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MASTER'S THESIS

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Abstract

Mooring lines that are used on floating offshore units are subjected to bending at the end termination. Over-bending in some cases may cause the mooring line to suffer damage from fatigue. A countermeasure for this over-bending has been to implement bend stiffeners at the end terminations. National Oilwell Varco procures wire ropes and two bend stiffeners from 3rd party suppliers. The scope of this thesis is to study the effect of these bend stiffeners and their impact on the fatigue life of the wire rope mooring lines.

Models of spiral wire rope, bend stiffeners and wire sheathing were created in CAD software Inventor™. Finite element analysis was performed using FAE software ANSYS™. The analysis was conducted using two different loads at three different load angles for both types of existing bend stiffeners.

The results indicate that the implementation of a bend stiffener has a positive effect on the fatigue life of the wire ropes investigated. However, there is an insignificant difference between the two types of bend stiffeners currently in production. A third type of bend stiffener was designed and FEA analysis was conducted in the same fashion as for the existing bend stiffeners. The results show a great increase in the service life of the wire ropes in all the load cases investigated when implementing this new design.

Acknowledgement

The process of selecting the thesis' subject started in the fall of 2013. I consolidated with my father who has been an engineer for over 35 years. He works for APL (Advanced Production and Loading), a branch of NOV (National Oilwell Varco). When I asked him if there are any possible subjects he thought APL could need help with, he came up with a list of several subjects, spread across the departments at APL. After some thought and based on my interests and abilities, I decided to narrow the choices down to two possible subjects; subsea oil separation or mooring line analysis. I would like to thank my father for his help and giving me the opportunity to write my thesis in collaboration with APL and NOV.

After a brief talk with Arnfinn Nergaard, a professor at the offshore technology department at the University of Stavanger, I decided on the topic for the thesis. I would like to thank Arnfinn for his help in this matter.

My father then gave me the contact info to the head of the Mooring and Risers department at APL, Geir Olav Hovde, which I contacted. He said he was willing to guide me through my thesis work and told me in depth about what the main goals for the thesis would be. Geir Olav has helped me a lot during the writing of this thesis and has given me all the guidance and information I required to complete the work. I would like to give a big thank you to Geir Olav for taking the time and helping me throughout this thesis.

Although Arnfinn Nergaard was my initial contact at the university, Eiliv Janssen has been my supervisor and mentor during the thesis work. He has helped me immensely with his knowledge and experiences as both a professional in the oil and gas industry and as a professor at the university. We have had semi-weekly meetings at the university where he has answered all my questions, both good and bad. I would like to give my warmest thanks to Eiliv for guiding me through the process of writing my thesis and for all his valuable comments and suggestions.

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Nomenclature

C	-	Wire constant
CAD	-	Computer assisted design
D	-	Diameter of sheave/drum
d	-	Wire rope outer diameter
FEA	-	Finite element analysis
ID	-	Inner diameter
k	-	Thousand
m	-	Fatigue factor
M	-	Million
mm	-	Millimeter
MPa	-	Mega Pascal
N	-	Cycles to failure
NOV	-	National Oilwell Varco
OD	-	Outer Diameter
R	-	Actual load range
R_m	-	Minimum load
R_q	-	Equivalent load range
UBL	-	Ultimate bearing load

1 Introduction

1.1 Selecting thesis topic

When selecting the subject of the thesis, I wanted to find a topic that was concrete and solvable. Writing vast amounts of theory, equations and proofs is not an ideal way to show the reader what problem solving skills and ingenuity that an engineering student possesses. By having a concrete goal to work towards and to be able to see actual results and solutions to a problem is what engineers strive towards. Finding a solution to a real life problem and being a part of something greater than just a theoretical study was the top criteria when selecting a topic, therefore the selection of the topic of analyzing a mooring wire rope was a perfect choice and being able to write it in collaboration with the established oil and gas companies National Oilwell Varco and Advanced Production and Loading was a bonus.

1.2 Objective of thesis

The purpose of a thesis is to enable the student to develop deeper knowledge, understanding, capabilities and attitudes in the context of the line of study. It also serves the purpose for the writer to expand his or her understanding of the engineering field and to come up with ideas and solutions to problems that others may not have thought of before. Although a master thesis is an individual task, communication between the writer, the company the thesis is being written in collaboration with and the university makes the thesis a part of something greater than just the student's work and the contents of the thesis.

National Oilwell Varco procures wire rope from subcontractors. At the end of these wire ropes there is an end termination that includes a bend stiffener which purpose is to stop the rope from "snapping" in the transition to the stiff part of the end termination. According to NOV, this bend stiffener is a standard product which does not go through any analysis. However, for one of their projects, the customer (undisclosed) requires them to look at the effects of this bend stiffener as well as the effect of fatigue caused by the bending of the wire rope at the end termination. This thesis will look at this in detail by modelling 3D models of the bend stiffener and wire rope and conduct analysis using FEA software. Based on the results in the analysis a further evaluation of fatigue of the wire rope itself shall be conducted.

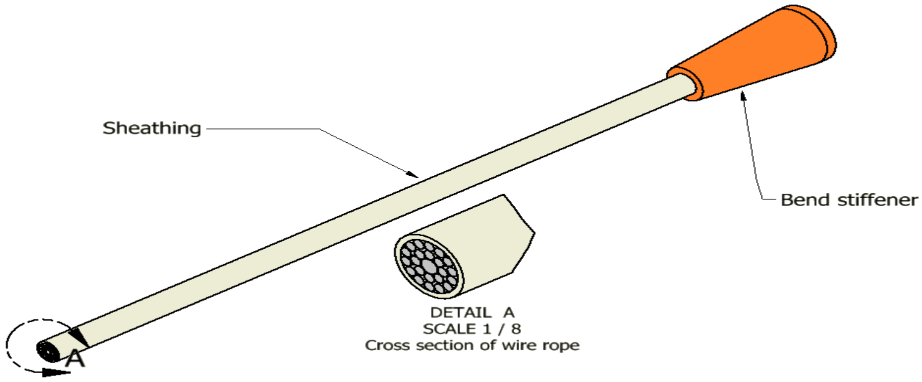

Figure 1: Sketch of 3D model

There are two existing types of bend stiffeners that NOV procures. One objective of this thesis is to look at which one of these is most suitable and what kind of benefit we get on fatigue life of the wire by implementing them in the mooring system.

Though there are already two existing types of bend stiffeners that NOV uses, there is room for improvement on these existing types. Therefore a 3rd option will be created in order see if a different design would improve the life span of the wire rope.

Though the main purpose of the thesis was to conduct this analysis, a major output of the work done on this thesis is the heightened skill in the use of both FAE and 3D modelling software. What has been accomplished in this thesis is not only what is written here in this rapport, but in great deal the experiences that has been gained while getting to know in depth and using the ANSYS and Inventor software.

1.3 Method

In order to achieve the objectives, a 3D model of both bend stiffeners, the wire rope and a protective sheathing will be created using CAD software. These will then be put together into an assembly and imported to a FEA analysis program. By then adding a fixed support to one end of the model and applying certain load cases on the other end of the wire rope we will compare the stresses in the wires and see what effect the bending of the wire will have and what influence adding the bend stiffener has to the stress levels.

When all the simulations have been run and the stress data has been collected, a fatigue calculation will be done to see what the fatigue life will be in each case, with and without bend stiffener, in pure tension, angled tension and in some extreme cases.

In addition to this there will also be an attempt to improve the fatigue life in some situations by designing a third bend stiffener. Comparing both stresses and fatigue life with this bend stiffener to investigate if this design may increase the operation life of the wire thus reducing the replacement cycle of the wire rope and increase safety factor for permanent mooring lines.

1.4 Introduction to computer software used in thesis

Some of the work done in this thesis has been conducted with the use of Ansys and Inventor. The extent of the subject was in such a nature that the aid of software was needed to reach a conclusion.

1.4.1 Ansys

Ansys is a finite element analysis program. It is developed by SAS IP Inc. It is used by many companies and is well known as a reliable and multipurpose FEA program. There are several other FEA softwares available, but the fact that the software was available to use at UiS and that it has been used as a part of some of the courses during the Bachelors and Masters courses made it the obvious choice in this case.

The program consists of several modules. The module used in this thesis is primarily the Workbench module. This module allows the user to import geometry from other rendering programs and create a model based on this geometry. A project in Ansys Workbench is divided into 7 steps.

1. Analysis system.
2. Engineering data
3. Geometry
4. Model
5. Setup
6. Solution
7. Results

The analysis system determines what environment the case is in. There are many different environments to choose from in Workbench (see Figure in Appendix). In this thesis the analysis system that has been used is Static Structural.

Engineering data is where the properties of the materials used in the analysis are set. Properties like tensile strength, poison's ratio, thermal expansion rate, etc. In Workbench, these properties can be changed at any time without having to create a new project.

Geometry allows you to create a 3D model of the geometry that is to be analyzed. Workbench has its own Design Modeler, but due to my experience with Inventor, I choose not to use this feature, and instead use Inventor to do the modelling and then import this model directly to Ansys.

The other steps in the analysis take place in the Workbench “Mechanical” module. Here the model can be meshed and forces, moments, constraints etc. can be added. After the desired setup has been created, before the solver is started, the desired result parameters have to be determined. I.e. if the user wants to know the total deformation of the model this has to be determined by adding a “total deformation” field in the results tab.

1.4.2 Inventor

Autodesk Inventor Professional is renowned 3D modelling software that is commonly used in the industry for design related tasks. It is also used for documentation and product simulation. Though it has its own built-in FAE software, this feature is not as powerful as the ANSYS software.

The main reasons for using Inventor, over other CAD software like Solid Works or Solid Edge, are my previous experiences and knowledge about Inventor and the fact that it is free to use for all students for academic purposes. The user interface and help/tutorial menus are easy to use and an import function between Inventor and ANSYS exists making it easier to convert the 3D assembly created in Inventor to an ANSYS model.

2 Pre-study

2.1 Wire rope in general

Wire rope has traditionally been used in mining operations since the mid-1800s. It replaced rope made by hemp and heavier steel chains. The earliest wires were constructed using a low carbon iron alloy. (In contrast to regular cast iron with high carbon content). Today however, steel is almost solely used. Most wire ropes consist of several metal strands which are laid in a helix pattern. [1]

Because of the way wire ropes are designed it achieves a very high tensile strength. They also have a redundancy when it comes to failures. Unlike if a chain has a complete failure of a link, if a wire strand is broken the tensile forces are taken up by the remaining strands. In addition to high tensile strength and failure redundancy, the geometric properties of wire rope allows it to bend over sheaves and drums with relatively small diameters.

2.2 Rope types and lays

In different usage cases it is necessary for wire ropes to have multiple kinds of properties. To achieve these properties wire ropes can alter two main geometrical attributes, the amount of wire strands in the rope and the lay of each strand.

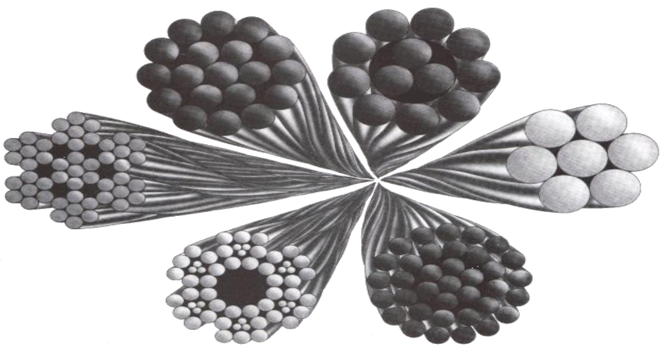

Figure 2: Selection of wire rope types

Figure courtesy of Viking Moorings Inc.

Wires in a rope may have different diameters depending on which strand they are in. Strands can also be divided into several smaller wires (as shown in Figure 2) where the two wire ropes on the bottom left have six outer strands with several wires in each strand.

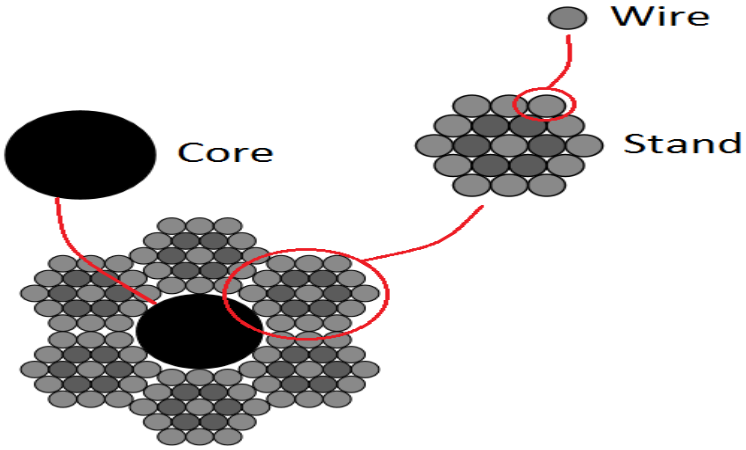

Figure 3: Composition of a wire rope

Essentially there are two categories of wire ropes; spiral wire rope and stranded wire ropes. Spiral wire rope has layers of wires in circles around a single core wire or a grouped wire core. A stranded wire rope will have normally six or eight strands or groups of wires in a circular form around either an empty core or another single strand.

2.3 Bend stiffeners

The general purpose of a bend stiffener is add local stiffness to cables, risers, flexible pipes, wires etc. They are used to control the minimum bending radius and to reduce bending stresses which occur at the topside and/or seabed connection points. They can be used on both rigid and floating structures and in static and dynamic settings. It is important for bend stiffeners to have sufficient fatigue resistance so that they have a service life greater than what they are connected to. Bend stiffeners are usually conically shaped polyurethane moldings with a cylindrical bore that slips over the pipe, cable, wire, etc.

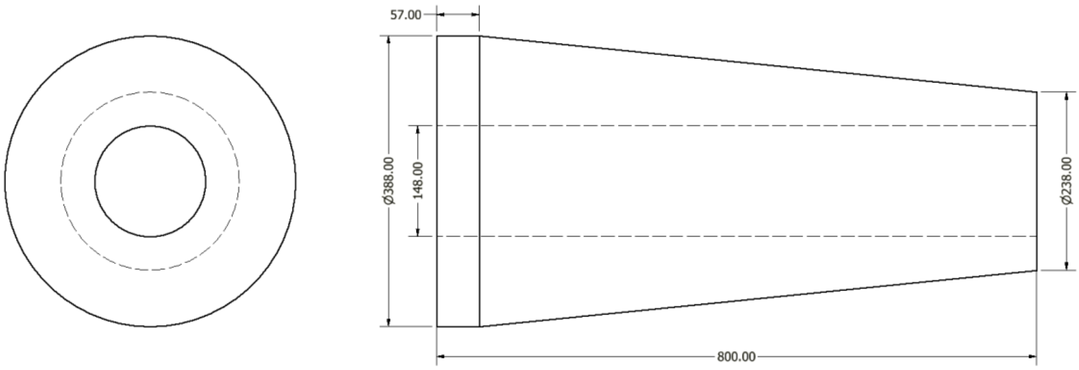

Figure 4: Sketch of 800mm bend stiffener with annotated dimensions

In this thesis we primarily look at a conically shaped bend stiffener with a constant inner diameter. It is made of a polyurethane mold with a Young's modulus of 150MPa.

2.4 General fatigue in wire rope

In essence the process of fatigue in metals involves crack propagation from stress concentrating defects, by mechanisms which involve local plasticity at the tip of cracks under the influence of a fluctuating load. In practice there are often worsening environmental factors and the “defect” may be some geometrical stress point such as a straight corner or some minor flaw in the geometry. In the fatigue of engineering components it is unusual to have other flaws than a single crack, and in laboratory experiments it is common to observe very significant scatter in the fatigue endurance recorded in similar fatigue tests. The parameter which mostly dictates the endurance or number of cycles to failure is the load or stress range.

Wire ropes are constructed of a complex assembly of steel wires. Typically the steel used has a very high strength, which may have five times greater strength than typical structural steel. This high strength is achieved by using plain carbon steel with high carbon content and a very fine grain structure achieved through isothermal transformation (patenting), and work hardening by successive drawing/stretching.

The splitting of the load bearing capacity between many “parallel” wires has two essential benefits. First it assures the essential combination of high axial strength and stiffness with bending flexibility. Second it allows the structural use of brittle steel at very high stresses by dividing the structure to isolate local fractures (if one wire has an imperfection it does not compromise the entire structural integrity of the whole wire rope). This is important in ensuring that a wire rope is “tough” in the sense that it is tolerant of local damage, particularly in the form of broken wires.

Wire ropes operate at high stress levels and are almost always subjected to fluctuating loads. In the ropes that are used in mooring lines one source of stress fluctuation is repeated bending and straightening caused by the movement of the unit.

In a mooring system, tension fluctuations due to motions induced by environmental loading on the floating structure, are the dominant source of fatigue stresses. Given time and a sufficiently high fluctuation in stress range, fatigue is inevitable. However in a wire rope, due to the loose coupling between wires, complete failure of the rope requires that many wires are broken close to each other.

Tut the fatigue of a single wire in the rope is regularly more than a simple matter of fluctuating stress, there is usually some other process which accelerates the fatigue, and which focuses the process to specific locations. This process might be fretting between wires (wear and corrosion between contact surfaces). In practice, rope fatigue involves a large number of fatigue processes going on in series (at different locations along each wire) and in parallel (similar processes along each of many wires). Rope failure occurs when the accumulation of wire breaks at a point is sufficient to initiate total failure.

The primary mechanisms responsible for stress fluctuations in ropes can be grouped under four headings: tension-tension, bending- over-sheaves, free bending and torsion. Each of these will be considered in turn, and then in combination.

2.5 Fatigue modes

In fatigue there are several ways in which the fluctuating load can influence the endurance of the wireline. Depending on the set-up, environment and other factors fatigue may occur on the basis of the following modes:

2.5.1 Tension-tension fatigue

This category of rope fatigue is probably the simplest, involving stress fluctuations from changes in axial tensile loading. In many fixed ropes tension-tension fatigue may be the only mode of fatigue. This fatigue mode is a major consideration for mooring ropes. Tension-tension fatigue is also relevant to lifting or hoisting applications where attached mass changes and accelerations are the primary sources of axial load fluctuation. For this type of fatigue dominant parameters are:

- tensile load range
- mean load (the requirement that load is always tensile limits this effect)
- rope construction and wire grade
- environment (including lack of effective lubricant, or exposure to corrosion)
- manufacturing quality.

Out of these parameters the dominant parameter is the load range and a good model for tension-tension fatigue performance is provided by a simple equation[2]:

$$\mathrm{N} = \left(\frac{\mathrm{C}}{\mathrm{R_q}}\right)^{\mathrm{m}} \qquad (2.1)$$

Where N is the amount of cycles to failure, C is a constant found experimentally according to wire rope diameter and R_q is equivalent load range. The constant m is usually set to between 3 and 5, but can be higher for wire ropes with smaller diameter wires. According to DNV-OS-E301, the factor m should be set to 4.8 for spiral wire ropes. With this fatigue mode, the frequency of the load fluctuation is not a factor, only the number of cycles[3].

The equivalent load range is given by:

$$\mathrm{R_q} = \frac{R \cdot 100}{UBL - R_m} \qquad (2.2)$$

Where R is the actual load range, UBL is the ultimate bearing load and R_m is minimum load.

2.5.2Bending-over-sheaves/drums fatigue

The primary sources of the stress fluctuations in this mechanism are the local changes in wire curvature as the rope adapts to the radius of a sheave or drum. However the restriction of the source of fatigue stresses to changes in wire curvature requires that wires can slide with respect to one another. Any constraint on this freedom, for example by ineffective lubrication or internal corrosion, can impair fatigue endurance. The principle parameters are[4]:

- D/d ratio, the ratio of sheave diameter to rope diameter;
- tensile load
- angle of wrap (arc of contact)
- bending length
- fleet angle
- groove geometry and material
- rope construction and wire grade
- environment
- lubrication
- rope quality

Out of the parameters listed above, for a well-designed and maintained system, the first two are normally the most important. However very short bending lengths or low angles of wrap can lead to a significant increase in life because the rope does not fully adapt to the sheave curvature. Opposing combinations of fleet angles and groove geometry can cause additional degradation which in turn reduces bending fatigue performance, causing wear, or inducing twist.

2.5.3 Free bending fatigue

Free bending fatigue involves fluctuating bending deformation of the rope without contact with another body. The curvatures developed in free bending are rarely as severe as those that occur when ropes are running over sheaves, or on and off drums. The frequencies of the oscillations however is often much higher. In fixed rope applications this type of bending often takes place adjacent to a termination which introduces additional local problems and the life time of the rope may be a concern. This type of fatigue is the most important mode in this thesis and will be, in combination with tension fatigue, the basis for fatigue life assessment later on.

2.5.4 Torsion fatigue

The construction of a typical wire rope, with a large number of wires combined so that they share the tensile load, results in overall properties that combine axial strength and stiffness with bending flexibility. An unintended consequence of such a construction is that the rope also has a low torsional stiffness in comparison to structural components with similar axial properties. A further consequence of the geometry of commonly used categories of rope is that they are torsionally active, generating a torque in response to the tensile load when the ends are constrained against rotation, or twisting about their axis when one end is not constrained.

There exist rope constructions designed to minimize the tendency of ropes to rotate, but often these have other disadvantages which might include being less robust, having a tendency to break up internally, and are usually more expensive. However it is also the case that in many applications a combination of fixed ends and no significant variation of tension along the rope mean that even for torsionally active ropes there is no rotation. But in some applications, especially where components with different torsional characteristics are connected end to end, torsional oscillations can be induced in response to tension fluctuations. An application where such problems arise is in the moorings of floating offshore systems. These oscillations are common with hybrid mooring lines combining polyester fiber rope and torsionally reactive, six-strand wire rope. Under these conditions the wire rope may experience a torsional mode of fatigue for which the dominant parameter seems to be twist amplitude.

2.5.5 Combined modes

In laboratory testing, it is easy to isolate the nature of the fatigue loading to aid in our understanding of the phenomenon. But in many real life applications, it is virtually impossible to determine to what extent each fatigue mode influences the service life time of a certain object. The practical operating conditions of wire ropes are such that not only do different types of fatigue modes operate in combination but also that the fatigue parameters are not consistent. Wear and corrosion, which physically alters the wire rope, also causes a lowering of its fatigue performance.

2.6 Fatigue notch factor

Fatigue notch factor is considered as the ratio of fatigue strength of a sample with no stress concentration to the fatigue strength of an identical sample with a notch or other stress raisers. Fatigue notch factor is usually lower than the theoretical stress concentration factor because of stress relief due to plastic deformation. An alternate term is strength reduction ratio. This is an alternative to the more common fatigue concentration factor which is a factor that decreases the fatigue resistance of a sample by a factor between 1 and 0. In this thesis an assumption of perfect wire ropes has been implemented, meaning that the fatigue notch factor will be set to 1.

2.7 Mooring of turret buoy

The mooring spread for a floating offshore structure, whether a fixed production system or a floating unit, involves ropes deployed in a pattern coming out from the moored structure to anchors on the seabed. At the seabed wire rope normally runs to a drag embedment anchor, possibly connected via ground chain. The main loading is simple tension determined by vessel motions induced by environmental loading from waves, wind and current, limited by the overall station keeping characteristics of the mooring system. There are also however bending movements at the connection point between the structure and the mooring line. These movements fall under the free bending category.

System design is typically limited by strength, and would relate to the most extreme environmental loading event considered for the location in relation to service life requirements, usually in a “damaged” state. But fatigue is also a design issue, especially for locations where extreme events are less demanding. Corrosion and winch related damage can also compromise fatigue endurance in the long term. As oil production is required from ever greater water depths the mooring requirements become more challenging, and not only because of more severe environmental loading. The greater lengths of line involved result in greater weight and a catenary which approaches the vessel at a steeper inclination. This reduces the inherent station keeping effectiveness of the mooring leading to a requirement for higher mean tensions, and thus larger heavier mooring lines.

3 Simulation set-up

3.1 Wire connection

In this thesis we look at a scenario where the wire rope is connected to a turret buoy with patented connection system as shown in Figure 5. The end termination allows for some movement both laterally and height wise due to its two-jointed construction. However the joints have limitations and cannot rotate a full 180 degrees. This restriction makes so that the mooring line will have a slight angle of bending with respect to the tension in the line. This angle is roughly estimated to be around 2 to 5 degrees.

Based on this, the tension force that is applied to the mooring line will be at a 2.5 and 5 degree angle relative to the fastening of the wire to the end termination. The wire rope itself is threaded through the steel part of the end termination in a brush-shape and then cast with an epoxy to make it stick in place. On Figure 5 the greyed out part indicates the plastic sheathing with the wire rope inside. The wire rope leads in to the bend stiffener which is the cone-shaped part with the bolt grommets.

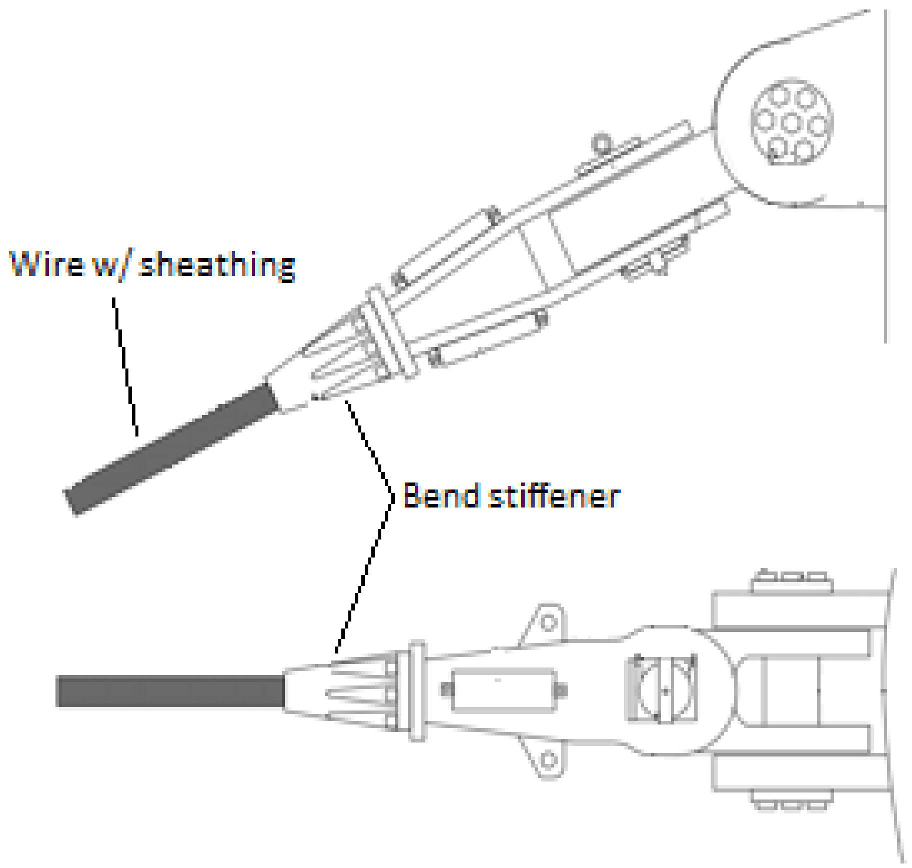

Figure 5: End termination for mooring line

Figure courtesy of APL, NOV

During the installation of the mooring line the angle in which the tension force acts in may exceed the operational angle, though only for a short period of time. An angle of 10 degrees has therefore been included in the analyses of the wire rope in order to cover the installation phase of the mooring line and the loads that occur during installation.

In the analysis it is assumed that everything from the bend stiffener and up towards the buoy is considered infinitely stiff, meaning that the model that has been analyzed has been simplified so that the infinitely stiff part is considered as a “wall” with zero degrees of freedom. Since the purpose of this thesis is not to look at stresses in the end termination itself but the bend stiffener and the wire rope with the plastic sheathing, the rest of the end termination is not a part of the analyzed model.

A 3D model of the end termination was created in Inventor, but due to complexity, increased analysis simulation time and time constraints, the end model that was used in Ansys did not include the end termination parts. A figure of the 3D model may be found in the appendix.

3.1.1 Wire geometry, dimensions and material

National Oilwell Varco, as stated in the introduction, procures wire ropes from subcontractors. These subcontractors include: Parker Scanrope AS, Bridon and ArcelorMittal, and deliver wire ropes for different environments and situations. In this thesis we have used a low torque sheathed galvanized spiral strand wire rope from ArcelorMittal. This wire rope has a rated lifetime of 25 years according to ArcelorMittal. Its construction consists of 13 layers of wire strands where the inner layer is a core wire with a diameter of 6.5mm. The other 12 layers consist of wires with 4.85mm nominal diameter. The layers are circular and consist of one wire in width. The total number of wires in the rope is 465.

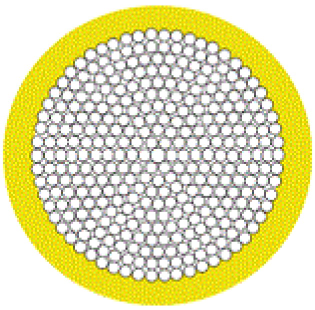

Figure 6: Cross section of the wire rope used

Figure courtesy of ArcelorMittal

Each layer has a lay angle and a lay direction. This lay angle is added to make the rope torsionally stable and reduce the torque and rotation of the wire rope (as mentioned in chapter 2.2). The lay direction is primarily counter-clockwise, but for layers number 5, 8, 11 and 13 (the outer layer) the direction is clockwise. This configuration of lay directions has proven to be the most torsionally stable for this wire type.

The material used in the wire rope is hot dipped galvanized steel with a corzal coating. It has a nominal steel grade of 1880MPa with an average tensile strength of 1940MPa. The Poisson ratio is assumed to be the same as regular steel, 0.3. Because thermal considerations will not be included in this thesis, the thermal expansion coefficient of the steel (and any other materials in the analysis) has been neglected. For a complete set of technical data about the wire se chapter 9.3 in Appendix.

Sheathing around the wire rope acts as a barrier and protects the wires from the external environment, thus shielding it from direct impact from external sources as well as the corrosive environment of sea water. The sheathing also provides additional stiffness and it is made of a material with an E-modulus of 700MPa. It does not take up any direct tension force but some stresses occur due to bending.

According to DNV-OS-E304, one or more of the following measures must be taken to protect steel wire ropes used in mooring applications against corrosion[5]:

- Sacrificial coating of wires.
- Application of a blocking compound on each layer of the strand during stranding. The compound should fill all crevices in the wire rope, strongly adhere to wire surfaces and have good lubricating properties.
- Surface sheathing of the wire rope by an extruded plastic jacket in order to prevent ingress of sea water and flushing out of blocking compound.

This wire has two (coating and sheathing) out of the three required properties and is thus suitable for use according to DNV-standards. In long term mooring, DNV also recommends the use of spiral strand ropes such as the one chosen in this thesis. This type of rope maximizes the steel area and provides high strength to size ratio.

3.1.2Bend stiffener geometry, dimensions and material

Geometrically bend stiffeners are shaped as a cone with a constant inner diameter. The ID of the bend stiffener is large enough to house both the wire rope and its sheathing. In this thesis the ID of the bend stiffeners is set to 148mm. There are two types of bend stiffeners that NOV APL procures as a part of their mooring system, both of these will be analyzed in this thesis. Material properties are the same for the two versions, there are only dimensional differences. One bend stiffener will have a length of 800mm and the other will have 1200mm. Both will have the same OD and ID.

The bend stiffeners are made of a cast polyethylene material with an E-modulus of 150MPa. They are designed to be flexible but still provide extra stiffness at the connection point to the end termination.

3.2 Model creation

As mentioned in the introduction, the models for the analysis have all been made using the CAD software Inventor. The initial thought was to create a model of the wire rope and end termination which was as close to the real components as possible. This turned out to be more difficult than expected.

The end termination was created, but in order to avoid an overly complex model it was assumed that the bend stiffener and wire rope could be fixed to an infinitely stiff "wall". This assumption should only have a minor, if any, influence the results, because of the previously mentioned load angle can be added in the analysis without movement in the end termination. The assumption of an infinitely stiff wall comes from the fact that the end termination parts are over dimensioned and any failure that may occur will not be localized within the geometry of the end termination.

In Figure 5 we see that the bend stiffeners have six bolt grommets where the bolts that attach the bend stiffener to the end termination are slotted. These grommets may be stress concentrating areas since they are so called "imperfections" in the rounded geometry. However since the stresses that occur in the bend stiffeners themselves are very low compared to their strength these grommets have not been added in the model. The main reason for this is to reduce complexity of the model and thereby reducing solving time for each simulation.

The sheathing was just modeled as a simple pipe with a length of 5 meters. The length of the wire rope and sheathing was not very important since the highest stresses in the wire rope would be located close to the fixed end of the model. Making a very long wire would be unnecessary and further increase simulation time.

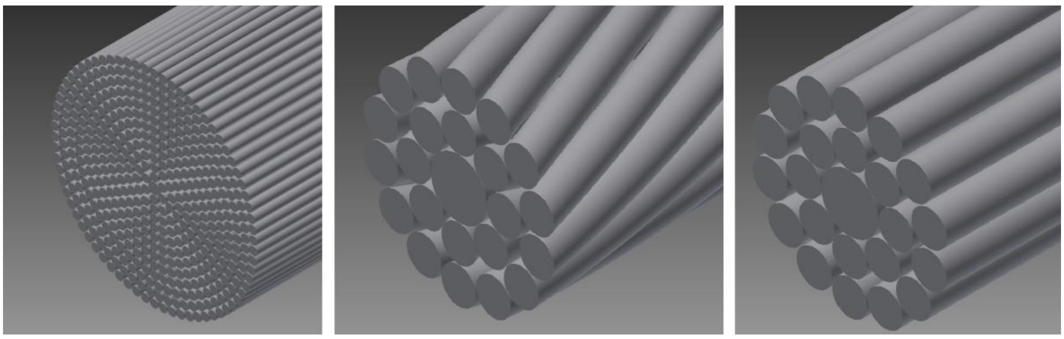

Figure 7: Cross sections of 3 different wire models. From the right: 465 parallel wires, 23 coiled wires, 23 parallel wires

During the creation of the wire rope model several problems occurred. As explained, the main thought was to create a real-as-possible model so I attempted to model a 5m long piece of the wire rope based on the geometry and dimensions mentioned in Chapter 3.1.1 and listed in Appendix 9.3. I started by creating the middle core then coiling each layer around the core. This worked for the initial 3-4 layers but as the number of accumulated wires increased the Inventor program started to become unstable and during the third last layer the program crashed outright. Based on this the model had to be simplified since the program could not handle the complexity of the wire rope.

The first simplification that was implemented was to reduce the total number of layers in the wire rope by increasing the diameter of each wire but keeping the nominal OD of the wire rope. Initially it was reduced down from 13 to 8, but even with this reduction both Inventor and Ansys experienced instability when working with the model. A simulation was initiated but failed after several hours. This is most likely due to the extreme amount of elements in the model. The many contact points between the wires may also cause the simulation to fail after only a few minutes because of convergence difficulties. The model was therefore further simplified and reduced to only three layers: a core, an inner and an outer layer.

Another simplification that then had to be implemented was removing the lay angle (helix/coiling) of the wire by laying the wires parallel to each other and to see if the programs were able to cope with the model. The effects of removing the lay angle was a bit uncertain so extra comparison simulations were run to see if the effects were significant. Removing the lay angle proved to be a key in achieving stability in the programs and also dramatically reduced the time it took to complete one simulation. Since this was the deciding factor, a comparison simulation with increased number of layers was run to see what effect the reduced wires would have on the results.

3.3 Optimizing mesh

Meshing is used by all FEA-analysis programs to divide the model into elements. Generally a finer mesh will increase accuracy of the results but will dramatically increase simulation time, while a coarser mesh will decrease accuracy but also decrease simulation time. This is because if the number of elements is high, there will be a high number of equilibrium equations the program need to solve. Getting a good balance between mesh quality and result accuracy is essential for reducing simulation time in complex analysis such as the ones carried out in this thesis.

There is a limit on how accurate the results become and how fine the mesh is. For example if a model has 10k elements doubling this number might yield a much more accurate result while if a model has 1M elements to start with and this amount is doubled to 2M there might not be any difference in the results but it will take almost twice the time to yield almost the same results. It is common to increase the number of elements in areas of interest in the model. For examples in welds, cracks, straight corners, bolt threads and so on. Other areas of the model can consist of a coarser mesh to reduce number of elements.

3.3.1Meshing types and outcomes

In Ansys there are several meshing methods that can be used. To achieve the desired meshing of the model a combination of different methods were chosen. Since the model basically consists of three parts, the bend stiffener, the sheathing and the wire rope, different meshing methods would be necessary. If the meshing method was set to "automatic" the mesh would be created with an uneven and overly fine mesh which in this case would cause the simulation time to increase drastically.

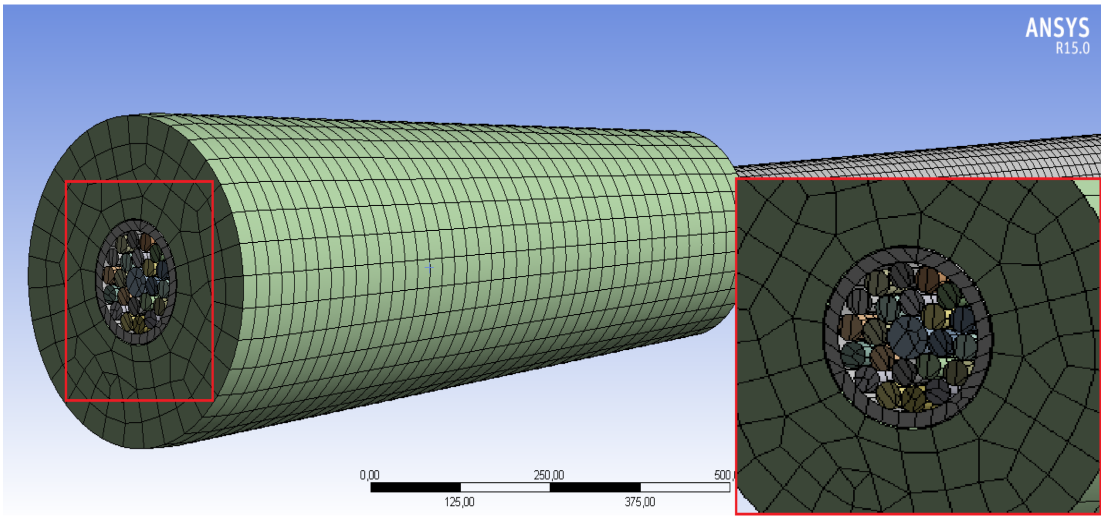

Figure 8: Example of poor auto-generated mesh

In Figure 8 we see an outcome of automatic mesh generation. The outer mesh looks decent and uniform but for the individual wires for example, there is no pattern or evenness in the meshing. This would make the results of a simulation very inaccurate. The total amount of elements with this meshing is an excess of 500k. It would not be advisable to run a simulation with this kind of meshing as it would be both time consuming and inaccurate.

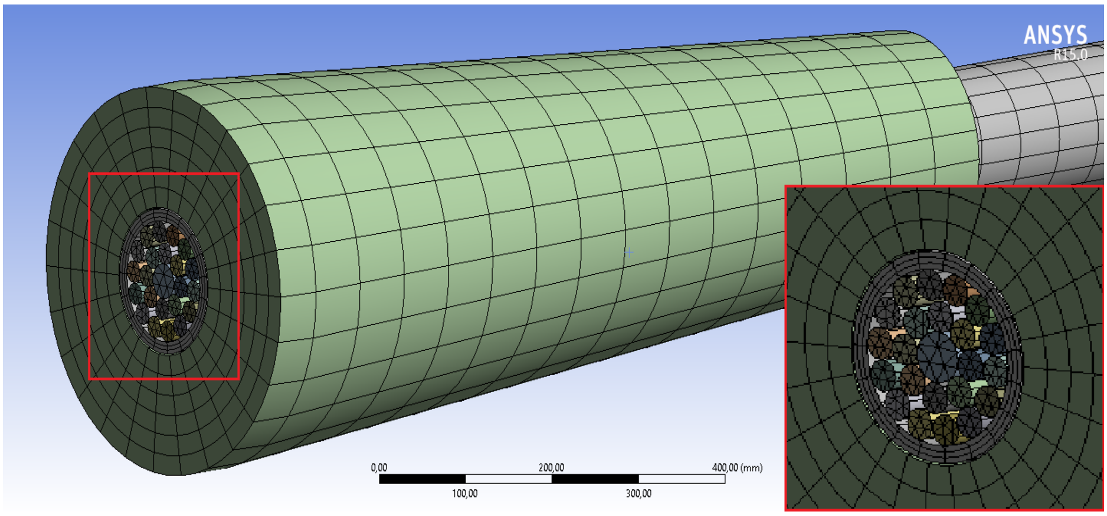

Figure 9: Customized user defined mesh

Figure 9 shows a custom mesh which where I have defined all aspects of the mesh. Although the mesh is slightly coarser than for the auto generated mesh, it is much more even and uniform. Doing this will increase accuracy of the results and since this mesh only consists of 130k elements it is also less time consuming than the auto generated mesh. Making the mesh finer than this could increase accuracy, but to maintain a good accuracy to simulation time ratio this mesh was considered accurate enough.

3.4 Model contact points

Physical contact between the different parts of the model has to be defined before the simulation can be started. The areas which have contact are:

- Bend stiffener and sheathing
- Sheathing and outer wires
- Wires and other wires in the rope

All these contacts have are defined as frictional. The friction factor has been set to 0.1 for all contact. Though the normal friction factor between steel and steel is between 0.3 and 0.4, there is a lot of grease in the wire rope which acts as a lubricant. This reduces the friction and is therefore adjusted down to 0.1. For the relative smooth surfaces of the bend stiffener and sheathing the friction factor is also considered to be 0.1.

Because of the removal of the lay angle, the wires will split up when put under an angled load. Therefore it was necessary to add an end block to the model. The end block was modeled with an infinitely stiff material and with a diameter the same as the OD of the sheathing. The wires and sheathing where then bonded to the end block. The tension force that the wire experiences was then applied to the end block so the wires would stay in place and not split up when the load was applied.

4 Procedures

4.1 Load cases

For a mooring line the primary loads that are experienced comes from movement of the unit that is being moored, in this case a buoy. The movement is normally caused by waves, current and/or wind. In order to determine the forces that act upon the mooring lines it is essential to know what sea state the unit will operate in and what the dynamic response the unit has. By using the response amplitude operator, shortened RAO, of the unit and the known wave heights and frequencies it is easy to run simulations in programs like Orcaflex and Moses to calculate the forces in each of the mooring lines. Doing this was suggested as a part of the thesis, but since this simulation had already been carried out by NOV at an earlier time it was not necessary to duplicate it.

For both the 800mm and the 1200mm bend stiffener a series of different load cases where to be considered from the results of the simulation that was carried out by NOV. The load cases are as follows:

Table 1: Numeration of load cases

Case nr	Tension force[kN]	Load angle [^{o}]
1	2000kN	2.5
2	6000kN	2.5
3	2000kN	5
4	6000kN	5
5	2000kN	10
6	6000kN	10

The first case is what classifies as typical operation situation when the buoy is in normal operating conditions. This will be the most important of the cases since it has the most common occurrences of the load cases and will therefore have the most estimated cycles per year. As stated in chapter 2.5.1 the amount of cycles, N, determines the lifetime of the wire rope, the more cycles the wire experiences in a year the fewer years the wire rope can be in operation. The second case is in a more harsh weather condition where there are higher waves and stronger winds, thus increasing the movement of the buy and increasing the load on the wire rope.

Load cases which have a load angle of 5^o are considered to be in an installation phase of the mooring line. When the wire rope is installed it is lifted from the sea bed with a crane and onto the buoy. When it is being lifted it experiences a certain tension. The total axial force may not be as high as when the wire rope is in operation but it will have a higher load angle.

The load cases with a 10^o load angle are in extreme cases. For example if one mooring line has broken off, the end termination mechanism had locked up or in extreme weather cases.

This would be a very rare occurrence and thus having a small amount of cycles and fatigue may not be the limiting factor in these cases. Breaking of wires because of stress levels above the strength of the wire material are more probable failure causes.

As a baseline test, a load case with pure tension (zero load angle) of 2000kN was also run to compare between the different wire models that was used. The baseline also serves as a comparison between pure tension loading and load at an angle that induces bending.

In Ansys the load is applied over a time with certain time increments specified by the user. In most cases the load was added in two time steps, first a load step which added 1% of the load, then a second load step which added the remaining 99% of the load. So for a 2000kN load case 20000N was added first then the rest for a total of 2000kN. This was done in order to not overload the wire in the start of the simulation and to make sure that Ansys did not have convergence difficulties.

4.2 Load cycles

The fatigue life is, as mentioned in chapter 2.5.1, based on the number of load cycles that the wire rope experiences. In the mooring line these cycles are governed by the movement of the turret they are attached to. We assume that for wave heights between 1.5m and 7m are considered normal operation conditions and all waves above 7m are considered as storm or extreme weather cases. If we look at the total amount of waves for a specific area of the North Sea we can conclude that a frequency of roughly 8500 waves/year in operational conditions and about 500 waves/year in stormy or extreme conditions[6].

For the fatigue calculation we will assume that the load is fully reversed, meaning that the minimum load, R_m, will revert back to zero due to the movement of the buoy in the trough of the wave. The maximum load will be at the crest of the wave where the tension of the wire rope will be at its highest.

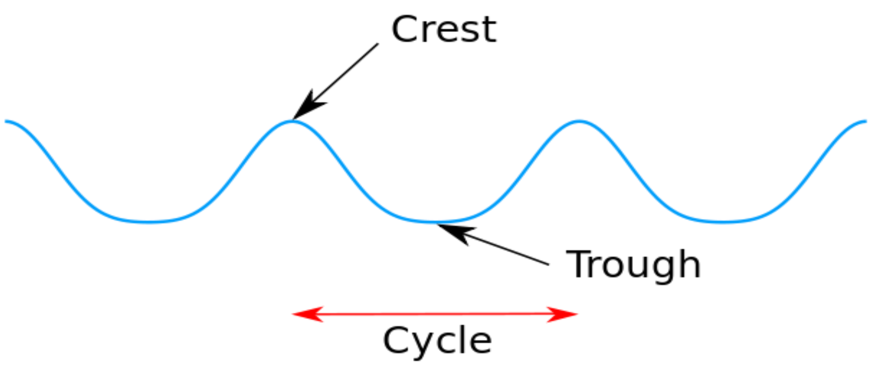

Figure 10: Illustration of wave crest and trough

When we calculate the fatigue life time we also assume that free bending fatigue will be the dominating fatigue mode, thus not taking the other fatigue modes into account. Studies from the University of Reading, UK, show that tension-tension fatigue and free bending fatigue could be compared and use the same fatigue life calculation formulas.[7]

4.3 Simulation computer system

FEA simulations usually require a lot of computing power to run through simulations. The main computer system that was used to run the simulations is custom built by me personally and has the following specs:

- CPU: Intel Core i7 3770K
- RAM: 16GB DDR3
- Storage: 120GB SSD + 1000GB HDD Raid 0

The processor has been overclocked to 4.5GHz in order to maximize its computing power. It has also been water-cooled to keep the operating temperatures at a safe level. Increasing the frequency and the voltage a CPU runs at will also increase the heat-output of the CPU, so doing this without adequate cooling would not be advised. The RAM has also been slightly overclocked to 2133MHz for increased performance of the system. A quick comparison test between the standard and the overclocked system shows an increase in computing time by about 30%. Having a lot of hard drive space was important as the Ansys project files got larger. The end project file was about 200GB in size.

5 Results and discussion

5.1 Pure tension results

In order to establish a baseline for stresses is different wire models and to see the effects that bending has on the stresses, a pure tension load was applied to all the models. Three different models where studied in pure tension; 23 coiled wired rope, 23 parallel wired rope and 64 parallel wired rope. This was done with a total tension force of 2000kN. For pure tension the plastic sheathing and the bend stiffeners are not taken into account as they would not influence the results in any meaningful way.

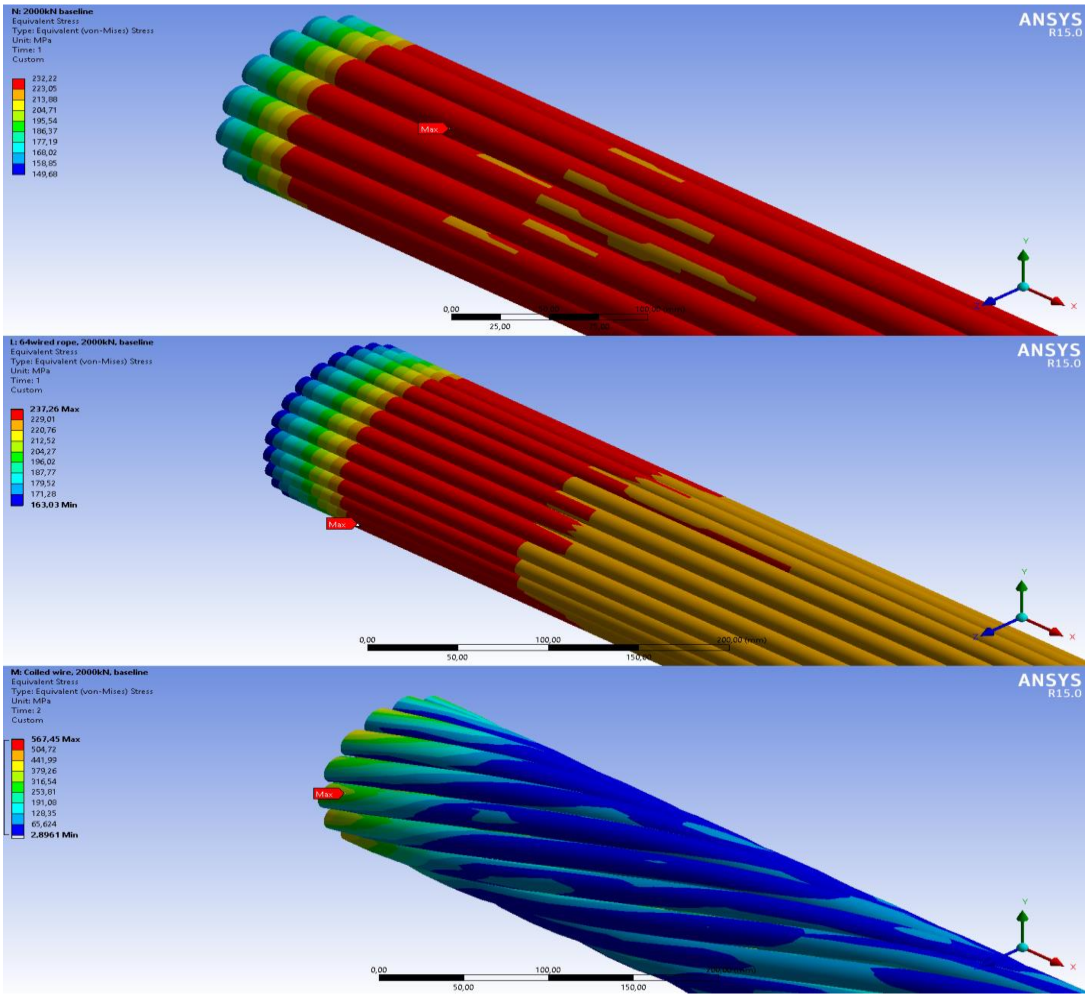

Figure 11: Comparison of the three wire rope models under pure tension

The results show only minor differences between the stresses in the 23 and 64 wired ropes. This can indicate that reducing the amount of wires in order to reduce complexity and computing time does not have a great impact on the results. We do however see that there is a very large difference when we introduce the helix of the wires. Maximum stress is more than doubled when the lay angle of the wires is applied. We also see that the stress is more concentrated at the fixed point where as for the ropes with parallel wires there is not such a distinguished stress concentration at the fixed point, but a more evenly spread stress level.

The main reason why this effect occurs is difficult to isolate but the most likely reason is that since the model is fixed at this point and the wires are laid at an angle the tension force that is taken up here will be at an angle therefore the stress level in the wire will increase since it is not in pure tension but is in fact experiencing some bending here. This can mean that the wire rope model with the coiled wires may be slightly torsionally active even though the pitch of the helix is the same as for the real wire rope.

Table 2: Maximum stresses in the inner and outer wire strands and the wire core and maximum total deformation of the entire model

Model and load case	Max Stress [Mpa]			Max Total Deformation [mm]
	Outer	Inner	Core	
Parallel wires, 23 wires, 2000kN, pure tension	232.65	232.6	232.45	5.59
Parallel wires, 64 wires, 2000kN, pure tension	237.26	237.11	237.06	5.72
Coiled wires, 23 wires, 2000kN, pure tension	479.35	567.45	545.86	13.836

We also notice that for the coiled wire model the stresses in the inner wire strand and the wire core are quite large compared to the outer core. This is as expected with the torsionally active rope. Friction from the rotation of the wires in the outer strand causes additional stresses in the inner wires. Why we see an increased deformation in the read out is a bit unclear. It may be because when the wires are twisted in a helix they act as a sort of spring which reduces the axial stiffness of the wire rope compared to the parallel models.

5.2 Operational condition

In operational condition the load angle is, as mentioned in chapter 4.1, set to 2.5 degrees. Here we introduce the sheathing and the bend stiffener to the model. Initially a simulation was run with the bend stiffener supressed, meaning the model only consisted of the wire rope and the plastic sheathing. This was done to have a comparison baseline to see the effect the bend stiffener will have on the stress levels.

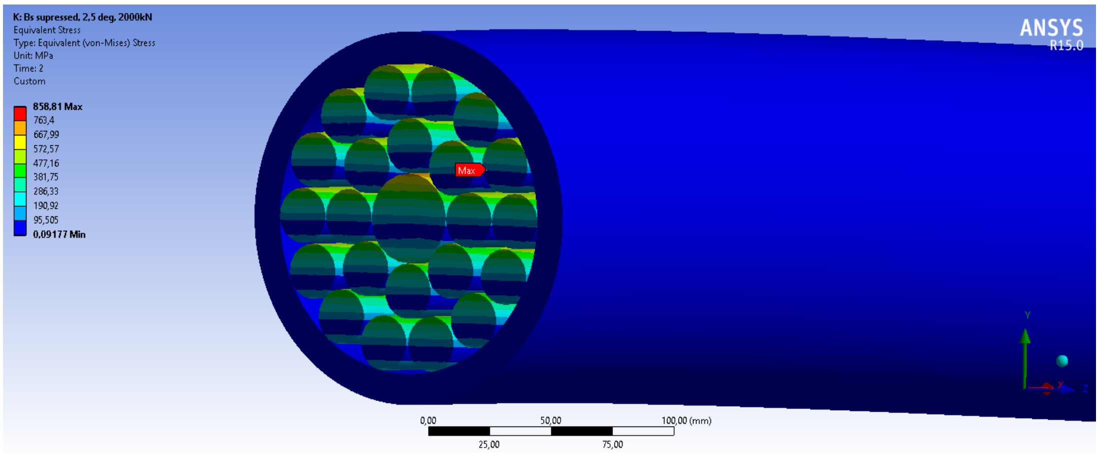

Figure 12: Cross section of wire at fixed end without bend stiffener with a 2000kN load at 2.5 degrees

The same simulation was then run again with the bend stiffener unsuppressed. First the 800mm bend stiffener then with the 1200mm version. Both where run with 2000kN and 6000kN load at 2.5 degrees. With this load case, due to extensive simulation time, only the 23wired rope was tested. Looking at the results from the pure tension comparison there is not much differentiation between the 23 and 64 wired ropes so the results are expected to be within margin of error, concluding with that the model differences are neglectable.

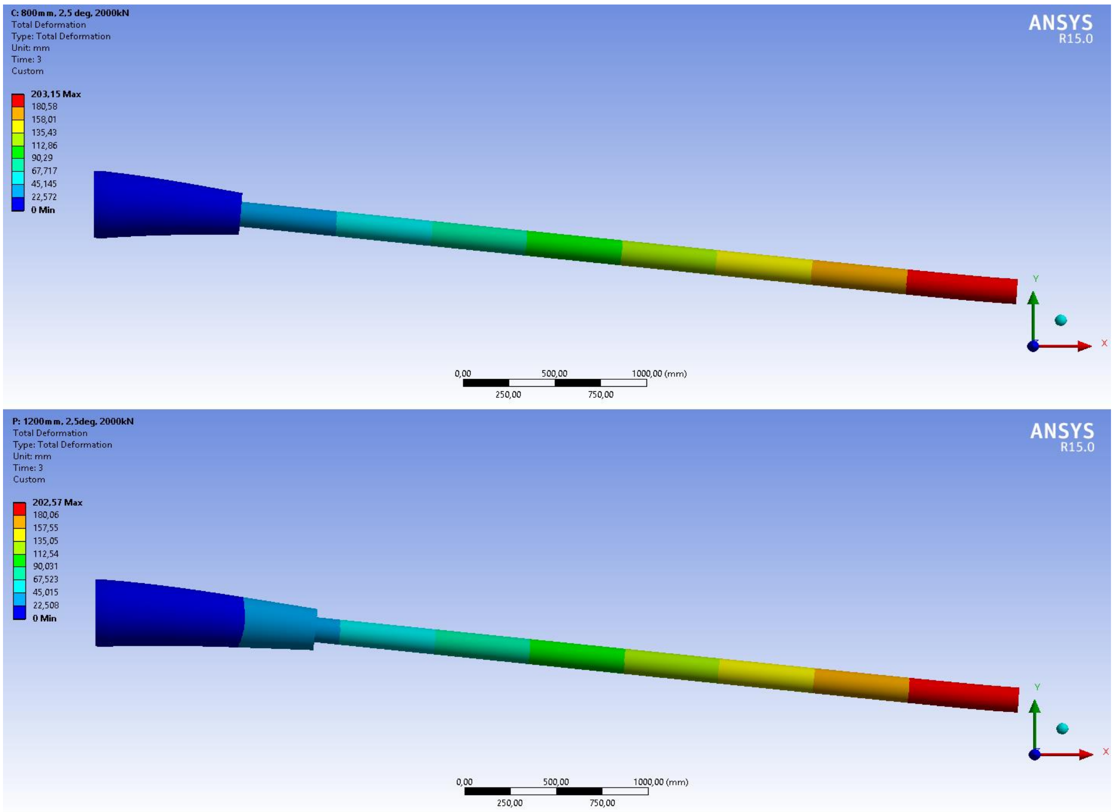

Figure 13: Bending of wire with sheathing and bend stiffener (800mm top, 1200mm bottom) with a 2000kN load at 2.5 degrees (deformation is shown at 2.5:1 scale)

Table 3: Maximum stresses in the inner and outer wire strands and the wire core and maximum total deformation of the entire model

Model and load case	Max Stress [Mpa]			Max Total Deformation [mm]	
	Outer	Inner	Core		
Parallel wires, 23 wires, 2,5deg, 2000kN, BS suppressed	673.57	667.14	858.81	206.82	OK
Parallel wires, 23 wires, 2,5deg, 2000kN, 800mm BS	590.64	566.39	722.47	202.64	OK
Parallel wires, 23 wires, 2,5deg, 2000kN, 1200mm BS	630.65	615.11	708.79	202.06	OK
Coiled wires, 23 wires, 2,5deg, 2000kN, 800mm BS	758.2	908.39	1019	213.23	Anomaly

For the model with the bend stiffener compressed there is a clear increase in stress compared to the models that include bend stiffener. This shows that implementing a bend stiffener will have a reducing effect on the stress levels in the wire rope. However though both bend stiffeners have a reducing effect on the stress, the increased size of the large type does not seem to have any significant advantage on the influence on the stress, in fact it may have an opposite effect for the inner and outer strands.

Though the maximum stress in the wire core is slightly reduced (can be within margin of error as it is only a 2% reduction) there is a more than 10% increase in the stresses in the inner wire strand and about a 7% increase in the outer wire. These increases are so large that they have an extensive impact on the fatigue life of the wire rope which will be shown in chapter 5.5.

Table 4: Maximum stresses in the inner and outer wire strands and the wire core in each of the models and total deformation of the entire model

Model and load case	Max Stress [Mpa]			Max Total Deformation [mm]
	Outer	Inner	Core	
Parallel wires, 23 wires, 2,5deg, 6000kN, 800mm BS	1230.00	1221.70	1462.30	211.15
Parallel wires, 23 wires, 2,5deg, 6000kN, 1200mm BS	1282.30	1262.00	1439.30	210.95

When we apply a force of 6000kN, the same stress pattern as for the 2000kN force appears. Increased stress in outer and inner strands and slightly reduced for the core. Why there is an increase in the stresses for the inner and outer core when using the 1200mm bend stiffener is at this point unknown.

A simulation with the coiled wire rope model was run as a comparison to see if the same effects occurred with bending as it did for the pure tension case. There is a difference, though it is not as great as for the pure tension case, it is still a large variance between the coiled and parallel results. These results indicate that the model of parallel wires may not be as accurate as the more true-to-life coiled wire model.

There was an attempted simulation done with the 64wired model, but due to some limitations in Ansys, the simulation failed to converged on a solution and the results from this simulation are therefore not included in this thesis.

5.3 Installation phase

When installing the mooring line onto the buoy the load angle will increase. Even though the tension force in the wire rope is a bit lower than for normal operation condition, the same load was applied in the simulation. Because it is more difficult to know whit certainty the exact load that occurs under installation, the same loads as for operation conditions were applied.

Table 5: Maximum stresses in the inner and outer wire strands and the wire core and maximum total deformation of the entire model

Model and load case	Max Stress [Mpa]			Max Total Deformation [mm]
	Outer	Inner	Core	
Parallel wires, 23 wires, 5deg, 2000kN, 800mm BS	940,47	911,14	1216,5	404,99
Parallel wires, 23 wires, 5deg, 2000kN, 1200mm BS	1009,2	985,7	1186,6	404,68
Parallel wires, 23 wires, 5deg, 2000kN, BS suppressed	1118,3	1094,5	1511	414,47
Coiled wires, 23 wires, 5deg, 2000kN, 800 BS	1048,1	1242,8	1473,2	420,44
Parallel wires, 23 wires, 5deg, 6000kN, 800mm BS	1756,4	1727,4	2023,3	420,98
Parallel wires, 23 wires, 5deg, 6000kN, 1200mm BS	1833,5	1831,0	1999,2	421,76

The same pattern as for the operational condition emerges. Without the bend stiffener there are very high levels of stress compared to the other simulations. This again illustrates the effectiveness of using a bend stiffener. There is not much difference however between the two bend stiffeners and the same fact that the larger bend stiffener actually increases stress levels in the outer and inner strands is also evident in the installation phase.

When 6000kN tension force is applied the stress levels increase to a point where the tensile strength of the wire material is below the maximum stress. This may be cause for concern depending on the safety factor of the wire rope. If the stress levels are too high the wire rope may experience plastic deformation and even breakage of wires.

For the coiled wire test there is also here large stress values compared to the other models. It also shows, contradictory to the two other models, a higher stress value at the inner strand than the outer strand. This confirms the results from the previous simulation and shows that there is a significant difference in axial stiffness between the two wire rope models. This may indicate that the parallel model has insufficient stiffness and the results may not be accurate enough to make a final conclusion.

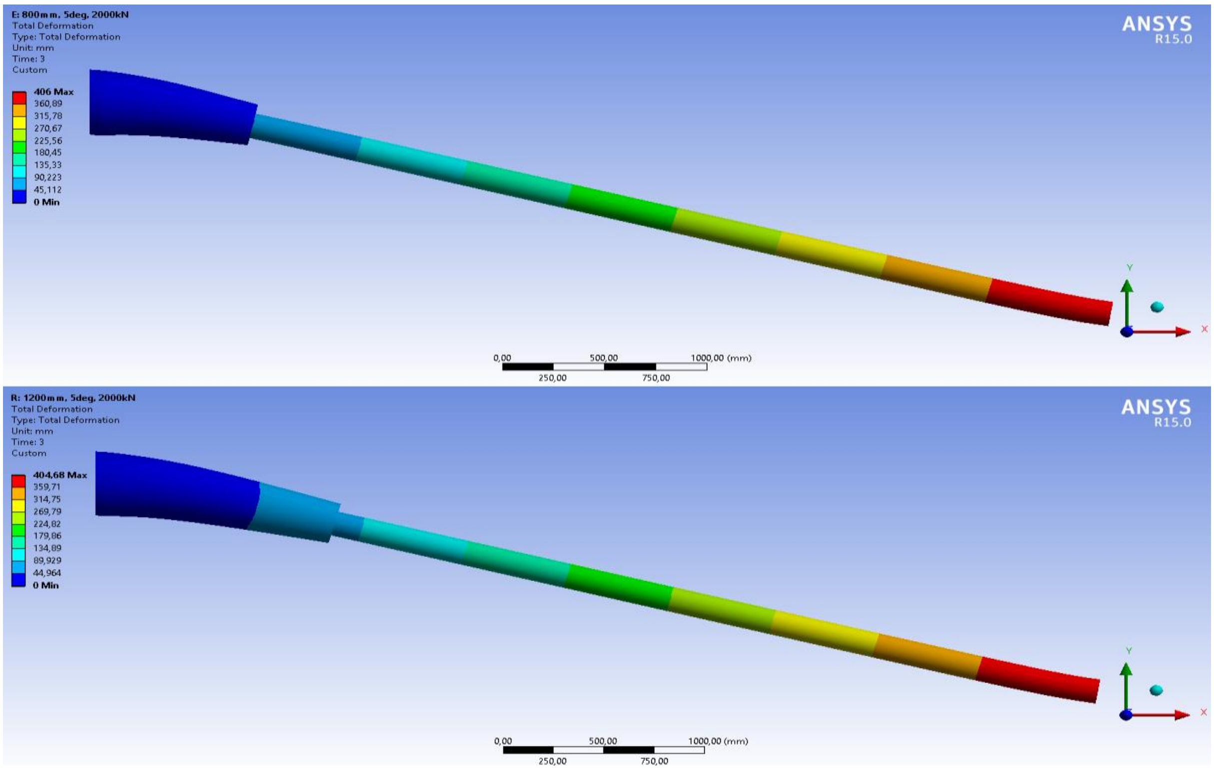

Figure 14: Bending of wire with sheathing and bend stiffener (800mm top, 1200mm bottom) with a 2000kN load at 5 degrees (deformation is shown at 3.1:1 scale)

5.4 Extreme cases

As mentioned in chapter 4.1 and 4.2, the extreme cases have an increased load angle but experiences a much lower number of load cycles compared with an operational state. Due to the uncertainties revolving extreme and damaged cases, the amount of cycles are not easily estimated. Probability considerations should be taken into account but due to time constraints some assumptions and simplifications have to be made.

For a damaged state or in an extreme weather condition it is assumed that a shift in the buoys position will cause at least one of the mooring lines to have an increased load angle, up to 10 degrees. At this load angle, the mooring line will experience the same forces as for the operational condition and installation phase, i.e. 2000 and 6000kN.

Table 6: Maximum stresses in the inner and outer wire strands and the wire core in each of the models and total deformation of the entire model

Model and load case	Max Stress [Mpa]			Max Total Deformation [mm]
	Outer	Inner	Core	
Parallel wires, 23 wires, 10deg, 2000kN, 800mm BS	1627.3	1552.5	2008.0	811.09
Parallel wires, 23 wires, 10deg, 2000kN, 1200mm BS	1798.2	1754.1	1976.9	808.92
Parallel wires, 23 wires, 10deg, 6000kN, 800mm BS	2119.1	2122	2169.1	839.6
Parallel wires, 23 wires, 10deg, 6000kN, 1200mm BS	2093.7	2097.7	2130.7	842.6

In the extreme cases there are several situations where the stress levels exceed the tensile strength of the wire rope material. This is cause for great concern as yielding and eventually complete failure of the wires. The stress levels may however be exaggerated because of increased stiffness in the model compared to the real life wire rope.

5.5 Fatigue calculation

Fatigue life will be based on the stress level results presented in chapter 5. By using a combination of equations 2.1 and 2.2 In order to get a more visual representation of how load range influences the fatigue life of a wire several graphs were created. The graphs are based on a wire with a wire constant, C, of 400 and a variable fatigue factor, m. A load range from 200 to 2000MPa was used to get a good representation of the load range that has been experienced by the wire rope during the simulations. These are also based on a total reversed load, meaning that the minimum stress is zero, and that the UBL is the same for all the mooring types.

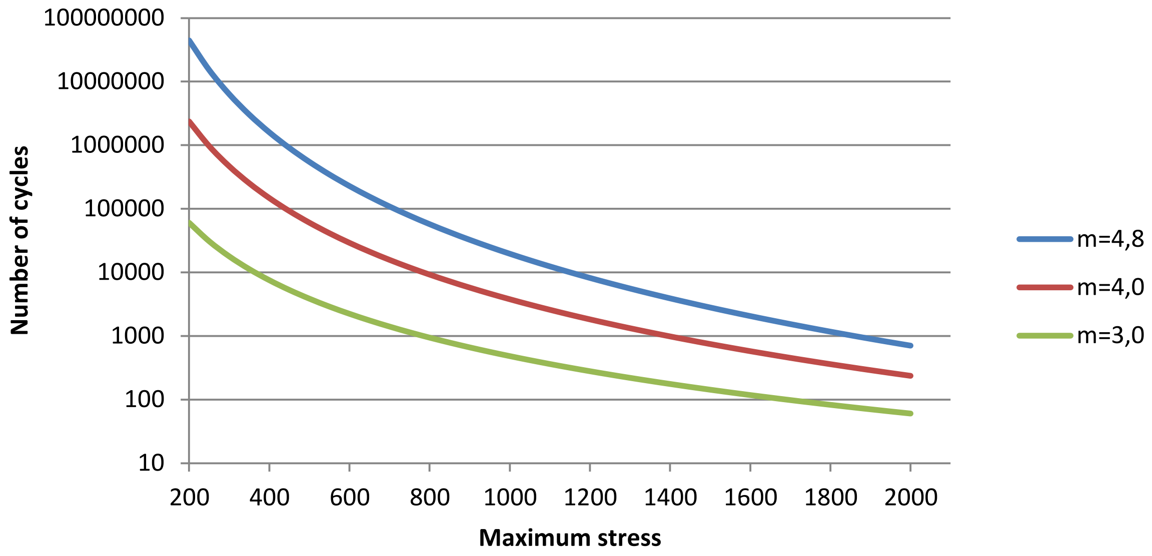

Figure 15: Graphical illustration of number of cycles for a load range between 200 and 2000 MPa. Note: y-axis is logarithmic scale.

From this graph it is clear that the stress level has an extreme effect on the fatigue life of the wire. The fatigue factor m that has been used represents DNV-OS-E301 recommendation for different mooring lines:

Table 7: DNV-OS-E301 recommended values for fatigue factor for different mooring types

Mooring line type	Fatigue factor, m
Spiral rope	4.8
Stranded rope	4.0
Stud chain/Stud less chain	3.0

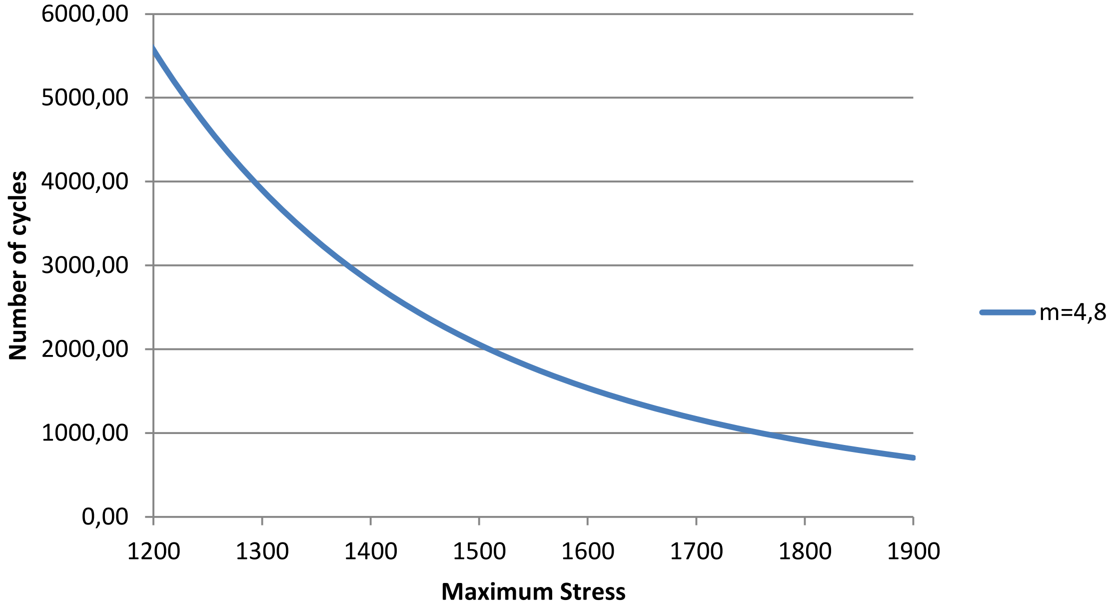

Figure 16: Higher resolution graphical illustration of load ranges between 1200 and 1900MPa for m=4.8

The graphs from Figure 15 confirms that the selected wire rope is the best choice with respect to fatigue life as the properties of the spiral rope is of a nature that withstands fatigue in a better way than the other two alternatives that DNV suggests.

For a pure tension-tension case (zero load angle) the fatigue life wouldn't be a factor. The stress levels are very low and it would take more than 50 years for the wire to fail due to fatigue if we assume that the wire will experience 8500 cycles per year (even higher for the un-coiled, parallel wire model).

5.5.1 Operational

Fatigue life has been calculated based on the equations from chapter 2.5.1, the stresses from chapter 5.2 and the number of cycles from chapter 4.2. A fully reversed load is assumed and the number of expected cycles per year will be 8500 for operational state with a total force of 2000kN and 2000 cycles for a total force of 6000kN. It is also assumed that the structural integrity is undamaged until the point of failure due to fatigue. The load range is based on the highest maximum stress level on any component which would be for the wire core in all the cases. The area of focus has been close to the fixed point of the model. In some of the cases there is a slight “ski jump” effect at the end of the wire rope model as shown in Figure 17.

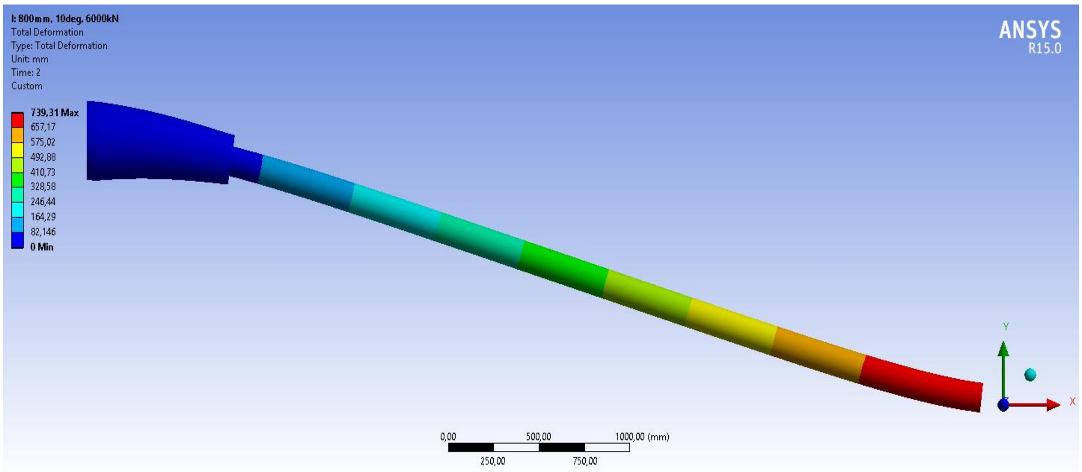

Figure 17: Illustration of the "ski jump" effect on a wire rope under 6000kN load at 10deg. The deformation is scaled up by a factor of 2.7.

This effect is probably caused by a limitation of the rotation of the end block. When the stress data was extracted however, the focus was at the fixed point of the model, meaning that the lower half of the model was excluded when retrieving stress data.

Table 8: Fatigue life calculations for operational state

Model and load case	Equivalent load range, R_q[MPa]	Total number of cycles	Years of service life
Parallel wires, 23 wires, 2,5deg, 2000kN, BS suppressed	43.82	40739.06	4.79
Parallel wires, 23 wires, 2,5deg, 2000kN, 800mm BS	36.86	93407.55	10.99
Parallel wires, 23 wires, 2,5deg, 2000kN, 1200mm BS	36.16	102384.15	12.05
Parallel wires, 23 wires, 2,5deg, 6000kN, 800mm BS	74.61	3166.23	1.58
Parallel wires, 23 wires, 2,5deg, 6000kN, 1200mm BS	73.43	3416.57	1.71

Based on these numbers, the implementation of a bend stiffener more than doubles the fatigue life of the wire rope. If the mooring line is going to be used for more than 5 years it is essential to add a bend stiffener to the mooring design. However if the design life is expected to be above 10 years, in order to have a sufficient safety factor against fatigue failure, it is suggested that an improved bend stiffener be implemented.

For the 6000kN cases it is obvious that the number of cycles to failure is drastically reduced. But knowing exactly how many cycles a mooring line goes through and what the responses are in the line and what force that is actually being taken up by the mooring line is quite difficult to accurately determine without measuring it in real time. The fatigue life is therefore based on an isolated case where there is a constant applied and reversed force and the number of cycles is based on the stress range that occurs in this isolated case.

In real life there will be a combination of both 2000kN and 6000kN forces as well as forces in between, above and below theses forces. In order to reduce the number of variables and to simplify the analysis, only these levels are chosen and are looked at individually. This accounts for all the fatigue life calculations done in this thesis.

5.5.2 Installation

The number of cycles in an installation phase is based on the same cycles as for the operational state. Even though the weather requirements when installing may be stricter than for standard operation, the amount of cycles will be assumed the same for both cases. But since an installation of the wire is not a matter of years, the scale of service life will be in days. Forces that the mooring line experiences are most likely smaller during installation than under operational conditions. This is because of less tension in the wire when the buoy is submerged.

Table 9: Fatigue life calculations for an installation phase

Model and load case	Equivalent load range, R_q[MPa]	Total number of cycles	Days of service life
Parallel wires, 23 wires, 5deg, 2000kN, 800mm BS	62.07	7659.12	328.89
Parallel wires, 23 wires, 5deg, 2000kN, 1200mm BS	60.54	8630.90	370.62
Parallel wires, 23 wires, 5deg, 2000kN, BS suppressed	77.09	2705.50	116.18
Coiled wires, 23 wires, 5deg, 2000kN, 800 BS	75.16	3055.35	131.20
Parallel wires, 23 wires, 5deg, 6000kN, 800mm BS	103.23	666.24	28.61
Parallel wires, 23 wires, 5deg, 6000kN, 1200mm BS	102.00	705.68	30.30

Normally an installation would only take hours to complete, but installation may happen several times for a buoy. If we assume one installation takes 3 hours to complete, and the load angle would stay at 5 degrees during the entire installation, for the worst case scenario the wire would be able to handle more than 200 installations. Even without a bend stiffener, the amount of installations would be high enough to not be a concern.

The wire would not likely fail by fatigue during installation itself but in real life this would be in combination with regular operation fatigue meaning that installation after a few years of services might be a larger cause for concern.

5.5.3 Extreme case

Extreme weather conditions happen only on rare occasions. Based on the wave criteria mentioned in chapter 4.2, an assumption of 100 cycles per year would be a good approximation. The extreme cases also include a damaged state of the buoys mooring system, i.e. if a wire line snaps and the buoy shifts position.

Table 10: Fatigue life calculations for extreme weather conditions and damaged cases

Model and load case	Equivalent load range, R_q[MPa]	Total number of cycles	Years of service life
Parallel wires, 23 wires, 10deg, 2000kN, 800mm BS	102,45	690,96	6,91
Parallel wires, 23 wires, 10deg, 2000kN, 1200mm BS	100,86	744,72	7,45
Parallel wires, 23 wires, 10deg, 6000kN, 800mm BS	110,67	477,06	4,77
Parallel wires, 23 wires, 10deg, 6000kN, 1200mm BS	108,71	519,77	5,20

5.6 Suggested new design

Based on the results from the wire analysis, there is a clear room for improvement on fatigue life. Because of the high number of cycles a small decrease in stress load range will have a great impact on the fatigue life of the wire rope. Reducing the stress should therefore be a top priority when trying to improve the fatigue life. In order to do this a new design proposal of a bend stiffener has been made and run through same simulation as for the existing bend stiffeners.

The new design features a slightly longer and much wider bend stiffener than the ones that are being used now. It is over-dimensioned but within a reasonable size, meaning that it is far larger than the others but still within what could be actually be used. This has been done on purpose in order to see how much improvement can be done by adjusting the dimensions of the existing bend stiffeners.

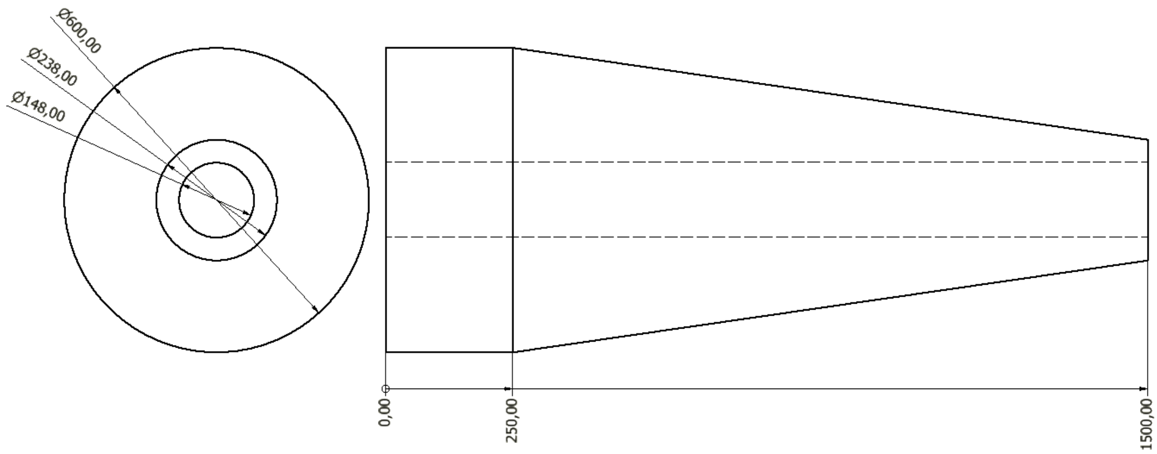

Figure 18: Drawing of suggested new design of bend stiffener

This design is too large for the existing end termination, so a slight modification here is needed in order to implement this new design. The OD of the bend stiffener is as shown in Figure 18. While the plate that the bend stiffener is bolted to is much smaller and must be enlarged in order to fit with this new design.

Table 11: Maximum stresses in the inner and outer wire strands and the wire core in each of the models and total deformation of the entire model

Model and load case	Max Stress [Mpa]			Max Total Deformation [mm]
	Outer	Inner	Core	
Parallel wires, 23 wires, 2,5deg, 2000kN, custom BS	510.37	488.00	510.88	190.63
Parallel wires, 23 wires, 2,5deg, 6000kN, custom BS	1176.8	1146.3	1243.2	204.57
Parallel wires, 23 wires, 5deg, 2000kN, custom BS	763.75	718.96	787.66	380.64
Parallel wires, 23 wires, 5deg, 6000kN, custom BS	1629.2	1585.8	1776.5	407.97

The results show a very high stress reduction across all the components in all the load cases. This shows that implementing a new design of the bend stiffener might be quite favorable when wanting to increase the fatigue life in certain situation.

Since the bend stiffener is in fact larger and a slight modification of the end termination is needed, a cost to performance decision must be made before choosing to change the design of the existing bend stiffeners.

5.6.1Fatigue calculation on new design

With the reduction in stress level there is an increase in service life as shown in Table 12.

Table 12: Fatigue life calculations for installation phase with suggested design of bend stiffener

Model and load case	Equivalent load range, R_q[MPa]	Total number of cycles	Years of service life
Parallel wires, 23 wires, 2,5deg, 2000kN, custom BS	26.07	492931.28	57.99
Parallel wires, 23 wires, 2,5deg, 6000kN, custom BS	63.43	6901.13	3.45

There is a massive increase in service life for the operational phase. Though the stress levels are not vastly lower, the amount of cycles makes a small decrease of stress have a huge impact on the service life of the wire rope.

Model and load case	Equivalent load range, R_q[MPa]	Total number of cycles	Days of service life
Parallel wires, 23 wires, 5deg, 2000kN, custom BS	40.19	61700.82	2649.51
Parallel wires, 23 wires, 5deg, 6000kN, custom BS	90.64	1243.95	53.42

The results show that the service life is more than 20 times longer with the suggested design than for a case without a bend stiffener at all. If compared to the best case scenario with the 1200mm bend stiffener, there is an increase of more than 700%.

The increases in service life are quite substantial. This confirms that the suggested bend stiffener design might be over-dimensioned. A slightly smaller design may not increase service life by such a margin but still provide a better cost to performance ratio.

5.7 Software and alternatives

Ansys was used in all simulations in this thesis. Though a powerful program with a good reputation, in the cases that has been studied here, Ansys may not be the most ideal program. Problems with the more real-to-life models and very long simulation times made the simulations ineffective and several times the simulations failed unexpectedly and hours of simulation time lost. Other programs such as Abaqus or Comsol might have been better alternatives than Ansys.

Inventor as the CAD software was the preferred choice. But also here there were some problems during the model creation. For some reason when creating the coiled wire rope the Inventor program started to freeze and stutter and eventually shut down completely. Whether using a different CAD software would have solve this issue is unknown.

6 Future work

In this thesis, a simplified wire model has been used for most of the analysis. For a more accurate representation of the real world case a model consisting of more wire layers and a precise lay angle to make the rope less torsionally active should be further investigated in order to make a final conclusion on the actual real effect of the bend stiffeners. This would require increased computing power and possibly different software than what has been used in this thesis.

The load cases have been limited to only include the tension forces of 2000kN and 6000kN. As mentioned in a real situation these tension forces vary a lot with different sea states and weather conditions and isolated cases like the ones looked at in this thesis is not the most true to nature cases. Looking at a more dynamic load with added effects from drag forces induced by movement of the mooring line and underwater currents should be something to consider for future study.

Effects of corrosion have not been taken into account either. Even though the wire rope is sheathed there is still a risk of seawater entering the sheathing and corroding the wire. This will in time degrade the structural integrity of the wire and may lead to stress concentration points such as cavities.

An alternative the steel wire rope used is a fibre rope mooring line. Fibre rope compared to steel rope is much more flexible and more lightweight. A common misconception with fibre rope is that it has a lower carrying capacity, but modern fibre ropes have almost equal tensile strength as its metallic counterparts and its reduced weight decreases the tensile forces in a mooring scenario as less of the mooring lines own weight contributes to the tension force. Because of its increased flexibility it also has a greater potential to withstand bending fatigue.

For the extreme cases, it may be possible to create an emergency bend stiffener which can be attached to the end termination when weather forecasts indicate that there may be severe storms on the way. This would then lock around the existing bend stiffener and provide extra stiffness when required. It would also be detachable and reusable so it could be used on other equipment as well. This could be a solution to the very high stress levels that occur in the most extreme conditions and if other mooring lines are damaged or broken.

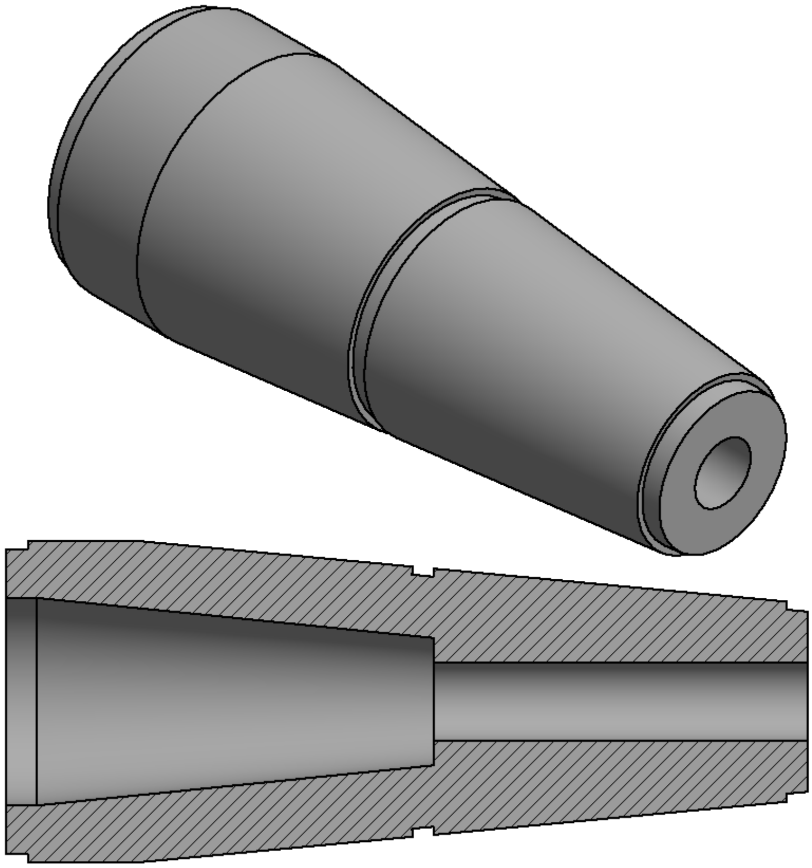

Figure 19: 3D model of a proposed emergency bend stiffener design

The design of this emergency bend stiffener would be two halves of a bigger version of a regular bend stiffener with one or more groves for fastening clamp(s) to around it to clamp onto the existing bend stiffener. The emergency bend stiffener would have a conical ID to fit the existing bend stiffener within itself.

7 Conclusion

The results shows that stress levels in the installation phase and the operation condition are well within the limits of the material, but for the extreme/damaged case the stress in the wires exceed the tensile strength of the wire material. This is a point of concern for the wire and measures should be taken to avoid wire rope failure.

The comparison between the existing bend stiffeners, the 800mm and the 1200mm models show insignificant differences in the stresses in the wires. A clear increase in stress can be seen when a bend stiffener is not installed. This concludes that the implementation of a bend stiffener is important but that there is little difference between the two existing models.

Fatigue life of the wire rope is rather low in the operational condition based on the amount of cycles that has been used in the fatigue life calculations. For installation phase the life of the wire rope is within an acceptable limit. Extreme weather or in a damaged state the fatigue life is acceptable with only a small amount of cycles per year. But if the extreme weather persists or if some of the mooring lines are broken and is not repaired within a short period of time, concerns arise of the longevity of the wire rope life time.

By implementing a new design of the bend stiffener it is possible to drastically increase the life time of the wire rope, especially in the operational conditions.

Due to limitations within Inventor, the wire models created were simplified to a wire rope with 64 and 23 wires. The lay angle was also removed because of problems with simulation convergence in Ansys. The final wire models were simulated in Ansys with different load cases. The simplification of the wire models may have caused the model to be stiffer than the real wire rope but the objective of the thesis was to compare the effects of the bend stiffeners, therefore it was concluded that as long as the wire models where consistent the results would still be valid and comparable.

8 References

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[6] F. Geos, *Wind and wave frequency distributions for sites around the British Isles*: Great Britain, Health and Safety Executive, 2001.

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9 Appendix

9.1 User interfaces

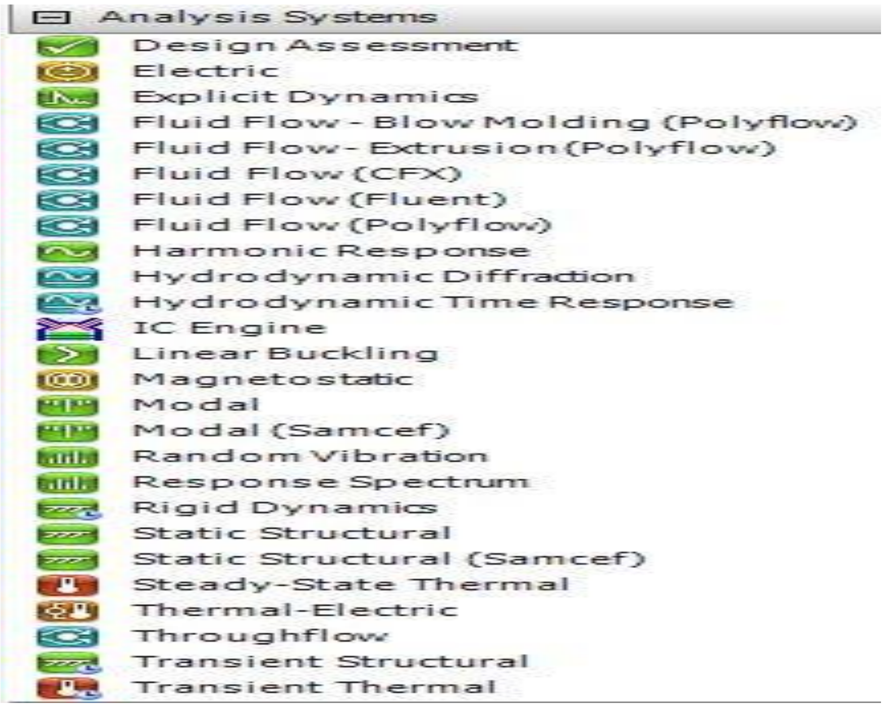


Figure i: The available analysis systems for Workbench

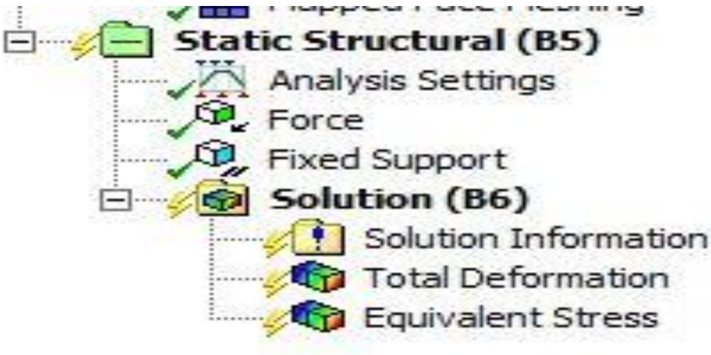


Figure ii: Solution tab in Workbench with total deformation and equivalent stress

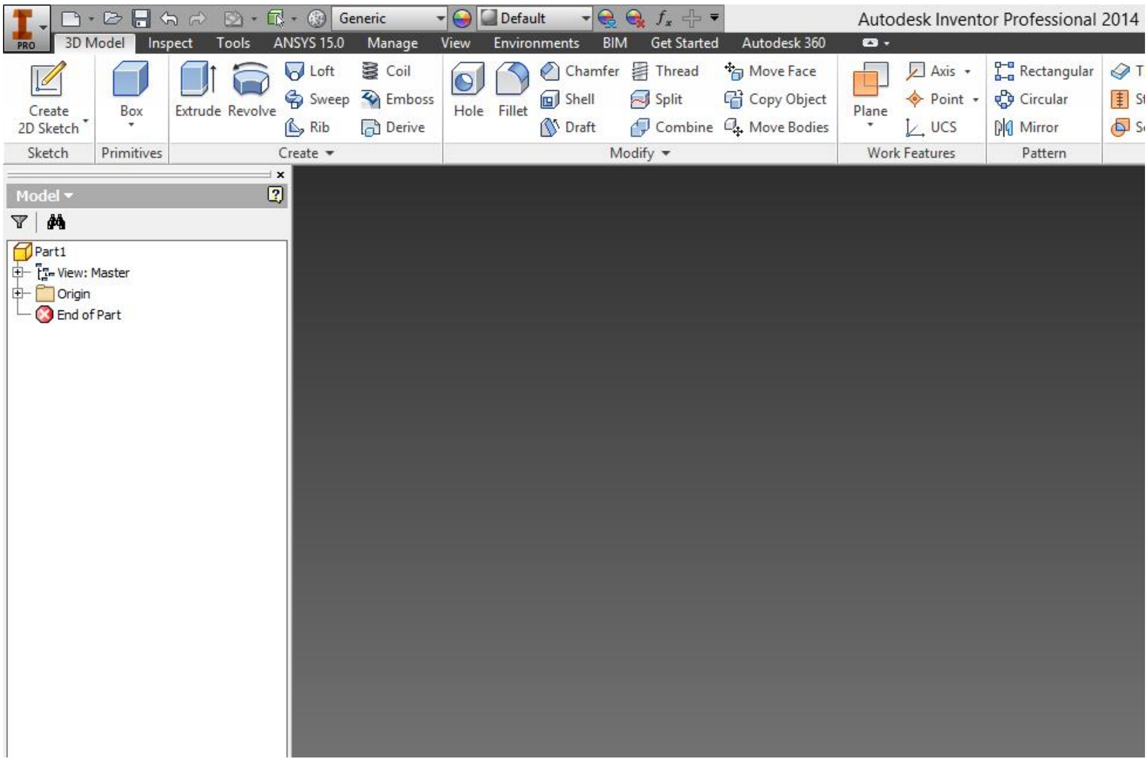


Figure iii: Autodesk Inventor 2014 user interface

9.2 3D models

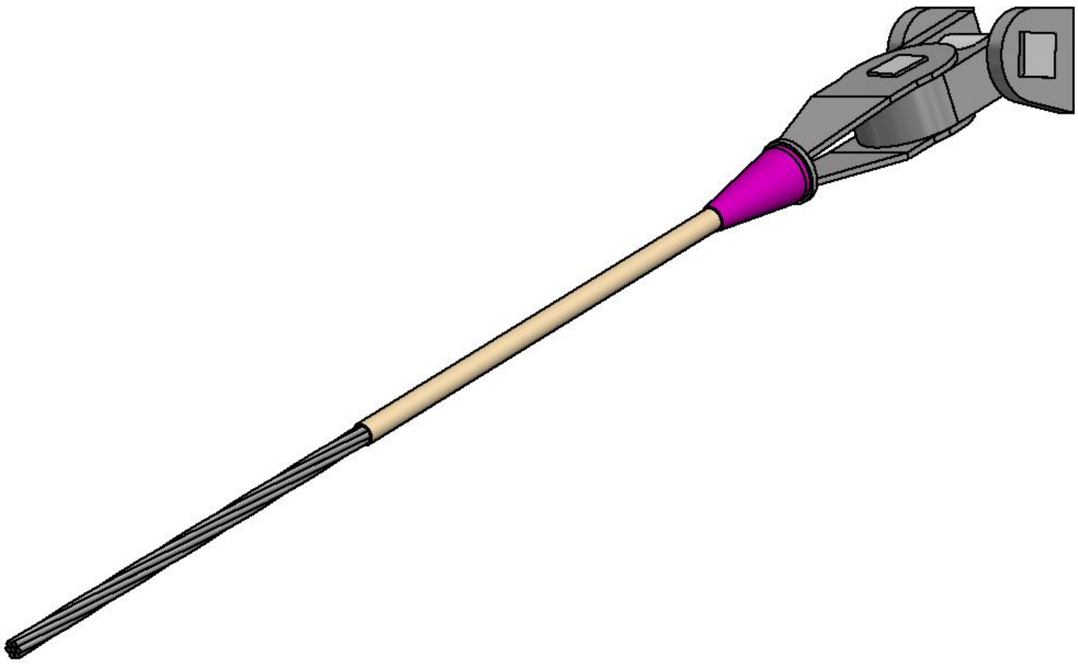


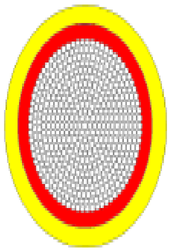
Figure iv: 3D model of complete end termination

9.3 Wire rope technical data

LOW TORQUE GALFANIZED SHEATHED SPIRAL STRAND

TECHNICAL DATA

<u>For 25 years life time</u>



Wire Rope parameters:

Wire rope steel nominal diameter :	120 mm	-0/+2%	on steel
Wire rope sheathed nominal diameter :	164 mm	-6,6/+9mm	on HDPE jacket
Metallic cross section :	8605.4 mm^2		
Steel cross section :	8254.5 mm^2	average wires Corzal coating thickness : 0,05+0,05 = 0,1 mm	
Total rope weight :	80.92 kg/m	+/-1.5 %	in air
Total submerged rope weight :	58.78 kg/m	+/-1.5 %	in sea water
Minimum breaking load (MBL) :	14800 kN		
E modulus value :	170 kN/mm^2	+/-10	after 3 (or more) cycles from : 10 to 60 % of MBL
EA value :	1.41E+09	N	
EI value :	2.54E+09	N/mm^2	

Construction : see separated calculation note on attached wire rope construction sheet.

All wires are hot dipped galfanized with "Corzal" coating on wires (305 gr/m^2 minimum, as per ASTM A 856 class 100 standard).

Steel grade : 1880 MPa nominal (1940 MPa average tensile strength)

Rope filling (blocking compound) : Ovoline (see attached data sheet)

Minimum bending ratio : Standard values (See our separated calculation note for more details)

D/d	26	under spooling tension (maximum 5 T)
D/d	38	under 5% of MBL tension
D/d	46	under 10% of MBL tension

Admissible twist during installation : 2 turns / reference length of 1000 m ie : 0.28 turn(s) for line of : 140 m

Torque factor :	Torque (in N.m) =	4.50E-01	x Tension (in kN)
Rotation factor :	Rotation (in Radian / m) =	1.60E-06	x Tension (in kN)

HDPE jacket thickness : 22 mm according to ASTM D-1248 (98) standard.

Maximum admissible radial pressure on HDPE jacket : 20 MPa

Friction coefficient between sheath and spiral strand (nominal value) : 0.1

Figure v: Technical data sheet of dual sheathed wire rope

LOW TORQUE GALFANIZED SHEATHED SPIRAL STRAND
DETAILED CONSTRUCTION

Lay N°	Wires number	Wires diameter	Layer diameter	Lay direction	minimum average tensile strength (/Mpa)
1	1	6.5	6.5	Left	1650
2	6	4.85	16	Left	1940
3	12	4.85	25.5	Left	1940
4	18	4.85	35	Left	1940
5	24	4.85	44.6	Right	1940
6	30	4.85	54.2	Left	1940
7	36	4.85	63.8	Left	1940
8	42	4.85	73.45	Right	1940
9	48	4.85	83.1	Left	1940
10	54	4.85	92.75	Left	1940
11	59	4.85	102.5	Right	1940
12	65	4.85	112.2	Left	1940
13	70	4.85	121.9	Right	1940

Theoritical minimum spinning loss factor (%): 11
Helix factor : 1.059
Outer lay length : 1055 mm
Aggregate breaking load (ABL) : 16684.78 kN

Figure vi: Detailed construction of wire rope

9.4 Simulation images

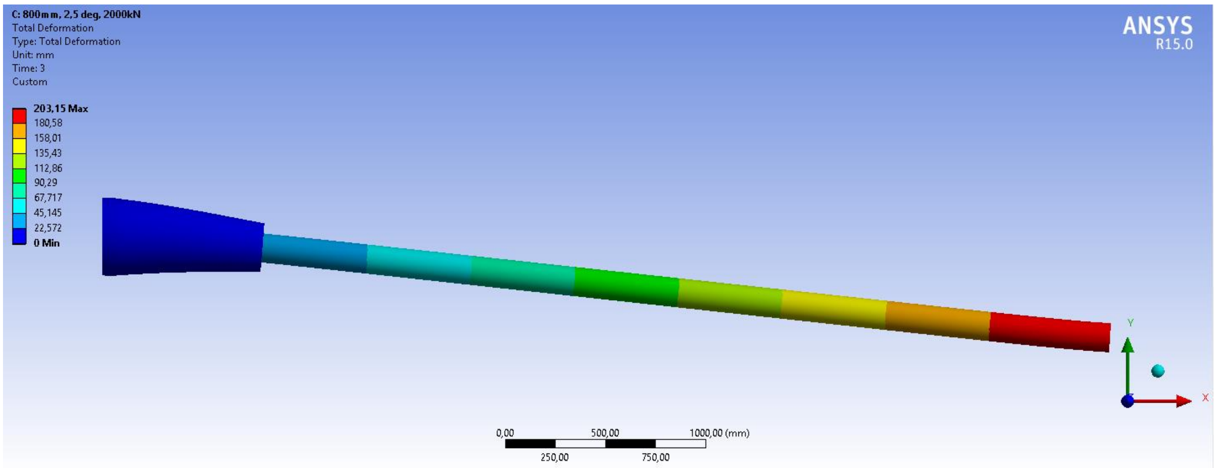


Figure vii: Screen shot of simulation environment with wire rope, 800mm bend stiffener and sheathing. Load case: 2000kN at 2.5 degrees

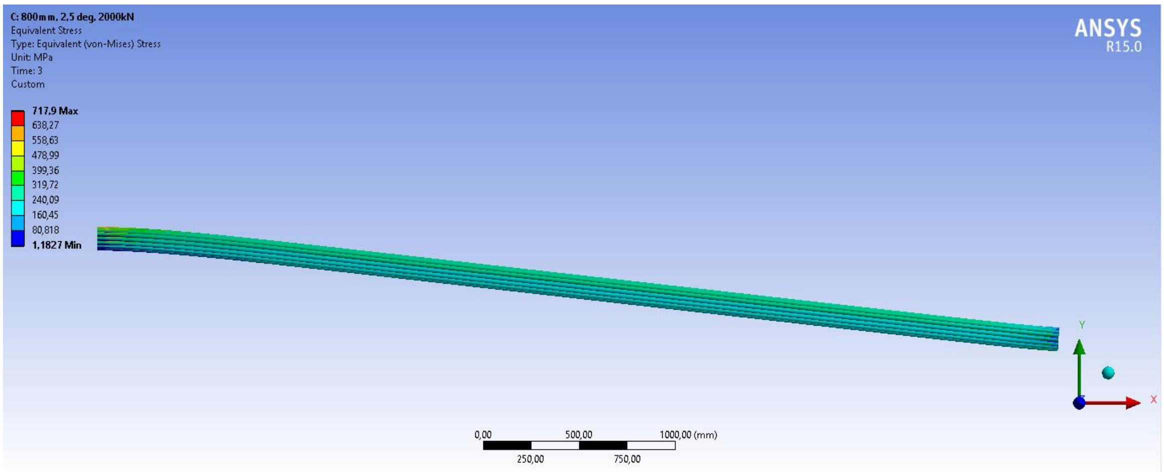


Figure viii: Same as figure vii, bend stiffener and sheathing are not shown.

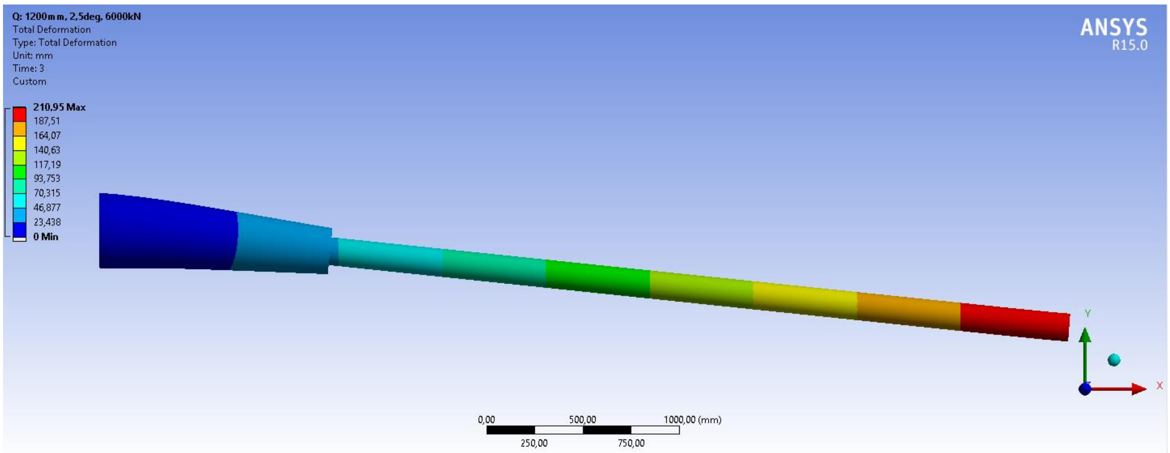


Figure ix: Screen shot of simulation environment with wire rope, 1200mm bend stiffener and sheathing. Load case: 6000kN at 2.5 degrees

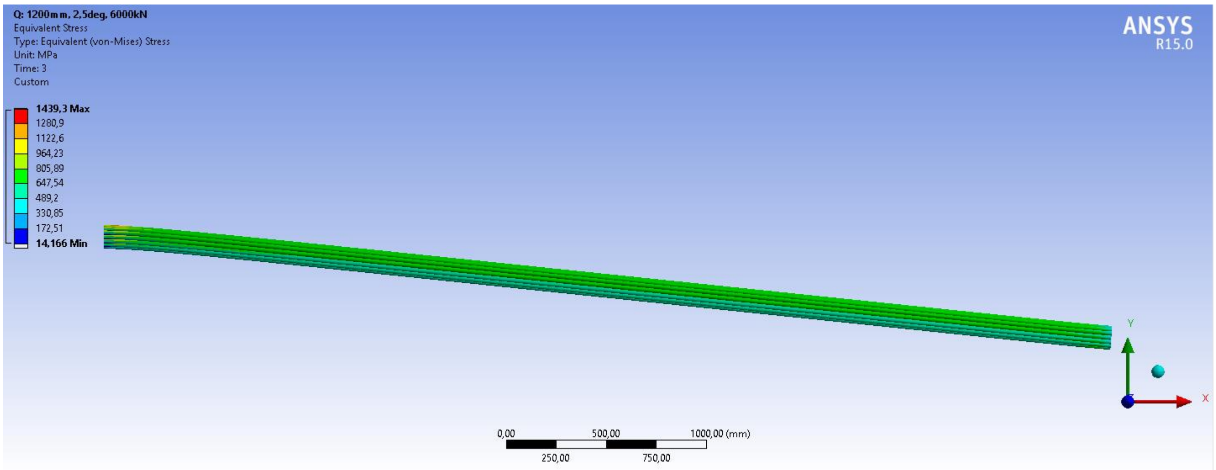


Figure x: Same as figure ix, bend stiffener and sheathing are not shown