PIONEERING SPIRIT

DESIGN OF FLEXIBLE CONNECTION SYSTEM ELEMENTS

AND

DAMAGE INSPECTION OF SLINGBUOY 2.0

Internship report

Bas Ooms
Allseas Engineering B.V.
University of Twente

CONFIDENTIAL

Internship report by:
B. Ooms BSc.

For the master programme of:
Mechanical Engineering
University of Twente
Faculty CTW
Drienerlolaan 5
7522 NB Enschede
The Netherlands

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At the company:
Allseas Engineering B.V.
Innovation department
Capitool 64
7521 PL Enschede
The Netherlands
www.allseas.com

Supervisors Allseas:	Ir. A. Huisman Ir. M. Esselbrugge
Supervisor University of Twente:	Prof. Dr. Ir. A.H. van den Boogaard
Coordinator University of Twente:	Dr. Ir. D. van de Belt

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1.0 INTRODUCTION

Allseas is an offshore contractor founded in 1985 by Edward Heerema. The company has worldwide experience in pipe laying and the installation of subsea structures. Allseas has more than 3000 employees worldwide, both offshore and onshore. Allseas' key qualities are: a no nonsense mentality, pioneering, in house development and rapid progress. This is best illustrated by the build of the newest vessel *Pioneering Spirit.* The *Pioneering Spirit* is the world's largest ship, in surface area, and is currently being finished in the port of Rotterdam. This ship has the ability to lift entire top sites of drilling platforms with a mass up to 48.000 t and is fully developed in house. The rest of Allseas vessels are conversions from bulk carriers, all conversions are engineered in house. Information about the fleet can be found in Table 1.1.

Table 1.1 Allseas fleet

Name	**Type**
Pioneering Spirit	Heavy lift vessel / Dynamic positioned pipelay vessel
Solitaire	Dynamic positioned pipelay vessel
Audacia	Dynamic positioned pipelay vessel
Lorelay	Dynamic positioned pipelay vessel
Tog Mor	Anchor positioned pipelay vessel
Calamity Jane	Support vessel
Bright Spark	Training vessel

I did my internship for the innovations department in the Enschede office. The innovations department is mainly situated in the office in Delft. A branch office in Enschede has been set up in January 2015 in order to be more connected to the University of Twente. Initially my internship assignment was about the *Slingbuoy* 2.0 project. This was a prototype of a floating sling for A&R operations aboard of the *Lorelay*. This prototype has been tested and got damaged during the tests. My assignment was to inspect the sling and determine how it had failed, and to validate this in a test setup. This will be elaborated in Chapter 3.0. Due to new insights and budget cuts this assignment has been put on hold. The result of this is that the last section of Chapter 3.0 is more concise and not fully finished.

This led to an alternative assignment for the remaining time of my internship. This new assignment was also about A&R equipment, but this time for the *Pioneering Spirit.* For this assignment I was involved in the early stages of the design a flexible connection system (FCS). This assignment will we elaborated in Chapter 2.0.

2.0 DESIGN OF FLEXIBLE CONNECTION SYSTEM ELEMENTS

2.1 General

2.1.1 Document scope

The focus of the report is on the design of the flexible connection system (FCS) elements. This FCS is part of the cable connector tool (CCT) for A&R operations on the *Pioneering Spirit.*

2.1.2 Abbreviations

The following abbreviations are applicable for this particular document:

A&R	- Abandonment and recovery	CCT	- Cable connector tool
ILT	- Internal lifting tool	FCS	- Flexible connection system
PRT	- Pipe recovery tool	WLL	- Working load limit
STF	- Stinger transition frame	FEA	- Finite element analysis
OD	- Outer diameter	WD	- Water depth

2.1.3 Introduction to CCT

An essential part of the pipe tensioning system of *Pioneering Spirit* is the A&R winch, which facilitates abandoning, recovering and laying-down of the pipeline. The A&R system consists of four 500-t traction winches. By combining the four 500-t traction winches a maximum tensioning capacity of 2000 t can be reached.

For the A&R winch an A&R Cable Connector Tool (CCT) has to be developed. This CCT shall be able to connect to and disconnect from the pipe, both in the firing line and at the seabed. To not limit the tension capacity of the vessel by the CCT, the CCT should have a capacity of 2000 t.

An A&R solution with a tension capacity of up to 500 t is already available for the *Pioneering Spirit*. Therefore a CCT solution for the range of 500 t to 2000 t is required. This solution should be able to cope with different stinger radii. These radii are the result of the installation requirements for projects like the pipe OD, WD and tension.

Due to time constraints, the initial A&R CCT design was based on the design of the current 1000-t connector on the Solitaire. However, upscaling of this system (a C-hook), resulted in a large and heavy connector that would not be feasible to handle. Another hook design, which resulted in a smaller, operationally more favourable hook was then developed. However, in the course of the development process, it came forward that the use of this concept on *Pioneering Spirit* would introduce problems that could not be solved in a satisfactory way. The main problems are due to the high rollerbox loads and the resulting geometrical limitations on the equipment traveling over the stinger. The connector (hook) that was developed was not able to meet the requirements set out for safe stinger passage.

Because of the suspension of the first pipelay project of *Pioneering Spirit* it became feasible to venture other options that would meet requirements for stinger passage and that would result in a fast and workable operation.

2.1.3.1 Current concept

Since upscaling the A&R hook from *Solitaire* is not an option the focus shifted to an internal lifting tool, something comparable with a Pipe recovery tool (PRT). A PRT can grab a fallen pipe on the inside and lift it to the ship in order to resume pipe laying after a broken or runaway pipe. The requirements of an internal lifting tool (ILT) are fully described in [1]. Figure 2.1 shows an overview of the global design of the CCT with an ILT. In order for this concept to work, the female connector should be in a vertical position when the system is at the seabed. This requires a flexible connection system (FCS) to connect the female connector to the A&R head.

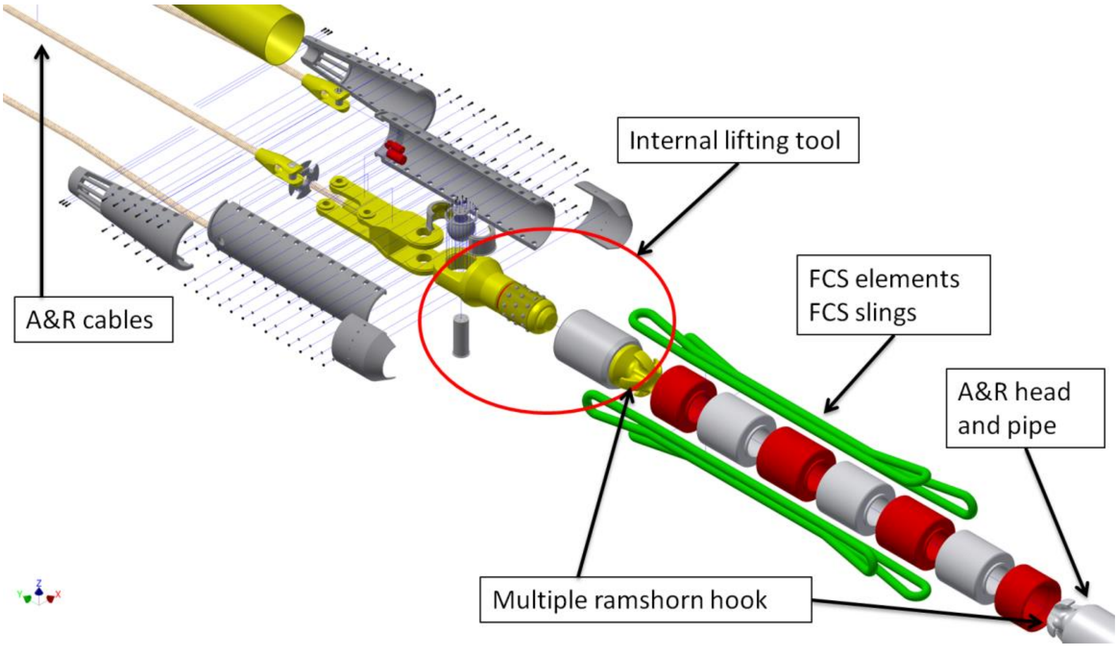


Figure 2.1 Overview of the cable connector tool

2.1.3.2 Flexible Connection System

There are two main components of the FCS, a sling and FCS elements, as can be seen in Figure 2.2. The sling is considered as a system, it can be a single sling with the full WLL of the system or multiple slings in a basket formation that combined has the required WLL of the system. To protect the slings and prevent the individual rollerboxes on the stinger to overload, surrounding elements are added. These FCS elements must have the same outer diameter as the rest of the CCT and be strong enough to pass over the stinger. For clarity the slings in Figure 2.1 and Figure 2.2 are on the outside, in reality they will be on the inside of the elements.

On both sides of the sling there will have to be an interface, to transfer the tension on the sling, this will be in the form of a multiple ramshorn hook. This hook can be seen in both Figure 2.1 and Figure 2.2 is made from the same piece of forged steel as the receiver, thus it is officially not a part of the FCS. The design of the hook will be incorporated in the design of the entire FCS. This way the best combination of hook, slings and elements can be achieved. At this moment the decision has been made to make use of triple horn hook. The images show an alternative design with a quad hook in combination with four independent slings.

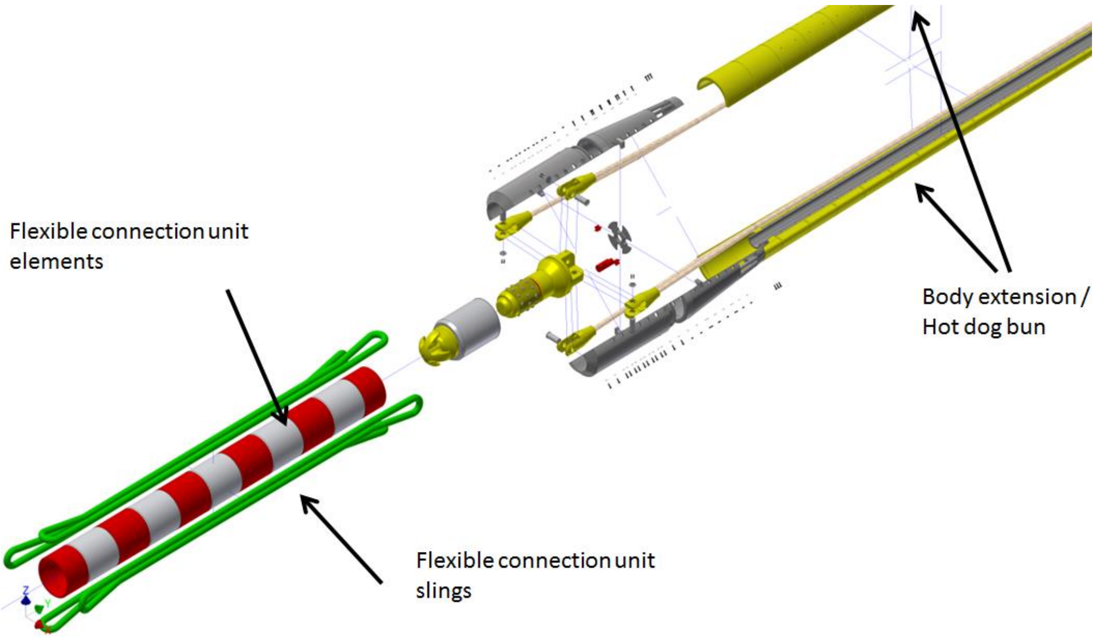


Figure 2.2 Flexible connection system (FCS)

2.2 Requirements

The requirements for the FCS are stated in the technical specifications [2]. Some of those specifications apply to the entire system and some are more specific for the sling system or the elements. The requirements stated in this document are based on a revised list of requirements as described in memo [3]. Only the most relevant requirements are stated here, but the elements must fulfil all requirements as stated in memo [3].

2.2.1 Functional requirements

The sling should be able to bear the tension of the A&R system. The nominal tension of the system will be maximum 2000 t. The brake capacity of the A&R winches is 550 t for each of the four winches therefore the WLL of the sling system will be set at 2200 t.

The functions of the FCS elements around the sling system are:

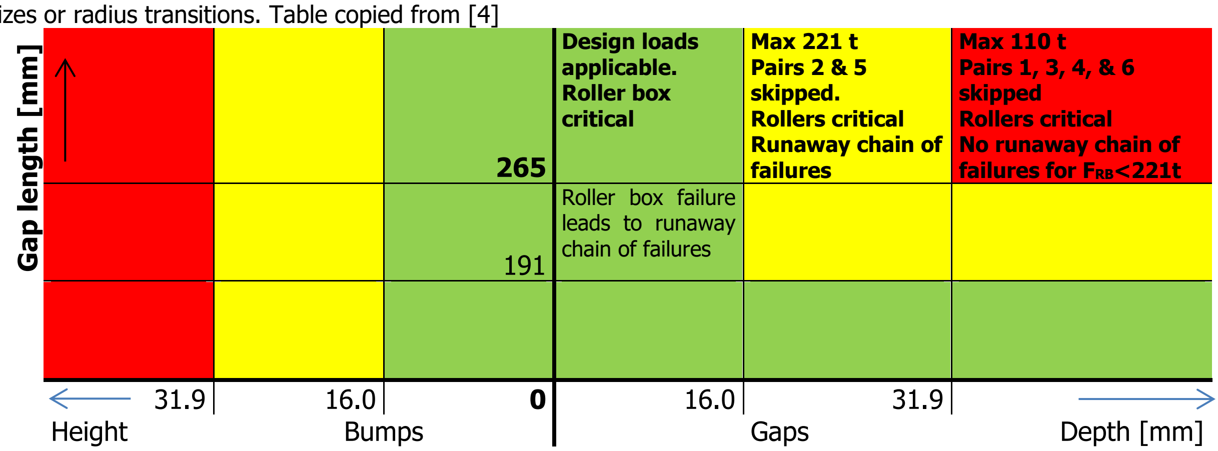
- Protect the sling;
- Provide the sling with safe stinger passage;
- Allow the sling to bend with a small bending radius at the seabed;
- Prevent the sling from detaching the hook.

2.2.2 Geometrical requirements

- The sling system shall be 12 metre long.
- The interaction between the sling and the FCS elements shall be safe for both parts.
- Regular inspection of the sling shall be possible.
- The outer diameter of the FCS will be 1150 mm.

Next to the requirements stated above, there is an important requirement about the outside of the entire system. Based on analytical calculations [4] on the roller and rollerbox loads, the outside diameter of the entire CCT may not have large differences in radius. This is graphically explained in Table 2.1, The conclusion is that all gaps or bumps should be shorter than 191 mm or have less than 16 mm difference in radius. These values are the most conservative. This calculation has been done for the rollerboxes on the stinger, Table 2.1, but also on the two types of rollerboxes in the STF. The rollerboxes on the stinger were limiting, so those values were used as requirement.

Table 2.1 Graphical representation of maximum allowed roller box loads for various combinations of limiting gap sizes or radius transitions. Table copied from [4]



2.2.3 Load cases

Multiple load cases are defined for the FCS, these are described in memo [3]. In order to aid the design process the largest load on one element is used during the preliminary design.

- A lateral load of 500 kN on one element, on one stinger roller pair, applied as can be seen in Figure 2.3.
- A vertical load of 500 kN at the seabed, with the receiver upright, the FCS should be able to withstand these loads (perpendicular to the water surface) in both up- and downward directions.

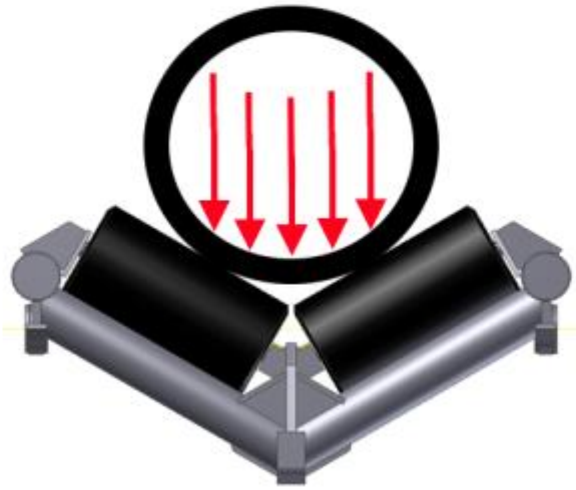


Figure 2.3 Simplified load case for one FCS element

2.2.4 Bending requirements

At the seabed the FCS should be able to allow the receiver to be in an upright position while half of the FCS still lays at the seabed. This will prevent unintended picking up of the pipe due to heave of the vessel. Therefore the FCS should be able to bend 120˚ with a bending radius of approximately 3 m. This will allow the receiver to be upright even on a downward slope of the seabed. And it will also allow approximately half of the FCS to remain on the seabed. In order to keep half of the FCS at the seabed a clump weight should be added halfway the FCS.

2.2.5 Lifetime

The hardware terminations shall be designed and specified such that their lifetime shall be 25 years and approximately 500 A&R operations. For components where 25 years is not realistic, the lifetime will be specified by the designer in accordance with their design. All components will at least be suitable for 30 A&R operations to ensure they do not require replacement during a single project.

2.3 Design of the FCS elements

In order to design the FCS elements some information about the hooks and the sling configuration is known. At this moment the information is limited. Based on the dimension of the sling legs calculations have been made to determine the smallest diameter of the sling bundle. A smaller diameter leads to a smaller sling or a higher factor of safety within a given diameter as described in [5]. Three slings leads to a smaller bundle compared to four slings. This results in the decision for a triple horned hook but the exact dimensions of the hook are not know yet. This hook design will lead to two possible sling configurations. One sling that is wrapped around all three "horns" of both hooks. The other option consists of three slings each on their own "horn" of the hooks. Based on a model [6] and reasonable manufacturing tolerances it is concluded that three slings is not the desired option. Since it would require manufacturing tolerances of less than 0.1% between the sling lengths of the multiple slings.

2.3.1 Selection of the interface between the FCS elements

In order to make a decent choice for the interface between the FCS elements some ideas are worked out and investigated in this section.

2.3.1.1 Selection criteria

To make a well informed decision on the interface between the elements all ideas are tested on five selection criteria. These five criteria are specified and elaborated here. In the next sections they are used to give scores to the ideas. The given scores are on a scale from one to five, five being the best and one being the worst.

Load in compression

The elements will have to be lightweight, in order to reduce the necessary buoyancy, therefore the elements will have to be made of a polymer. Polymers tend to have a lower yield stress in tension compared to the yield stress in compression. Because of this it is an advantage if the FCS elements are only loaded in compression. A high score will be given if the FCS element is only loaded in compression, a low score if the element is also loaded in tension.

Force path
The force path from the sling to the rollers should be as direct as possible. To illustrate the meaning of this, two extreme cases shown in Figure 2.4 and Figure 2.5. A more direct force path will lead to more uniform distribution of stresses through the FCS element. The more direct the force path is the higher the score, less direct will lead to a lower score.

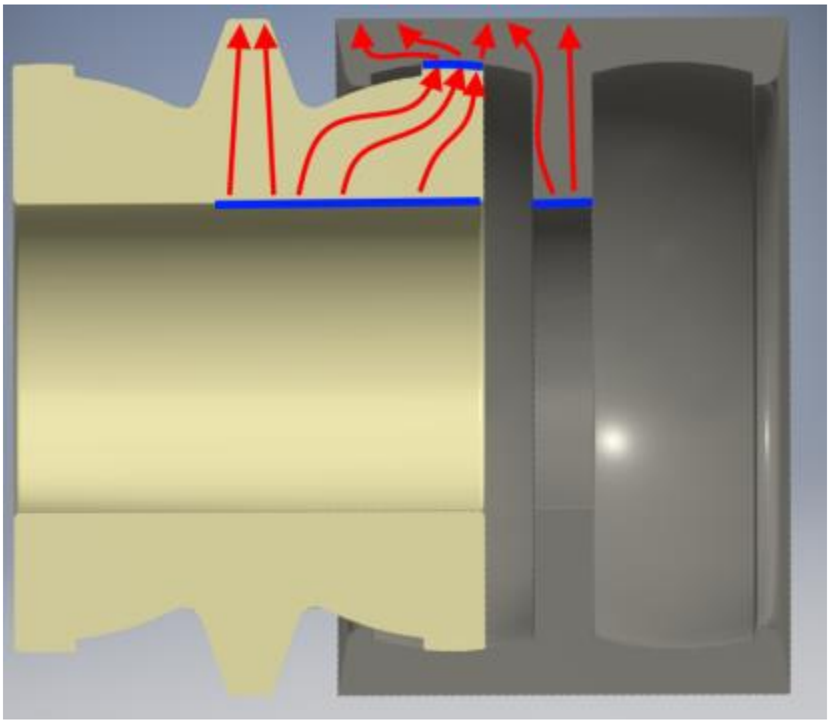


Figure 2.4 Indirect force path through multiple bodies. Blue is application point of the force, red is the force path.

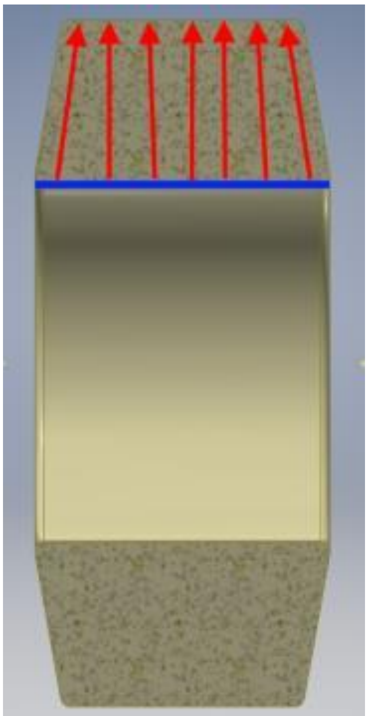


Figure 2.5 Direct force path through one body. Blue is application point of the force, red is the force path.

Smoothness of outer surface
In order to travel safely over the stinger, the FCS elements should have smooth outer surface. The smoothness should be evaluated in two scenarios. When the system is still aboard the vessel it is horizontal and the system is not bending. The other scenario is on the stinger. The system follows the curve of the stinger and has only contact on the inside of the curve. The smoothness should also be determined on this side. The more smooth the system is the higher the score.

Elements together
In order to fulfil the geometrical constraints it would be convenient if the FCS elements would be able to keep themselves together. This would mean the geometrical constraints are met in the section with the elements. If this is not the case, another solution has to be found to hold the elements together and fulfil the geometrical constraints.

Location of the neutral line
If the neutral line is in the middle and there is some room for the sling system to settle, the sling will not have to elongate to allow the entire system to bend. The neutral line will be in the middle if the FCS elements are able to hinge around a point on the longitudinal axis of the element. If the elements hinge around a point that is located away from the longitudinal axis, the neutral line will move through the hinging point. This will result in elongation of the sling while bending the system. If the location of the neutral line is in the middle, a high score is given and for deviation from the middle low scores are given.

2.3.1.2 Ball joint (idea 1)
The ball joint is based on a convex and a concave surface in contact with each other and is able to rotate in all three directions. This element will hinge around the centre of the concave and convex surface, reference is made to the yellow dots in Figure 2.6. This way the neutral line of the system will be in the middle. When the system bends further than the design value the elements will start to rotate around the contact at the edge of the elements, therefore the neutral line will start to move to the edge of the

elements. This will put tension on the sling and therefore prevent the sling system from bending any further. An assembly of these elements can be found in Figure 2.6. This idea can also be made in an alternating sequence. That will result in balls and rings with concave inner surfaces. The score of this idea can be found in Table 2.2.

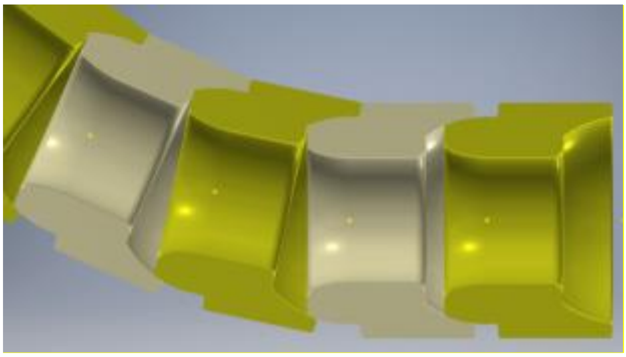


Figure 2.6 Ball joint elements

Table 2.2 Scoring of the ball joint

Criteria	Score
Load only in compression	5
Force path	4
Smoothness	5
Elements held together	1
Location of the neutral line	5
Total	**20**

2.3.1.3 Bend restrictor (idea 2)

This idea is based on the bend restrictors that are already on the market. The elements allow the sling system to bend a certain degree and prevent the sling system from bending any further. This can be worked out in two different ways, the first is the conventional way and can be seen in Figure 2.7. Another option is to make two different parts and put these in an alternating sequence, this can be seen in Figure 2.8. The scoring of the idea can be found in
Table 2.3.

Table 2.3 Scoring of the bend restrictor

Criteria	Score
Load only in compression	1
Force path	2
Smoothness	1
Elements held together	5
Location of the neutral line	5
Total	**14**

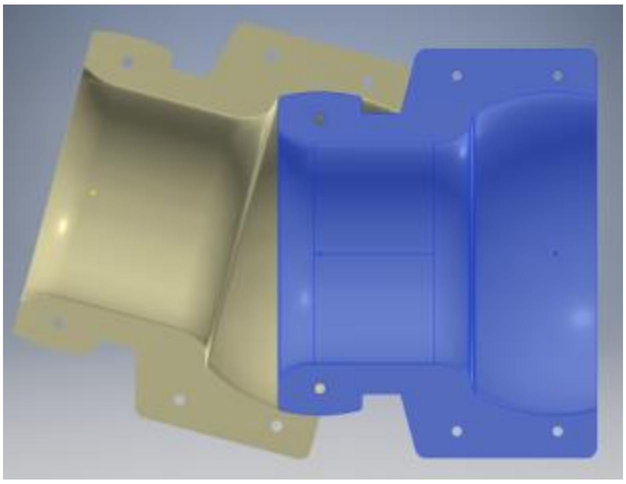


Figure 2.7 Conventional bend restrictor

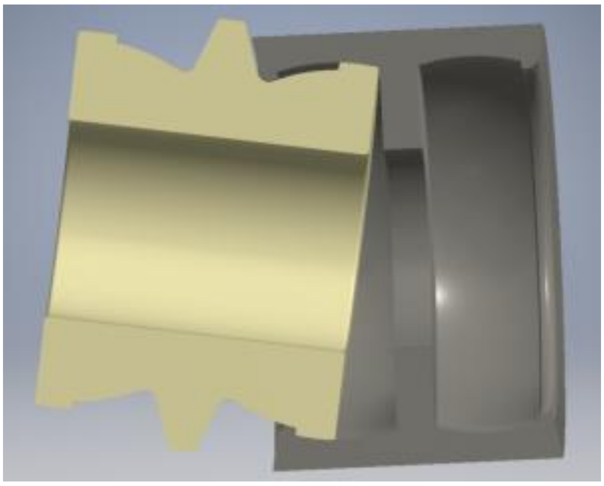


Figure 2.8 Alternating bend restrictor element

2.3.1.4 Spacer elements (idea 3)

The spacer elements are simple spacers that increase the diameter of the sling bundle to the diameter of the rest of the equipment. After modelling these elements in two types, see Figure 2.9 and
Figure 2.10, the conclusion was that the neutral line is not in the middle of the elements. The elements hinge only around the contact points at the edge of the elements. This requires the sling to elongate before allowing the sling system to bend. Since the sling will be made for much larger
loads than occur on the seabed it is assumed that the sling will not have enough elongation to allow bending with this idea. The scoring of the spacer elements can be found in Table 2.4.

Table 2.4 Scoring of the spacer elements

Criteria	Score
Load only in compression	5
Force path	5
Smoothness	4
Elements held together	2
Location of the neutral line	1
Total	**17**

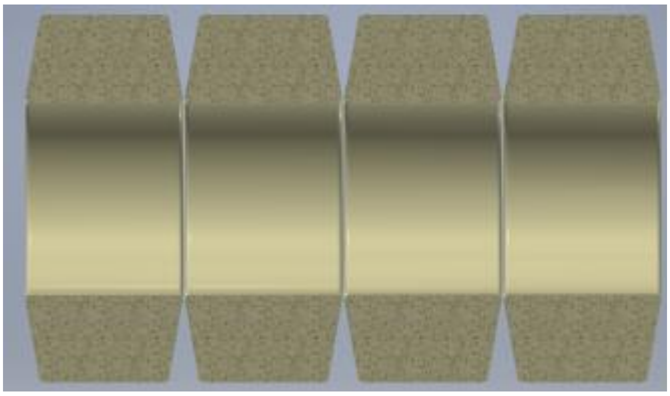


Figure 2.9 Spacer elements

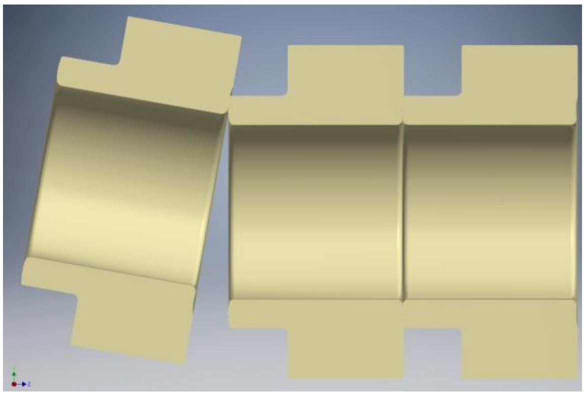


Figure 2.10 Spacer element

2.3.1.5 Selection

All selection criteria are weighted in order to give the most important criterion the most influence. The weights used are 1, 3 and 5, this will result in larger differences between important and unimportant criteria. The comparison can be found in Table 2.5, a summation is made with and without weight factors. The results are quite clear, with and without the weight factors, the ball joint idea is the most feasible and will be used as the basis for the rest of the design.

Table 2.5 Comparison of the scores with weight factors

Criteria	Weight factor	1 Ball joint	2 Bend restrictor	3 Spacer elements
Load only in compression	5	5	1	5
Force path	3	4	2	5
Smoothness	3	5	1	4
Elements held together	1	1	5	1
Location of the neutral line	5	5	5	1
Sum	-	20	14	17
Weighted sum	-	78	44	58

2.3.2 Strength of the FCS elements

In order to provide the FCS elements with sufficient strength and stiffness for stinger passage some options are proposed.

- An open core with a large wall thickness
- A three spoked core for a better force path
- A six spoked core for a better force path

The spokes can serve multiple options. For one they can add strength by providing a better force path through the element. Next to that they can be seen as divisions between the individual sling legs (6 spokes) or divisions between sling leg pairs (3 spokes). These options are visualized in Figure 2.11. In order to make a decent decision about the core, a finite element analysis (FEA) has been performed. Next to that the torsional stiffness and the force path will be discussed. The analysis has been performed with a model based on the ball joint element as can be seen in Figure 2.6. The analysis is described in full depth in memo [7]. The results are repeated here in order to get an insight in the stresses and displacements in the elements.

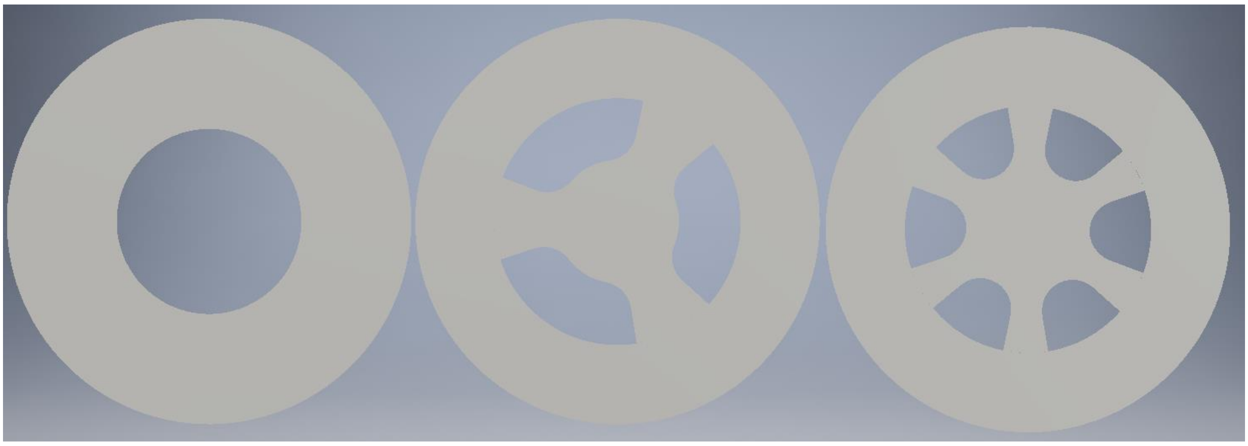


Figure 2.11 Overview of the core options

In this analysis two different models have been analysed, the open core with a larger wall thickness and the three spoked option with a smaller wall thickness, the first and second image in Figure 2.11. The three spoked model has been analysed in two orientations, one with a spoke vertical on the bottom and one which has been rotated 60 degrees, thus with one spoke vertical on the top. The assumption has been made that one of these two orientations is the worst case and all orientations in between have stress levels in between. Since the open core is fully axisymmetric this simulation has not been run in multiple orientations. Two load cases have been tested, the load is applied on the inside of the element normal to the surface and in vertical direction.

Only the worst case load cases, and the highest occurring stresses and displacements are presented in this document. The deformation shapes of these three simulations are presented in Figure 2.12, Figure 2.13 and Figure 2.14. As can be seen in Table 2.6, the order of magnitude of both the stresses and displacements is the same and the absolute differences are small. This leads to the conclusion that a core compared to thicker wall thickness does not have a big influence on the stress levels. Both options are well below the yield and do not form any problems.

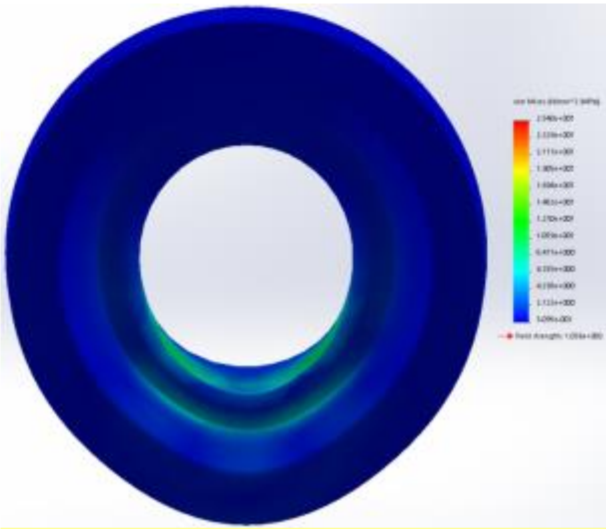


Figure 2.12 VM stress open core vertical load, deformation scale 100

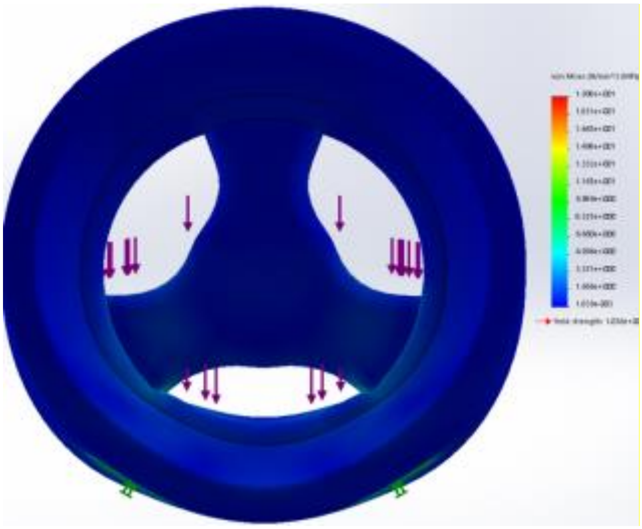


Figure 2.13 VM stress spoked core 0° vertical load, deformation scale 100

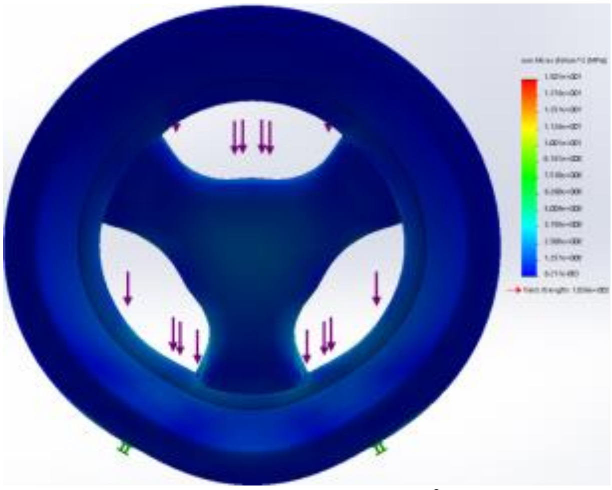


Figure 2.14 VM stress spoked core 60° vertical load, deformation scale 100

Table 2.6 Comparison of simulation results, vertical load case

	Open core	Three Spokes 0°	Three Spokes 60°
Max stress (Von Mises)	25.40 MPa	20.16 MPa	15.01 MPa
Max displacement	1.04 mm	0.99 mm	0.73 mm
Max displacement x direction (width)	0.16 mm	0.25 mm	0.31 mm
Max displacement y direction (height)	0.12 mm	0.03 mm	0.01 mm
Max displacement z direction (length)	0.95 mm	0.03 mm	0.15 mm

Another consideration is the requirement, as stated in [3], is that the FCS shall have a low torsional stiffness and it should be able to allow three full revolutions of twist. Twist in the slings will result in a shorter effective distance between the hooks and thus compression of the FCS elements. Inserting a spoked wheel will push the slings in outward direction and will increase this shortening effect. This will lead to an increase in torsional stiffness. Next to that it will disrupt the alignment of the spokes and therefore increase the possibility for the slings to get damaged during operation.

The force path from the roller contact to the inside of the element is also considered in this section, since this will have influence on the stress distribution in the elements. This element is designed in such a way that the force path is always as direct as possible. As can be seen in Figure 2.15 The sling will have contact with the FCS elements at the blue lines. This will results in the red force paths, most are direct through just one body. The same design can be seen in Figure 2.16 there the contact surface is larger and the force paths are all straight and the majority goes through one body. Both Figure 2.15 and Figure 2.16 have an increasing inner diameter, this is done to realise the direct force paths. In Figure 2.17 The same design is shown but this time without an increasing inner diameter. This will result in force paths that will go through multiple bodies in the majority of the cases. In order to have a direct force path combined with a spoked core, the diameter of the centre of the core should be decreasing. This will most definitely lead to a weaker core. In the FEA, as described earlier in this section, this decreasing core diameter has not been modelled.

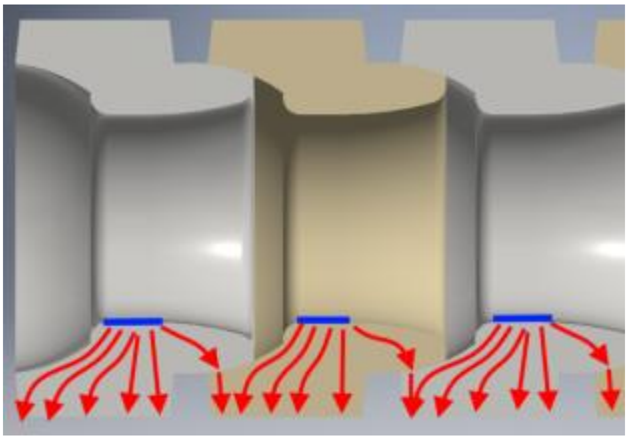


Figure 2.15 Force path in a straight state

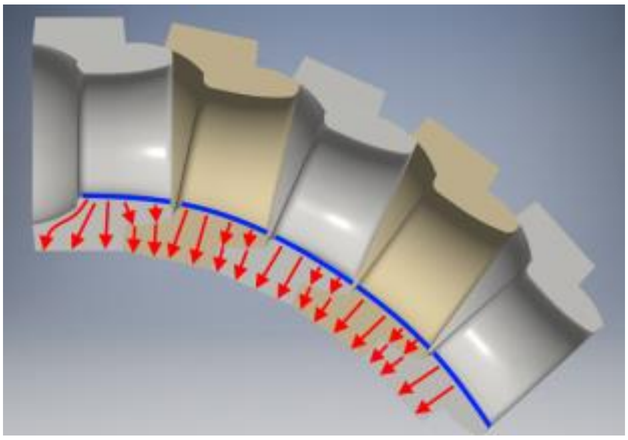


Figure 2.16 Force path in a bended state

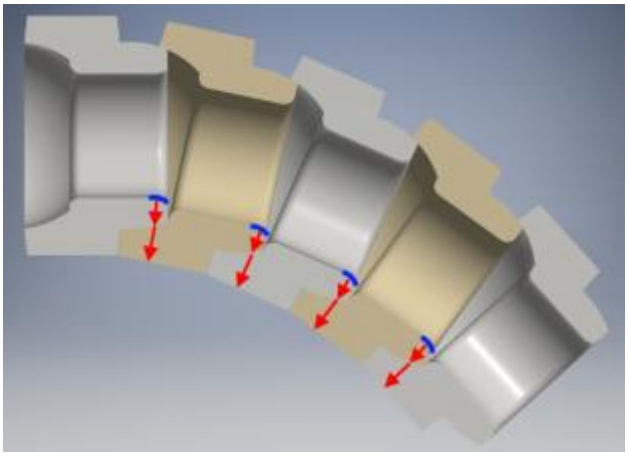


Figure 2.17 Force path in a bended state, without diameter variation

Since the FEA does not give a large difference between the two options. Other considerations will have to determine the basis for this decision. Both the torsional stiffness and the force path give the advantage to the open core. The open core will also lead to easier assembly and less difficulties in the design. Therefore this is the most feasible option.

2.3.3 Handling and installation

The size of the elements, makes them heavy and hard to handle. Based on the density of nylon, they will be over 400 kg per element. This will require overhead cranes for handling and complicates the assembly process. In order to aid the assembly and improve the handling of the elements it is investigated if the FCS elements can be made out of two parts. These two parts, half shells, should together form one FCS element, this idea is visualized in Figure 2.18. In order to determine if this is feasible another FEA is performed.

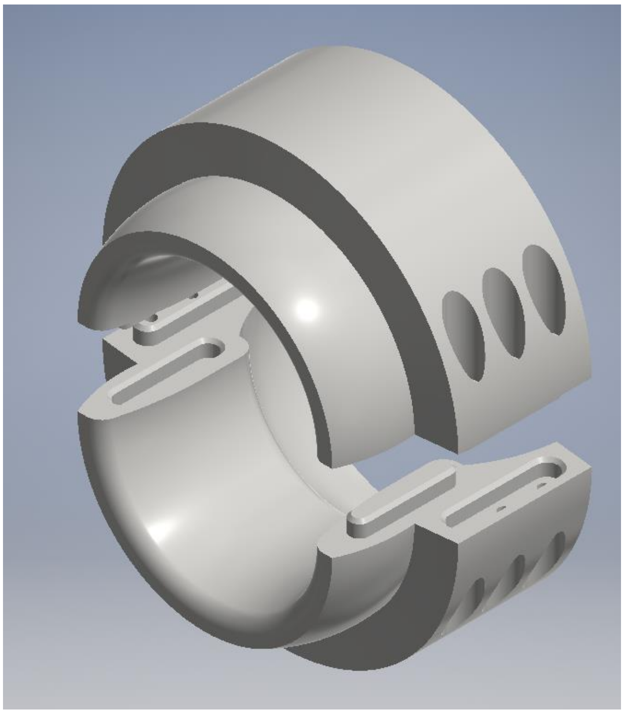


Figure 2.18 FCS element made from two half shells with a bolted connection

This FEA is fully described in [8] and the results of this analysis are repeated in this document. The occurring stresses in the analysis are summarized in Table 2.7 and the estimation of the biggest gap in Table 2.8. The hoop force that is present in this simulation is based on the calculations mentioned in [9]. Since the hook diameter is larger than the inner diameter of the FCS elements, the slings have to be bundled under tension, as can be seen in Figure 2.19. The force necessary to bundle these slings is calculated and is a function of the deviating distance of the sling leg and the distance between the hook and the bundling element.

Figure 2.19 Bundling the slings

The hoop force in this simulation is based on a deviation of 200 mm at a distance of 1 m. This results in a hoop force of 20% of the tension in the system, thus 400 t around the circumference. Next to that a force of 2%, thus 40 t, is necessary in order to prevent longitudinal translations of the bundling element. This bundling force will only be present in two FCS elements, the last elements on each side. Which have a smaller diameter compared to the diameter of the bundle coming from the hook.

Table 2.7 Maximum occurring VM stress

	Split line vertical	**Split line horizontal**	**Single shell**
Load normal	52.69 MPa	59.02 MPa	20.78 MPa
Load vertical	69.19 MPa	59.02 MPa	25.40 MPa
Load normal + hoop force	172.06 MPa		37.83 MPa
Load vertical + hoop force	170.44 MPa		40.00 MPa

Table 2.8 Estimation of largest gap at split of convex area

	Split line vertical	Split line horizontal	Single shell
Load normal	0.95 mm	0 mm	Nvt
Load vertical	0.4 mm	0 mm	Nvt
Load normal + hoop force	15.5 mm		Nvt
Load vertical + hoop force	13.5 mm		Nvt

The conclusion that can be drawn from this FEA is that an element that consists of two half shells is able to withstand the roller loads but is, in this form, not able to withstand the bundling force.

2.4 Design of the end pieces

The most relevant requirements for the end pieces are repeated here:

- Cover the hooks to prevent the slings from coming off
- Comply with all geometrical requirements as stated in Section 2.2.2
- Adjust for the elongation of the sling 0.7% (according to Lift-tex) at WLL (84 mm based on 12 m working length)

2.4.1 Options

In order to fulfil the geometrical requirements the FCS elements are not allowed to move away from each other. This will results in gaps between the parts that will be larger than allowed. To achieve this, there are two options. Exact manufacturing and assembly in combination with small possible gaps in the design to account for assembly and production tolerances. This will make the assembly and production more expensive and complicated. Next to that it will have a high risk of not fulfilling the requirements. This option is not robust. Another option is to design the end pieces in such a way that they will apply a force on the FCS elements and compress all FCS elements together. This option requires a clever design but is not as vulnerable to assembling mistakes and therefore this solution is more robust and thus the most favourable.

2.4.2 Working principle

By bundling the sling with a ring with a smaller diameter than the bundle coming from the hook, the resulting forces will push the ring towards the middle of the system. This principle is best visualized by a toy for children, as can be seen in Figure 2.20. The distance between the boys' hands represents the hook diameter and since this is larger than the inner diameter of the ball, the ball will be forced in the direction of the other person. In the FCS this will lead to the two outer FCS element applying a compressive force on the all the FCS elements in the middle. This will keep the FCS elements together.

Figure 2.20 Working principle is also applied in toys for children.

2.4.3 Design

The design of the end pieces, as can be seen in Figure 2.21, is based on the FCS element in order to have a good interaction between the two. Pockets have been added in order to transfer the shear stresses from one half to the other, this can be seen in Figure 2.18 and Figure 2.22. As already discussed in section 2.3.3 a bundling element that consist of two half shells will result in issues. Since the end piece should be the bundling element for the principle to work a solution has to be found. One solution will be to insert a steel ring in order to cope with the large hoop force. This steel ring can be seen in Figure 2.22. The end pieces are made quite long in order to be able to move over the hook and the receiver or A&R head. This way they will prevent the slings from falling of the hooks without tension. As can be seen in Figure 2.22 the end pieces form the bundling parts and the FCS elements in the middle will be compressed in the middle as explained in section 2.4.2. Since the outer diameter of the receiver and A&R head is smaller than 1150 mm this diameter will have to be increased. Covers are made to increase the diameter, if these will touch the end pieces than both end pieces will have 191 mm of space to slide without creating a gap larger than allowable. Thus the system will be able to adjust for the elongation (84 mm) of the sling easily.

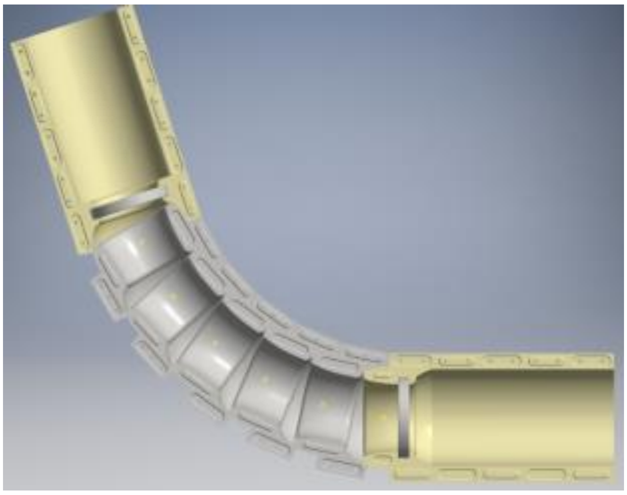


Figure 2.21 Overview of the system bending, majority of the elements are left out for clarity

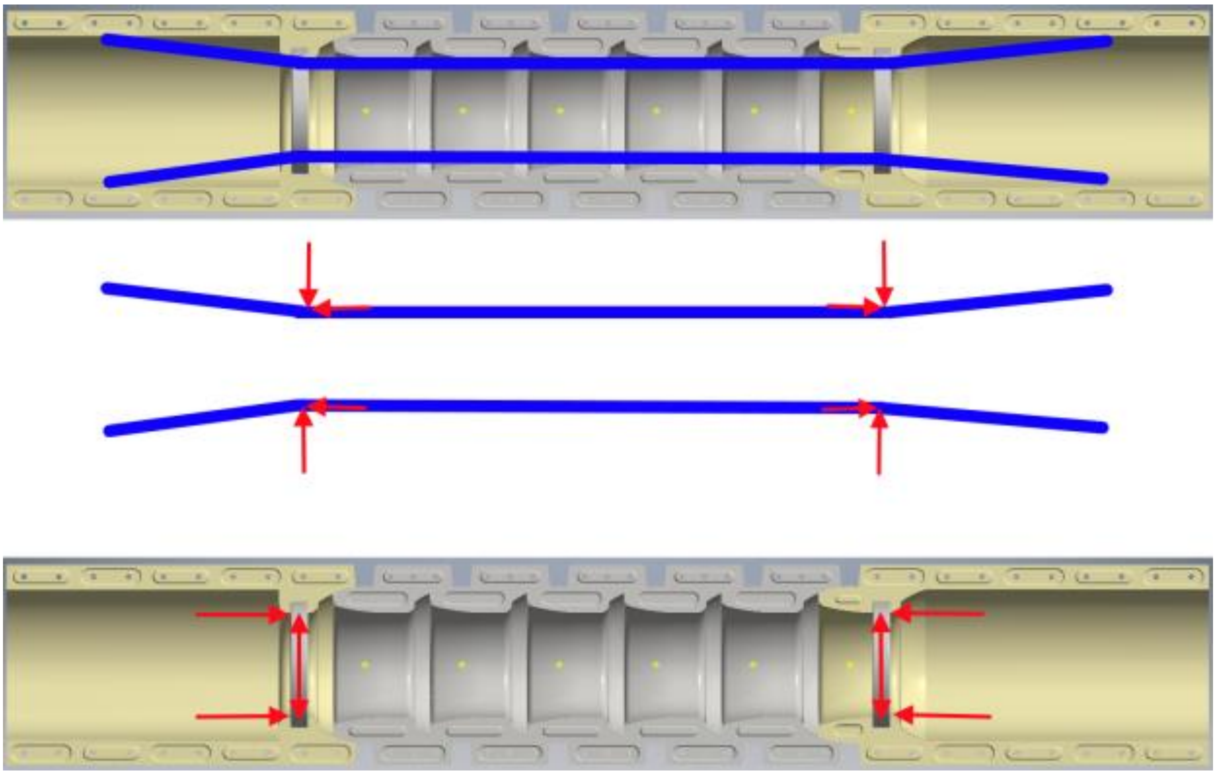


Figure 2.22 On top, system with schematic view of slings (blue). Middle the forces (red) from the system on the sling. Bottom forces from the sling on the system. Majority of the elements are left out for clarity.

2.5 Assembly

Based on the current design, the assembly procedure would be relatively simple. First the bottom half of all the elements should be prepared. Al the half shells should be put in a row, including the end pieces. Once this has been done the sling should be reeved in, including the rings that fit in the end pieces, see Figure 2.21. Once this has been done, the slings, rings and hooks can be put in the bottom of the half shells. From this moment on piece by piece all the top side half shells should be bolted on their counterparts. After this only the gaps at the end of the end pieces should be filled. For this purpose some covers (see Figure 2.23) are made to increase the diameter of the receiver and the A&R head to the outer diameter of the system. These rings should be cut to size when the FCS is fully assembled and at a very low (<1% of WLL) tension. This way the sliding end pieces have no gap and the adjustment for the elongation can be two times 191 mm. Which is more than the required 84 mm of adjustment. An adjusted configuration can be found in Figure 2.24.

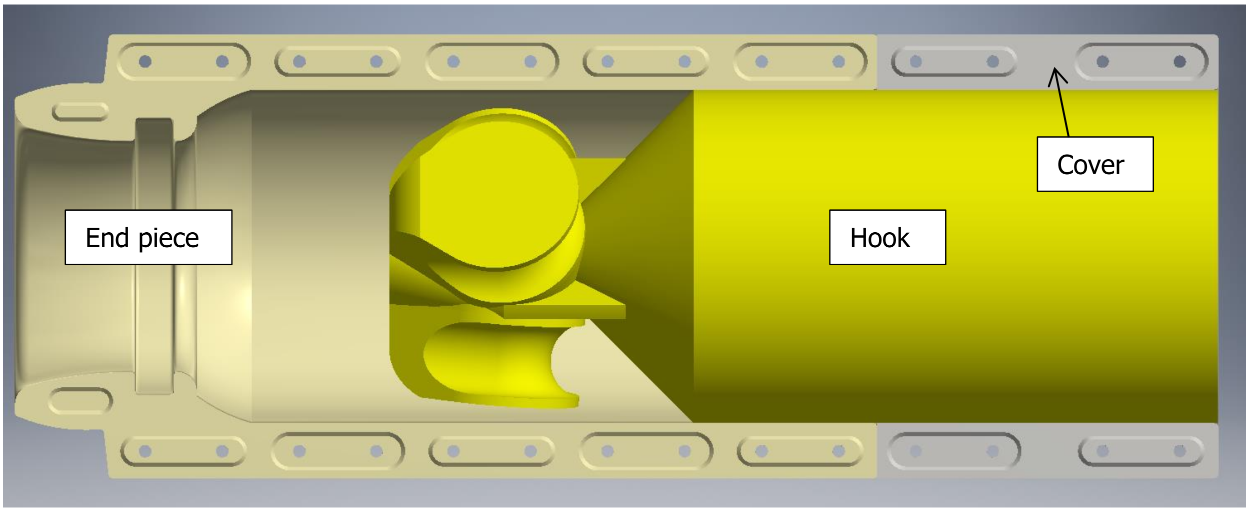


Figure 2.23 End piece, hook and cover. As assembled, no gap between cover and end piece, because the cover is made to size at location.

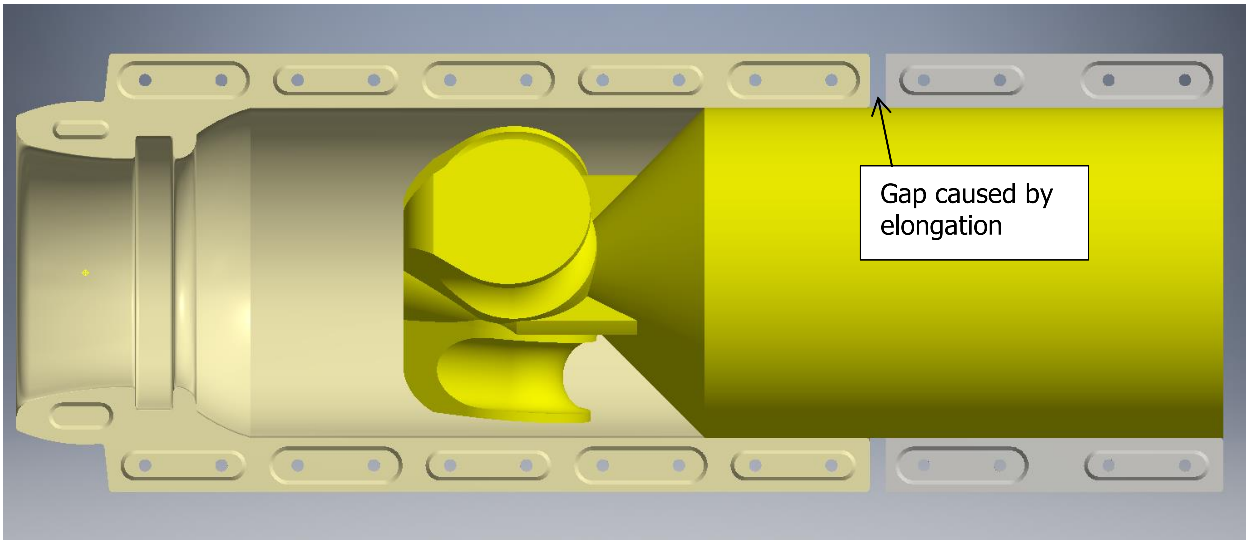


Figure 2.24 End piece, hook and cover. Adjusted for the elongation.

2.6 Material selection

2.6.1 Material and production method

Since the FCS elements have such large volumes the density and the price should be low. The FCS will have to be held vertical by a buoy this is an extra reason to keep the density of the FCS elements as low as possible. Stinger passage requires a strong material thus the yield stress should be high. These properties are the most important material properties for the FCS element material. Looking at production the material will have to be able to be casted or be available in stock of the right dimensions. In both cases machining is necessary in order to get the required tolerances. A comparison is made in between some materials with high yield strength, low density, low price and that are able to be casted.

Table 2.9 Comparison between possible materials, values based on [10]

Material	Density	Price	Yield	Castability
Nylon	1300 kg/m^3	3.20 EUR/kg	72.4 MPa	Possible
Polyurethane	1180 kg/m^3	3.25 EUR/kg	46 MPa	Good
POM	1410 kg/m^3	2.30 EUR/kg	60 MPa	Possible
PTFE	2170 kg/m^3	12.00 EUR/kg	20 MPa	Possible

This data points to the conclusion that nylon and POM are the only suitable materials when looking at the yield stress. Since there is already experience with the use of nylon with an even lower density and higher yield stress, this will be the preferred option. The material Oilon from Nylacast has been used in the past and has a low density of 1138 kg/m^3 [11]. In order to achieve the required tolerances the parts need to be cast an machined afterwards.

2.6.2 Suppliers

A search for suppliers of large cast nylon products has been performed. This resulted in only a short list of possible suppliers around the world. The list of possible suppliers can be found in Table 2.10. For reasons of communication and possible visits to the suppliers, European suppliers are favourable.

Table 2.10 Summary of possible suppliers

Supplier	Main office	Notes
Nylacast	Leicester (UK)	Known supplier, max shot mass 750 kg
Quadrant EPP	Tielt (Belgium)	Office/production in Almelo
Ensinger GmbH	Nufringen (Germany)	Max shot 900 kg
Tekmar	Newton Aycliffe (UK)	Makes cable protectors comparable with desired product. Uses Polyurethane. Not strong enough.
Cast Nylons Ltd	Willoughby (US)	Far away, less favourable for visits
Cast Nylon India	New Delhi (India)	Far away, less favourable for visits
Dotmar	Dingley (Australia)	Far away, less favourable for visits
Castor Plastics GmbH	Pfalzfeld (Germany)	Machining up to 500x500x750 too small

2.7 Final design

The final design will require more detailed and more comprehensive FEA simulations. These simulations will inevitably lead to alterations of the current design. Next to that, all tolerances need to be determined. The design should be verified by the supplier for reasons of manufacturability. And lifting points should be added for handling.

3.0 DAMAGE INSPECTION OF SLINGBUOY 2.0

3.1 General

3.1.1 Document scope

The focus of the report is on the failure inspection after the tests aboard the *Lorelay* and the following validations and recommendations of the *Slingbuoy 2.0* project.

3.1.2 Introduction to the *Slingbuoy*

In the event of upcoming bad weather or a mechanical breakdown, the pipeline is abandoned using the A&R winch. A sacrificial wire rope is in between the pipeline and the A&R cable of the vessel, which is cut in the event the vessel needs to leave the field.

A second wire rope with a buoyancy module is required in order to recover the pipeline and continue pipelay. This buoyancy module ensures that the sling is upright to greatly facilitate the hook-in operation. However, these modules can be very large and do not travel easily over the stinger rollerboxes. Upon recovery these are therefore removed by ROV which complicates the operation and makes it sensitive to weather and ROV breakdowns.

To make these operations more efficient, the subject of development has been a buoy that can travel over the stinger and can be re-used, making the operation more robust and cost efficient.

The first *Slingbuoy* design was based on rigid PP segments connected to the sling in the centre. The choice of material (with still a relatively high density of 980 kg/m^3) resulted in a large and heavy buoy which was difficult to handle on board.

Slingbuoy 2.0 concept revolves around a buoyancy gel rather than rigid buoyancy modules. Additionally, the buoyancy gel allows for a much lower density of 500 - 600 kg/m^3. The gel is stored in flexible hoses, buoyancy hoses, with a filling level of 70 - 80 %. This allows the gel to be displaced during the loading on the rollerboxes. Since the material does not take up rollerbox loads, a lower density can be achieved, which leads to a more lightweight concept, ultimately making the *Slingbuoy* easier to handle in the firing line. The total weight of the *Slingbuoy 2.0* prototype is approximately 1500 kg. The resulting submerged weight of the sling is around -160 kg, i.e. positively buoyant.

The current design is for a steel sling (WLL 165 t SWRP sling) and requires a number of buoyancy hoses to achieve the required buoyancy. The hoses are connected the sling by two nylon (PA) connection cones at both ends. The connection cones are clamped around a dedicated steel ferrule which is crimped on the main leg of the sling, some distance behind the normal ferrule required for cable integrity. Figure 3.1 presents a general arrangement of the *Slingbuoy*.

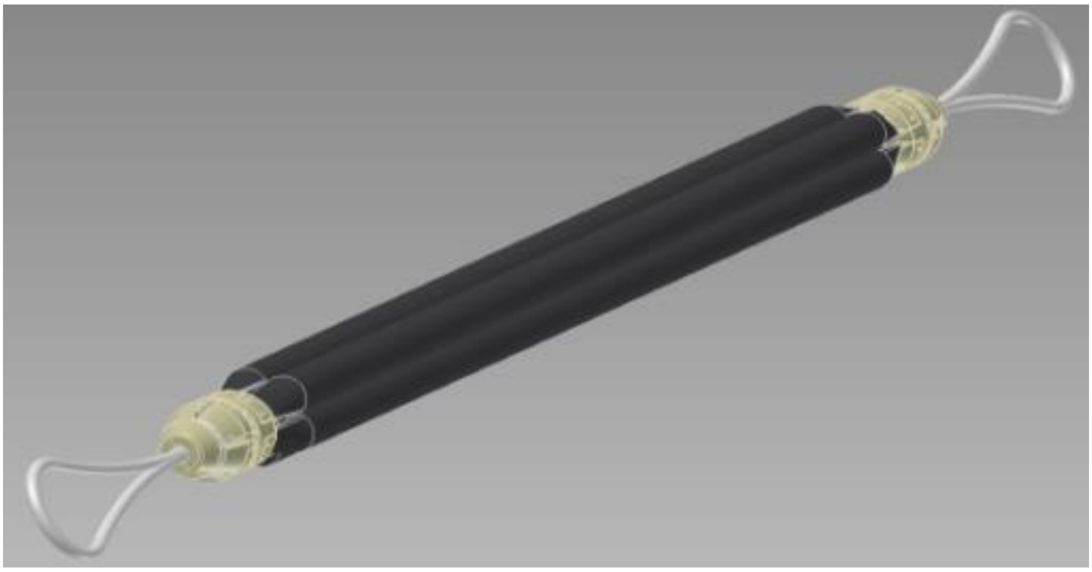


Figure 3.1 *Slingbuoy* general arrangement, without cover

3.1.3 Abbreviations

The following abbreviations are applicable for this particular document:

A&R	- Abandonment and recovery
WLL	- Work Load Limit
PA	- Polyamide
RB	- Rollerbox
ROV	- Remotely operated vehicle
PU	- Polyurethane
PP	- Polypropylene

3.2 Design and test procedure

The design and the test procedure are discussed very briefly because the basis of the design and the full scale test procedure are already described in [12] and [13].

3.2.1 Design

The main design of the *Slingbuoy* consists of a wire rope, two connection cones, five buoyancy hoses and a cover. Reference is made to Figure 3.2. The wire rope has a small eye (left side of Figure 3.2) and a large eye (right side of Figure 3.2). The connection cones function as an attachment point on the sling for the other components. The buoyancy hoses are attached to the cones and covered with a slotplate in order to maintain a smooth surface of the cone. All components are protected and held together by the cover. A note is made that the inlay plates are shown in Figure 3.2, are not in the finished prototype. This decision was made due to problems with manufacturing and the new insight that the inlay plates were not required for proper functioning of the *Slingbuoy.*

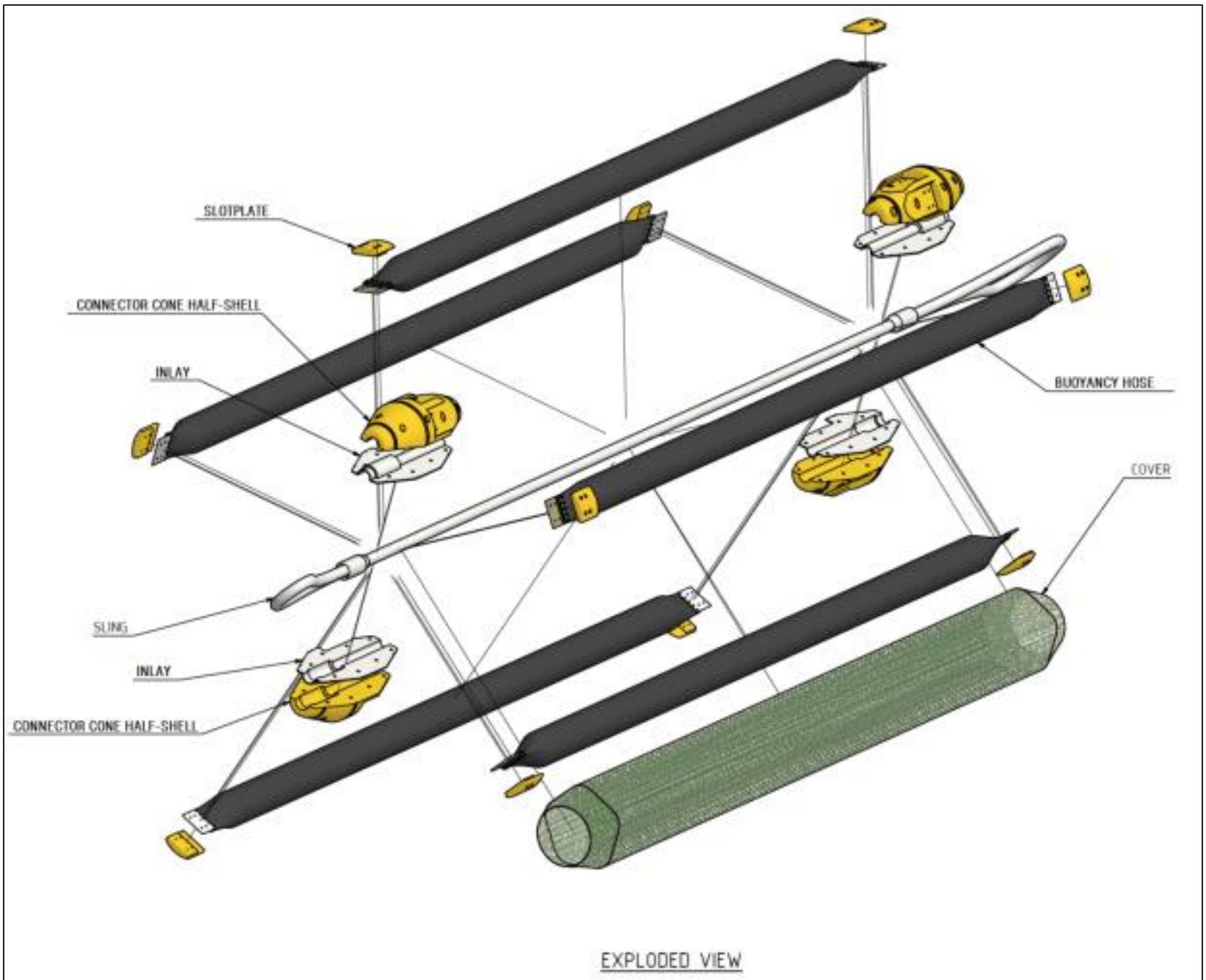


Figure 3.2 Exploded view of the *Slingbuoy*

3.2.2 Test procedure

According to the calculations in [14] the largest loads occur at rollerbox 6 (RB6). The stinger is designed to have high rollerbox loads at RB6, this is done by setting the height of the RB6 at a high setting. Therefore the load test is performed at this rollerbox. This is also conveniently located aboard of the *Lorelay.* Once on RB6 the *Slingbuoy* is pulled 12 m forwards and 12 m backwards. After each three sets, of back and forwards, the tension is increased. The following tensions are used: 25 t, 50 t and 85 t.

3.3 failure

At the first run with 85 t tension the *Slingbuoy* started leaking gel. At that moment the test was terminated and the *Slingbuoy* was prepared for storage. In this section the location and time of failure are determined. And based on that the failure method is deducted.

3.3.1 Location of the failure

Upon inspection two of the five buoyancy hoses were damaged, both on the side of the large eye around the end of the connection cone. At that location, the bags are still folded. Both tears are perpendicular to the length of the hoses and approximately 5- 10 cm long. One hose failed with a cut in two plies of the hose. Reference is made to Figure 3.3. In order to keep a good overview, from now on this hose is referred to as hose one. The second hose also has a tear around the edge of the cone but also some other damage. Reference is made to Figure 3.4. This hose is referred to as hose two.

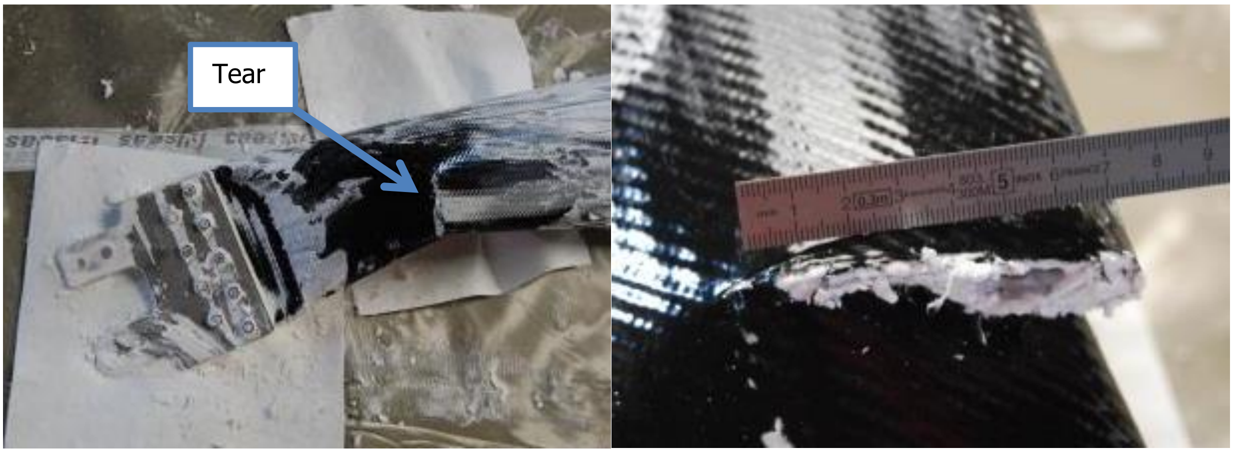


Figure 3.3 Left, overview of the tear in buoyancy hose one. Right, details of the tear in buoyancy hose one

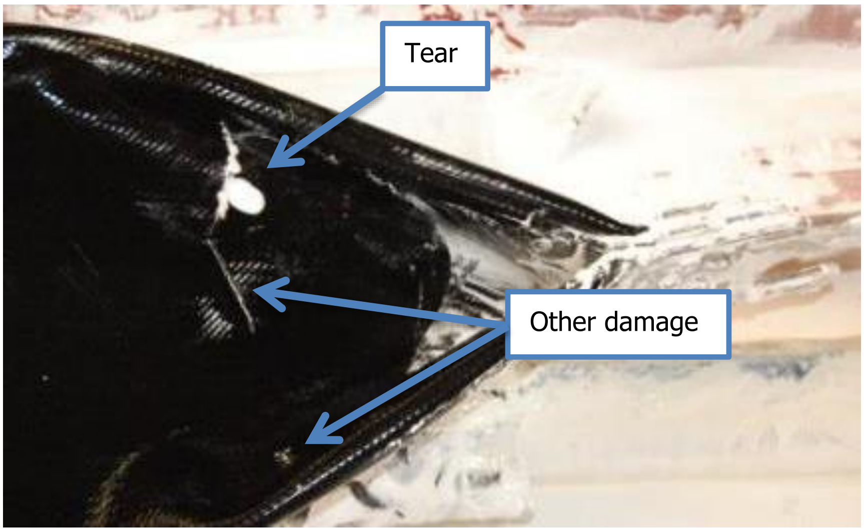


Figure 3.4 Overview of the tear and other damage in buoyancy hose two

3.3.2 Moment of failure

Based on the location of the failure and a video of the test [15], the failure is most likely to have occurred during the last 50 t tension run but was noticed on the first 85 t tension run. This is concluded because the part where the bags failed was not yet in contact with the rollerbox at the moment the leak was noticed. Reference is made to Figure 3.5. Furthermore the assumption is made that the hoses failed at the moment the cone left the rollerbox, reference is made to Figure 3.6. This is the moment that the side load of the roller on the bag is assumed to be the highest.

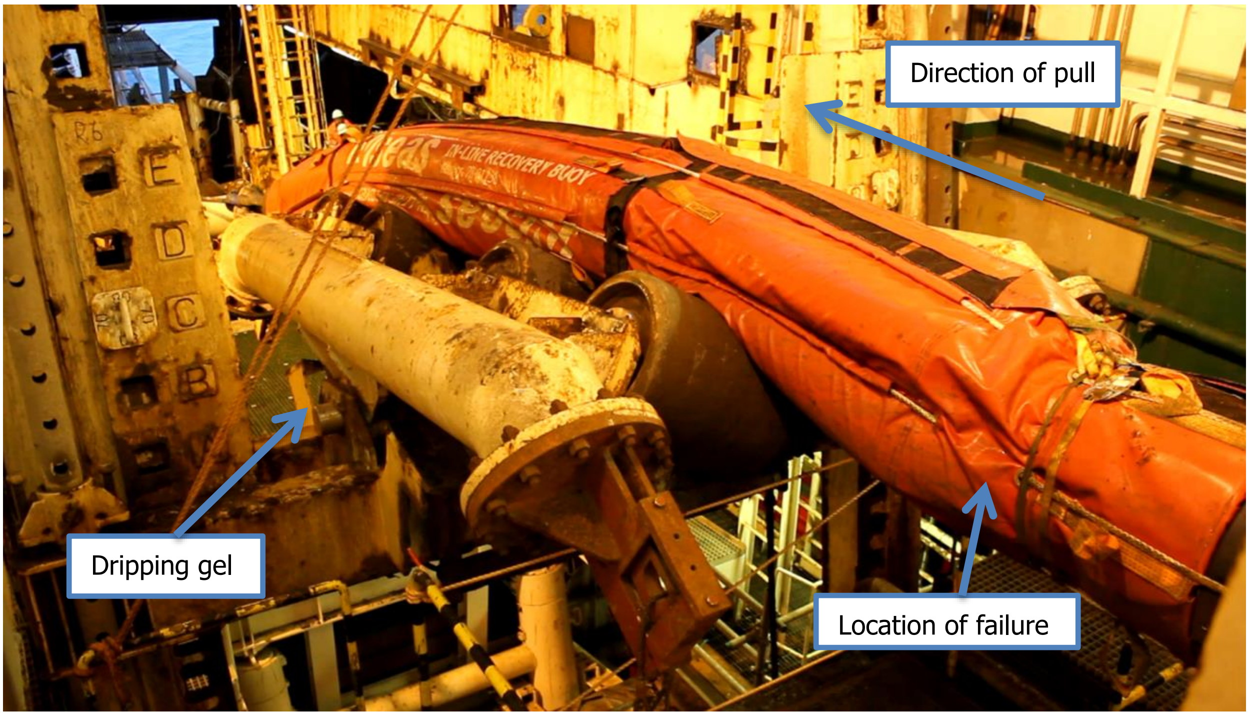


Figure 3.5 Still image of the moment of leak detection, first run with 85 t tension

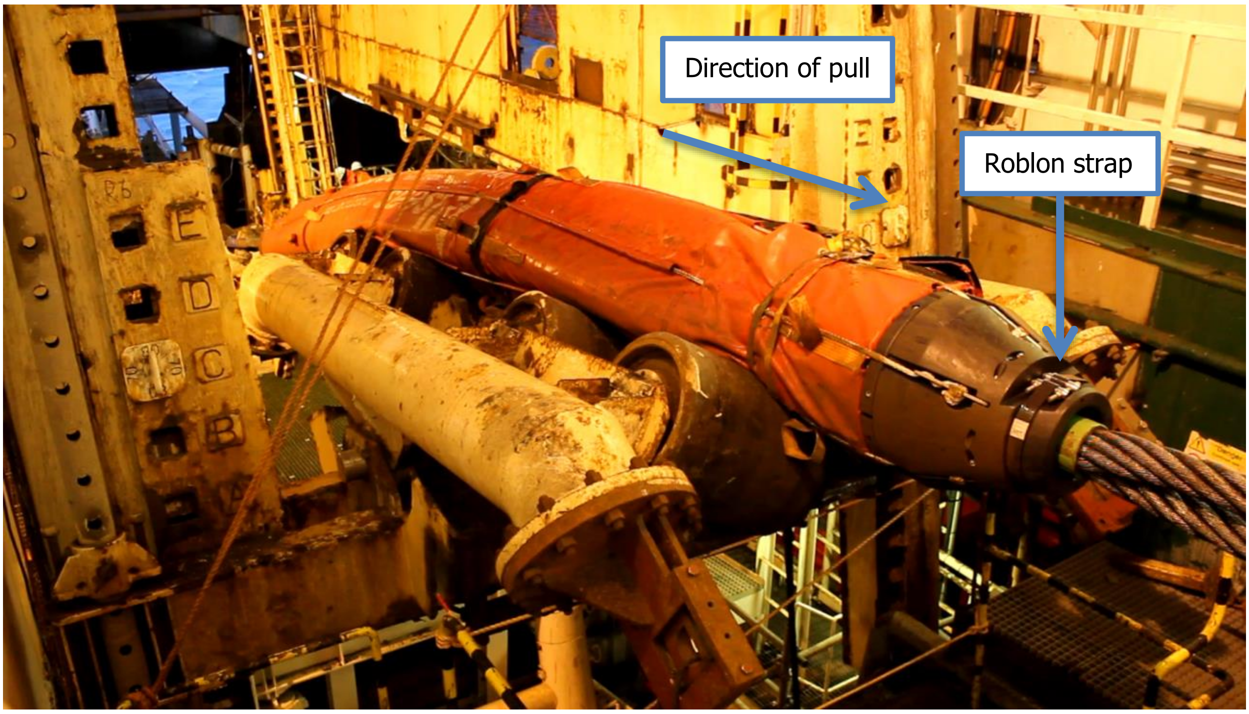


Figure 3.6 Still image of the most likely moment of failure, last run with 50 t tension, cone on the edge of the roller

3.3.3 Hose orientation

Based on the video, the location and the assumed moment of failure, a sketch is made. This sketch shows the orientation of the bags inside the cover. Reference is made to Figure 3.7. The sketched orientation is based on the drawings and the orientation of the roblon strap on the connector cone. Reference is made to Figure 3.6. Hose one has the open side of the fold, see Figure 3.8, towards the roller. The angle between hose one and the roller is 48°, based on the observation that the connection of the roblon strap is in horizontal position at the moment of failure, as can be seen in Figure 3.6. Hose two has an angle of 24° with the roller, and has the closed side of the fold facing the roller. These angles are visible in Figure 3.7.

The sketch based on the video and the location of the failed hoses on the cone correspond with each other. This is an indication that the sketch is a valid approximation of the real load situation.

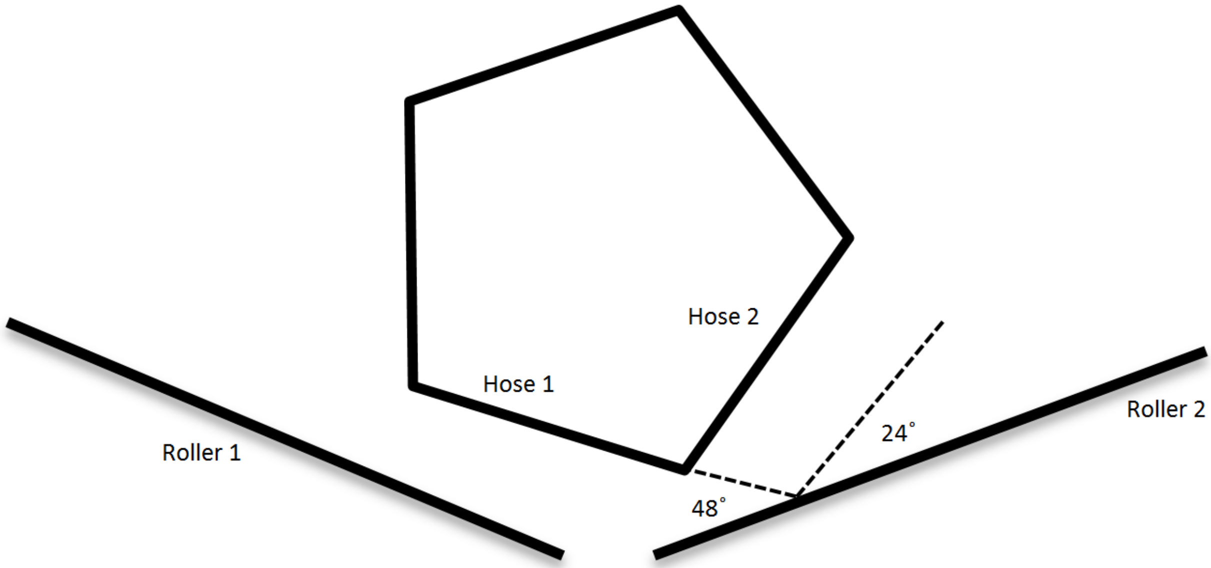


Figure 3.7 Hose orientation on the rollers

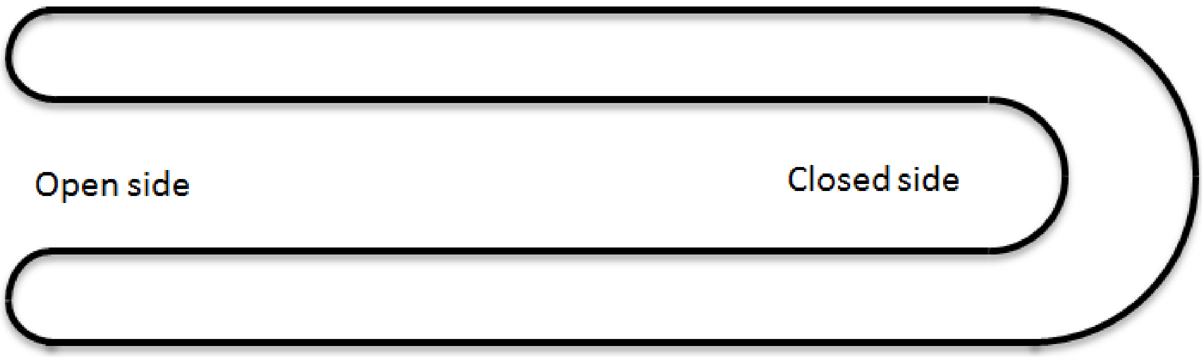


Figure 3.8 Folding of the hose, cross section

3.3.4 Data of the hose and connection cone

Both failures occurred at the end of the cone. This seems to be the point with the highest loading conditions on the hose. Therefore the focus of the report is on the loading conditions at this location. In this section, available information about the hose and the connection cone is presented.

The limited information that is known about the hoses is described here. The hose is made of PU, reinforced with high tenacity polyester yarn in 0/90 degrees. The directions are observed from a sample. Reference is made to Figure 3.9. The hose is a Hilcoflex PU hose made by Gollmer & Hummel and purchased at Mees van den Brink. The properties of the hose are summarised in Table 3.1.

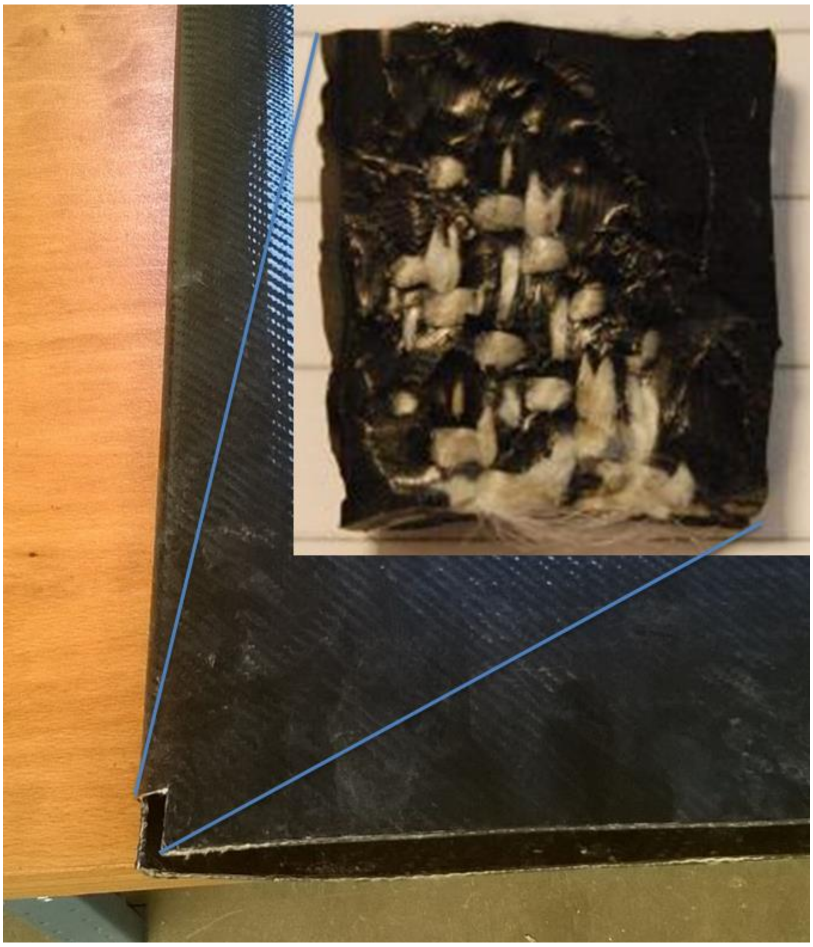


Figure 3.9 Hose reinforcement direction

Table 3.1 Information about the hose provided by Gollmer & Hummel, the manufacturer

Property	**Value**
Inner diameter	254 mm
Outer diameter	260.4 mm
Weight	2850 gr/m
Wall thickness	3.2 mm
Breaking strength	23.5 t

The edge of the cone has a radius of 5 mm in order to prevent damage to the bags. However, geometrically the difference in distance between the conus and the wire rope is relatively big. As can be seen in Figure 3.10, the diameter of the edge of the conus is 280 mm. With deduction of the wire rope diameter (83 mm) and division by two, the distance between the edge and the wire rope, is 98.5 mm. This distance is indicated in Figure 3.10 as the “gap”.

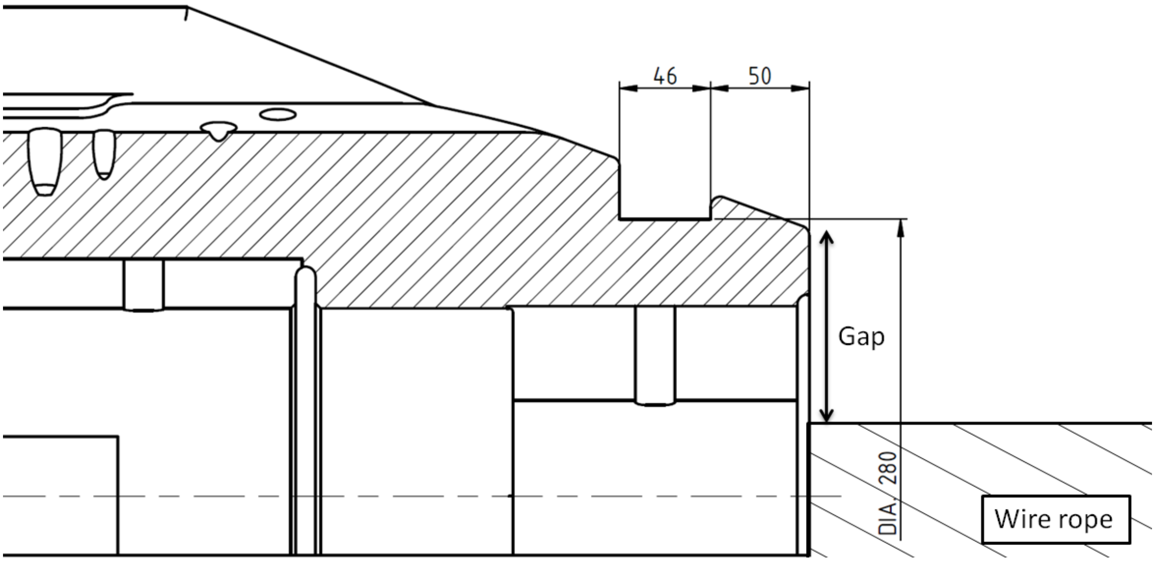


Figure 3.10 Drawing [16] of one of the conus halves, wire rope and gap added for clarity

3.3.5 Failure mechanisms

This section describes the failure analysis and the underlying assumptions. It is assumed that the orientation of the hoses and the moment of failure are as deducted earlier in this chapter.

The cut in hose one looks like the hose is cut with a scissor. A double layer is cut from the side at the location of the end of the cone. The hypothesis is that the high side load in combination with an axial load, a large transition distance and a large angle (48°) is likely to be the combined cause that the roller was able to tear of the hose.

Hose two has a different type of tear but around the same location in length direction, at the edge of the cone. The same assumptions hold for this failure but the angle between the roller and the hose attachment is much smaller as described in Section 3.3.3. As can be seen in Figure 3.11 the damage is composed of three different types of damage:

- On the left the large tear is visible, which is also the location of the main leak.
- In the middle, connected to the tear but at an angle to it, some surface damage is visible.
- On the right a puncture is visible.

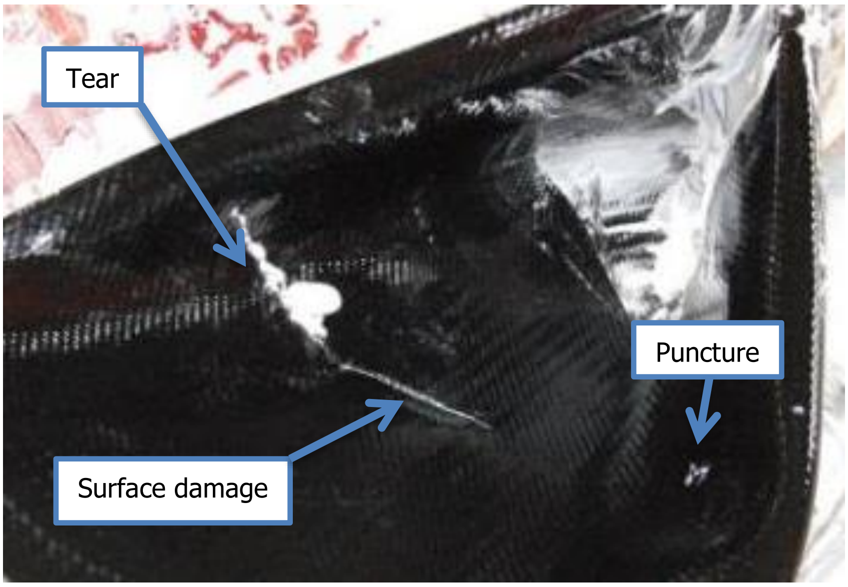


Figure 3.11 From left to right: tear, surface damage and puncture

The tear and surface damage are treated as one damage, because of the high likelihood of having the same cause. The connection between these two damages is not in the middle which would be logical since there is a fold there. The intersection of the two damages is a not in the centre of the bag. This can be explained by Figure 3.12. In this figure the hose is filled, which results in a different folding pattern compared to the that of the end clamp. If the folding was indeed as can be seen in Figure 3.12 the two damages are in the same spot but on opposing sides of the fold. This could indicate that the damage may have been done by a foreign object that was enclosed in the fold of the bag. This foreign object has not been found yet.

Tests performed, before the choice for the type of bag [12], have proven that the hose is able to withstand the indentation of bolts with a side load of 25 t. The difference between the tests and the real situation might be the presence of axial tension on the hose. This might explain that the hose failed at a lower side force (9.5 t side load) compared to the test setup where the hose did not fail at all (25 t side load).

Figure 3.12 Folds in a filled bag

The puncture seems to be in an outward direction and no damage is visible on the other side of the hose. The puncture is clearly more than superficial damage and upon further investigation it is concluded that the puncture is a leak. This was not initially noticed because of the large tear very close to the small puncture. This led to visible leaking through the tear and no visible leaking at the puncture. The damaged material seems to be pointing in an outward direction. This can indicate a puncture from an object inside the hose or it can indicate that an object from outside has punctured it and pulled some material along with it on the way out. Further research into the actual failure mechanism is required. The test setup, as described in the next section can be used to try to replicate the damage.

3.4 Test setup

A test setup is designed in order to validate if the failure mechanisms are determined correctly. For hose one the main focus was to recreate the distance between the cone and the wire rope, in combination with a side load. In order to recreate the failure, the angle between the attachment and the roller is also of importance, together with the option to mount the hose under tension and the shape of the substitute cone.

The design, shown in Figure 3.13, consists of a plate with a section of pipe welded. This pipe section functions as a replacement of the cone. Due to available pipe, the pipe has a smaller diameter, 219 mm, compared to 280 mm of the cone but the effect of this is expected to be of no significant influence. It is fitted with a 5 mm radius just as the cone is. The plate is fitted with the same thread pattern as is used in the end clamps of the hoses, for easy fitting and comparison to the real situation.

In order to be able to use the setup for both cases, a two piece design is chosen. It consists a simple block of steel which is bolted to the plate from the bottom. The top of the block has a slope corresponding with the angles of the hoses in the buoy. On this sloped surface the same thread pattern is made in order to fit the hoses with the mounting plates, used in the *Slingbuoy* design.

To prevent the plate from bending due to the tension in the hose, the base plate is bolted to the bottom of the hydraulic press. The shape of the hydraulic press has not been determined yet. This should resemble the shape of a rollerbox roller.

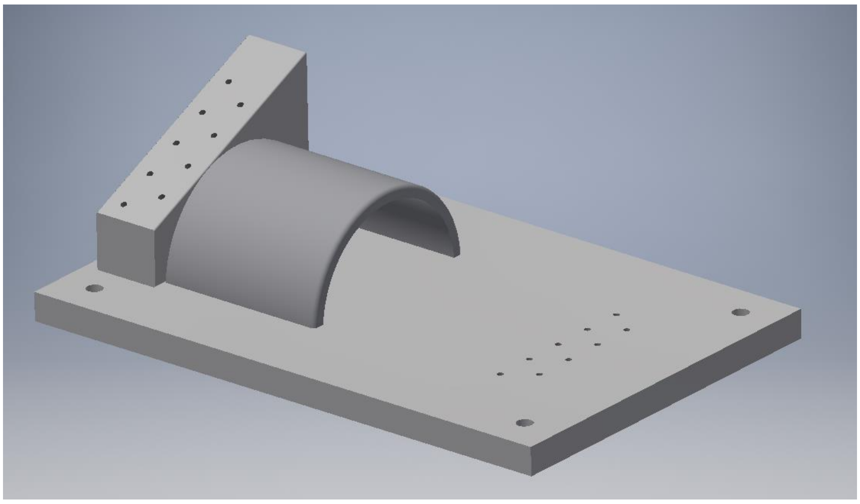


Figure 3.13 Design of the test setup for hose two. Connector at an angle of 24°

3.5 Design changes

Based on the findings of the previous chapter the most straight forward solution is to extend the slope of the connection cone. This solution terminates the large gap between the cone and wire rope. The loading condition becomes a side load and an unknown axial tension. The hoses should be able to withstand these forces because they are sufficiently supported.

In order to implement this solution the connection cone needs a redesign and should be produced again. The project is under some time pressure to perform a new test with an improved *Slingbuoy* before *Lorelay* leaves the Netherlands at the end of summer 2016. Therefore the decision is made to design a transition cone. This transition cone is clamped on the wire rope and functions as an extension of the connection cone. This way the *Slingbuoy* can be assembled and retested in a smaller timeframe, before *Lorelay* leaves the harbour. The drawing of this cone can be found in Figure 3.14. This test is not performed because the project has been put on hold due to a cost reduction.

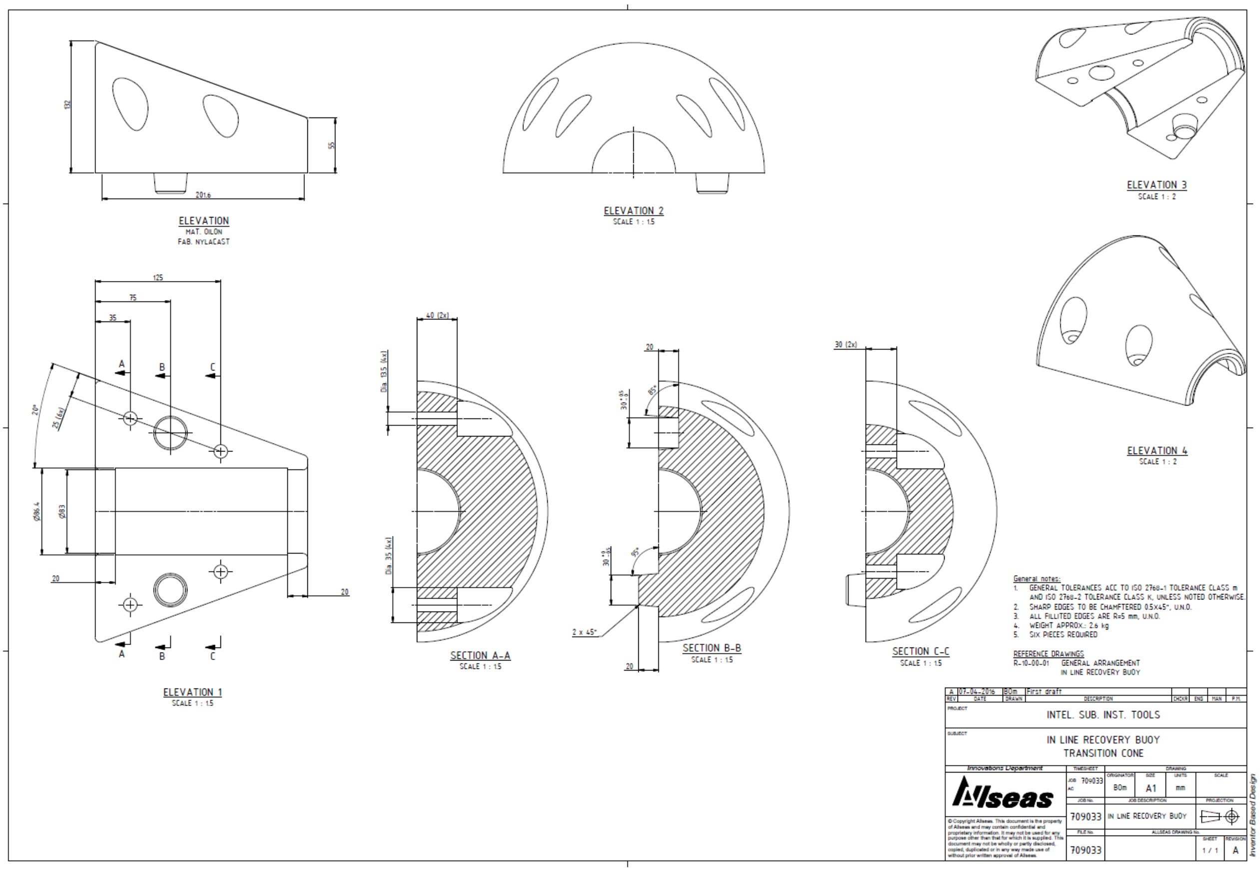


Figure 3.14 Drawing of the transition cone

3.6 Conclusion

Based on the data known so far some basic conclusions can be drawn. The tear in hose one is probably torn off at the edge of the connection cone. This failure can be probably be prevented by modifying the connection cone or by making use of the transition cone.

The failures in hose two are probably caused by a foreign object in the system. This object has not been found yet. Further research is necessary in order to validate if it was indeed a foreign object and how it was able to damage the hose. Especially in the case of hose two some validation tests need to be done to determine of the failure mode has been determined correctly.

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5.0 PERSONAL EVALUATION

At the start of my internship at Allseas I had to get used to quite some things, first of all the small office. I had some experience in a large company but I performed my internship in a branch office of the innovations department. This branch office is in Enschede and currently houses only three employees and five students (both graduates and interns) while the main office is in Delft and houses around 800 employees. All my colleagues in Enschede were working on their own projects and had contact with people in Delft but there was not much work done as an "Enschede team". Next to the office it took me some days to get used to working regular working hours. But after one or two weeks I was used to the daily routine. Allseas makes use of the 3D software inventor, this is the commercial counterpart of the SolidWorks software I was used to. They share a lot of features but the differences take some time to get used to. Other things I was surprized by were the companywide focus on safety and the huge amounts of force and money involved in the offshore industry.

In the first part of my internship, *The Slingbuoy* project, I had a really nice balance between practical work in the hall and theoretical work in the office. I really enjoyed the opportunity to combine those two.

In the second part of my assignment, the design of the FCS elements. The practical work in the hall was not necessary anymore. This led to the situation that I was behind my desk most of the time. In a while I noticed I missed the practical side of the first assignment. Nevertheless I liked the new assignment. Designing a part of a bigger system requires communication with the rest of the design team. The fact that the rest of the team was in Delft made the communication harder. I learned that I function better in a team, if the team is in the same location and you are able to drop by someone to discuss something. What I liked was my active involvement in a couple of meetings with possible suppliers. This really gave me the feeling I was part of the design team and not just an intern.

Overall I enjoyed my internship at Allseas. And since the second assignment was much bigger than would fit in my internship period, I will continue to work on the FCS elements. During the summer following my internship I will work as an Assistant R&D Engineer on the same design.