22	All cases identical except for clump weight mass
0	4500 6750 9000	3	770	111.17 124.45 149.2	-44.85 -71.66 -89.22	All cases identical except for clump weight mass

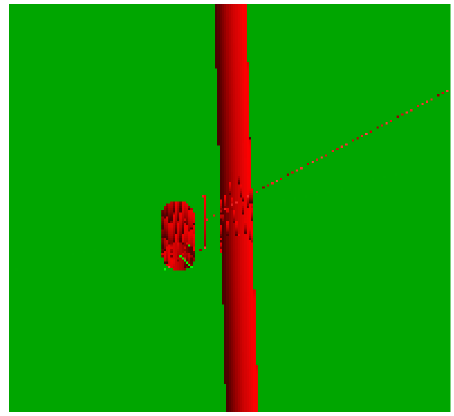
B.1-1: Maximum pullover forces from clump weight simulations for all selected parameters.

Appendix C: PULLOVER SCREEN SHOTS

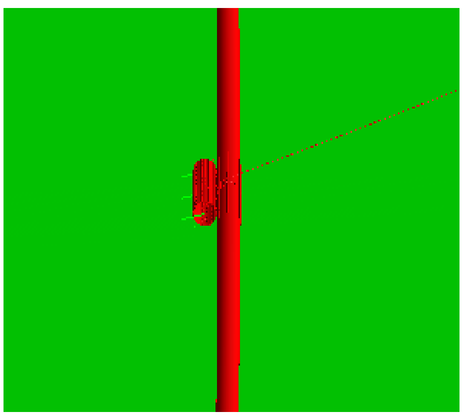
C.1 Pullover for 0 m span height without axial compressive force



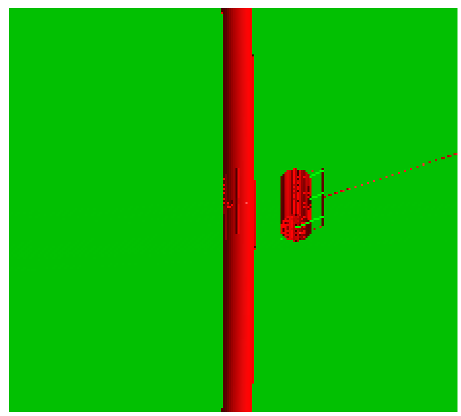
Pullover time = 10 s



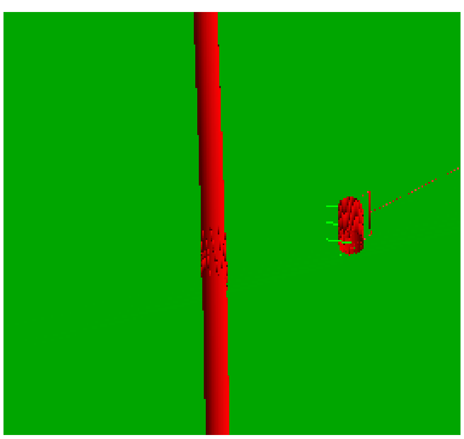
Pullover time = 16 s



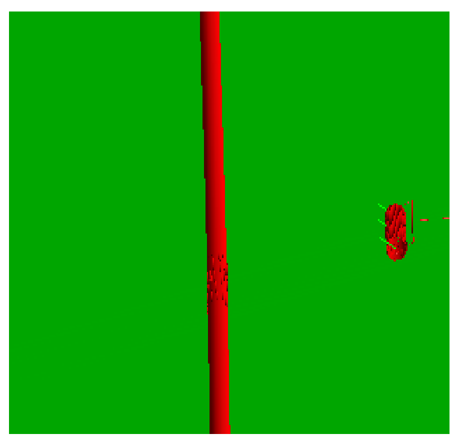
Pullover time = 17 s



Pullover time = 19 s

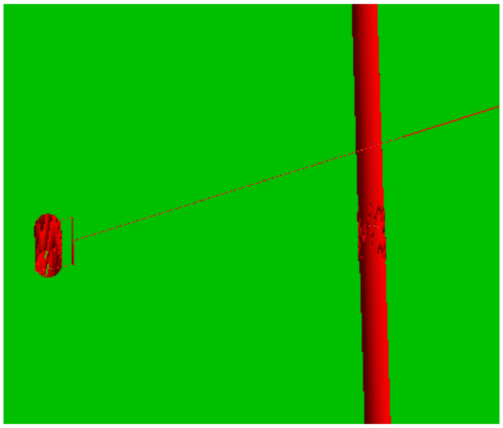


Pullover time = 21 s

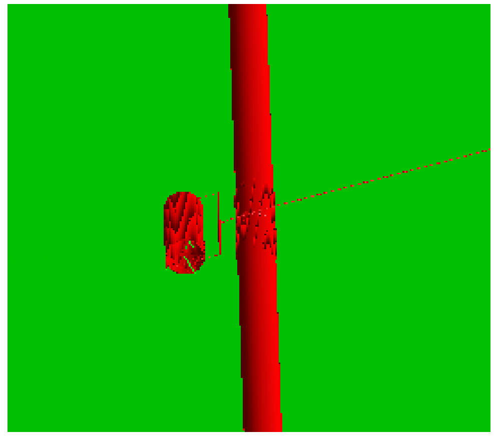


Pullover time = 24 s

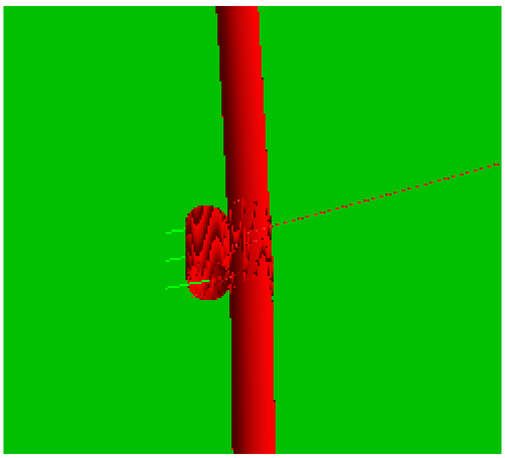
C.2 Pullover for 0 m span height with axial compressive force



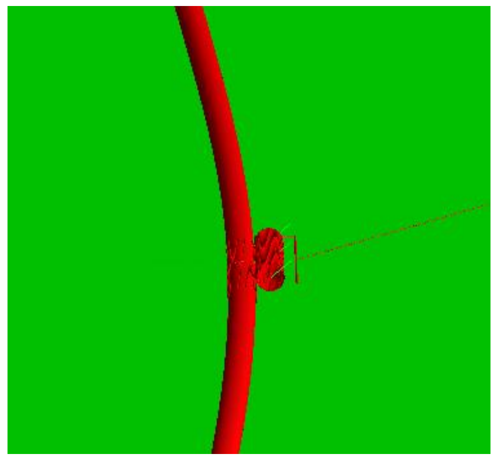
Pullover time = 10 s



Pullover time = 16 s



Pullover time = 17 s



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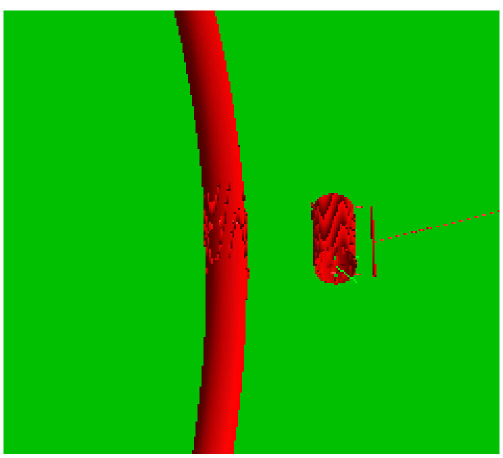

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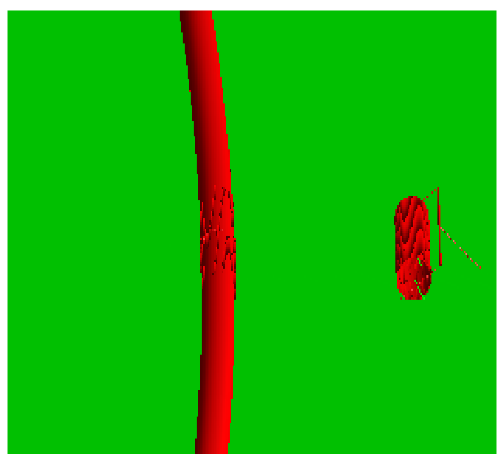
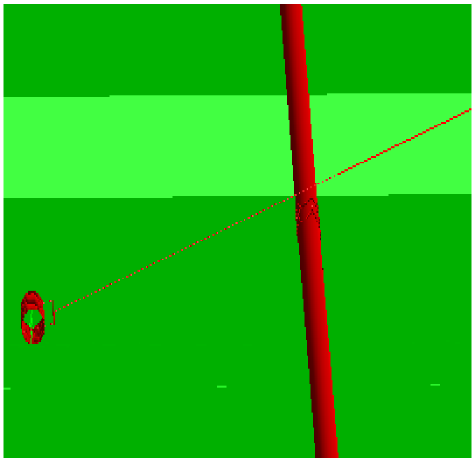
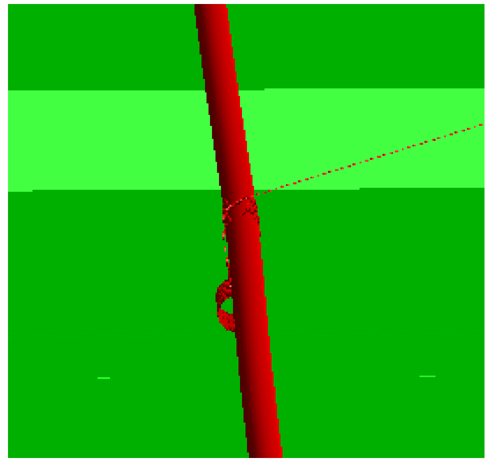


Pullover time = 24 s

C.3 Pullover for 2 m span height without axial compressive force



Pullover time = 10 s



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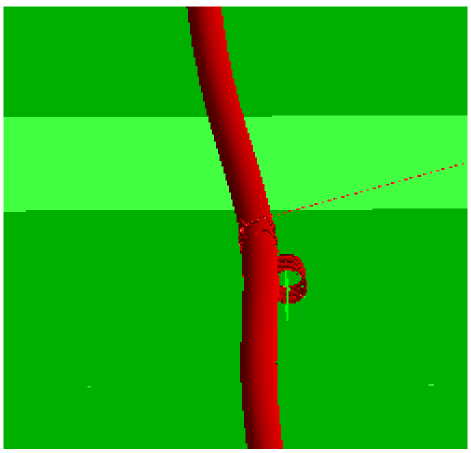

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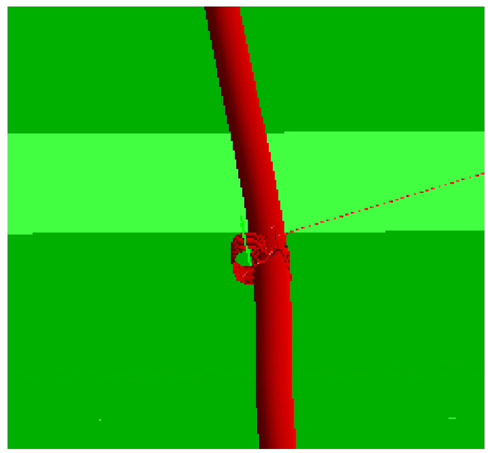

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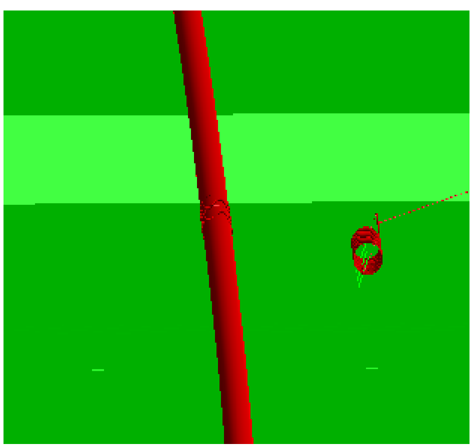

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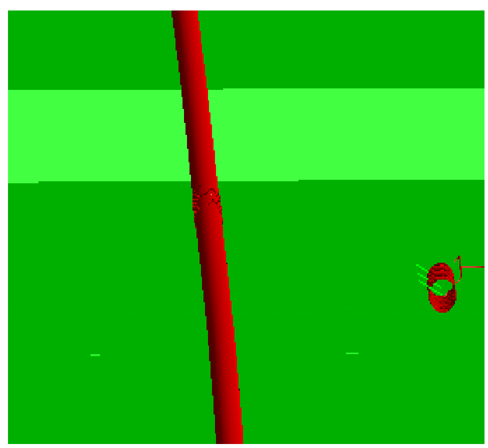
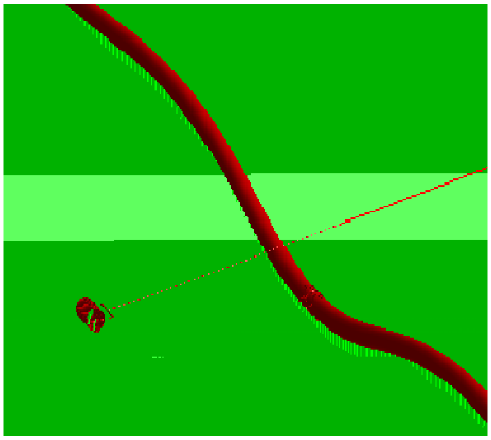


Pullover time = 24 s

C.4 Pullover for 2 m span height with axial compressive force



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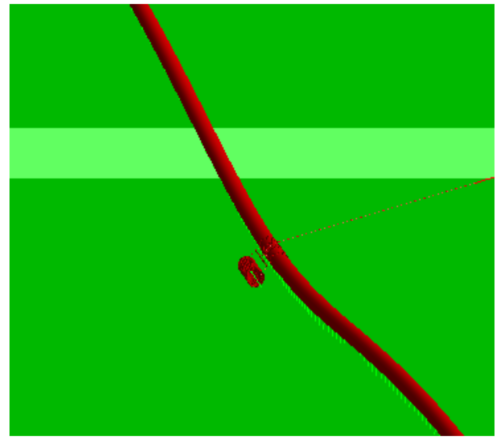

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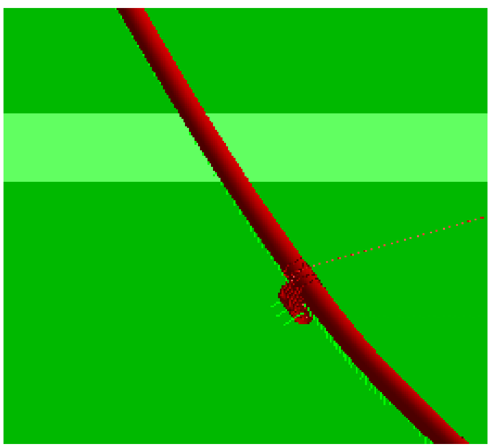

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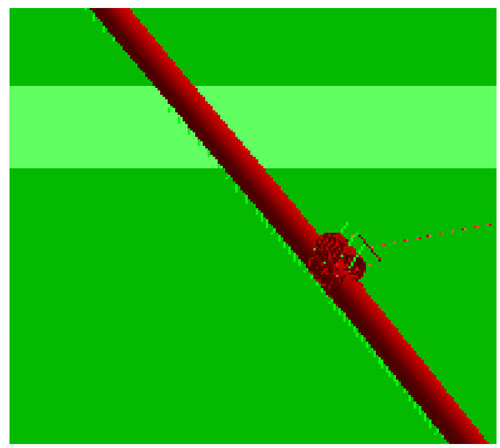

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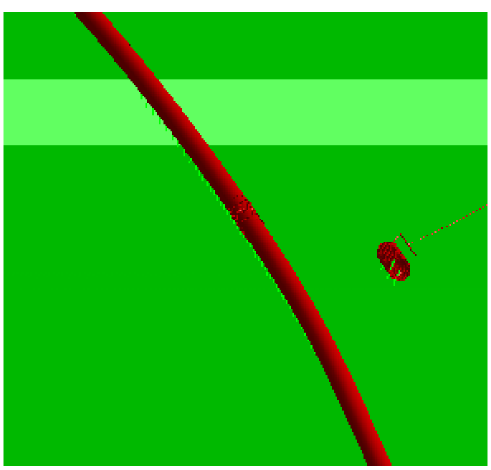

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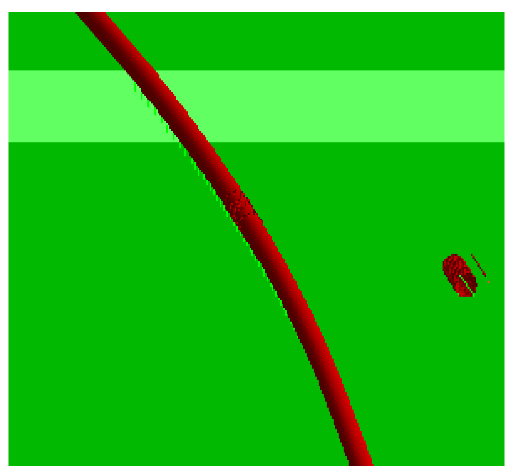

Pullover time = 24 s