Structural Design of a Bow - Coupling Bend Pipe

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Master Thesis

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by

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Abstract

Despite the technological improvements in the dredging industry in the last decades, there are components that have not been investigated accordingly. One of them is the bow coupling bend pipe which can be found on trailing suction hopper dredgers. In contrast to most pipelines, the specific bend pipe is not only a means of transport for the dredged material but has a structural role, as well. Currently, it provides support to the subsequent pipes, which have no other connection to the vessel than the bow coupling bend pipe. Moreover, the abrasive nature of the slurry causes severe wear in the dredging pipes, with even more damage in bend pipes. These two factors lead to the relatively frequent replacement of a large amount of material, which also translates to an uneconomical procedure.

The objective of this thesis is to explore the potential of designing a new bow coupling bend pipe, which focuses on the reduction of the replaced material and the extension of its lifetime. This can be achieved by analysing the two aforementioned factors, in order to improve the management of the original material, resulting in an economical solution.

This study does not only analyse the design of bow coupling but also contributes to the understanding of slurry wear in bend pipes. Precisely, the report starts with a thorough explanation of slurry transport, describing also the three mechanisms of slurry wear and the parameters that influence the wear rates in a bend pipe. Furthermore, a literature review on the wear profile of bend pipes takes place in order to determine the expected wear pattern in the bow-coupling bend pipe. A total of five different wear zones are proposed to describe the wear rates in specific areas in the bend. The lack of research on slurry wear in large-scale bend pipes necessitated the use of assumptions to adapt the literature outcome on the bow coupling bend pipe. The information obtained from that research is used for the development of various concepts, considering also other aspects that contribute to the main goal of the thesis. Several concepts are investigated and compared based on the priorities of the project. The final design is further discussed, proceeding to modifications that improve its performance. The final geometry and materials of the system are defined in the finite element analysis. To do that, company guidelines and related standards were used to define the loads on the system as well as required realistic scenarios in which the design had to prove its structural feasibility. Finally, before providing recommendations for future work, the proposed design is compared with the current one, highlighting improvements in the three aspects of the thesis objective.

In particular, using almost 60% of the material required for the current design, the new solution extends the bend pipe's lifetime at least three times. That proves the remarkable effectiveness of the new design in the management of the pipe material, which is also directly linked with economic benefit. It is estimated that the capital expenditure of the new system will be twice as high as the current case, while the replacement cost is reduced by about half.

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Abbreviations and symbols

List of abbreviations

CFD	Computational Fluid Dynamics
COG	Centre Of Gravity
DE	Dredging Equipment
DEM	Discrete Element Method
FEA	Finite Element Analysis
FEM	Finite Element Method
LC	Load Combination
SF	Safety Factor
TSHD	Trailing Suction Hopper Dredger

List of symbols

t	Pipe wall thickness (mm)
v	Mixture flow velocity (m/s)
v_c	Mixture critical velocity (m/s)
c_v	Volumetric concentration of solids
d_m	Mean grain diameter (mm)
D	Pipe diameter (m)
R	Bend radius (m)
R/D	Bend curvature ratio
θ	Bend pipe angle $(degrees)$
ϕ	Pipe cross-section angle $(degrees)$
A	Cross-section area of the pipe (m^2)
g	Gravitational acceleration (m/s^2)
P	Pressure (bar, Pa)
m	Mass (kg, t)
$\dot{m}$	Mass flow rate (kg/s)
ϱ	Density (kg/m^3)
C_d	Drag coefficient
v_1	Inlet flow velocity (m/s)
v_2	Outlet flow velocity (m/s)
θ_1	Inlet orientation angle $(degrees)$
θ_2	Outlet orientation angle $(degrees)$
$a_{x,y,z}$	Vessel accelerations in x,y,z axes (m/s^2)
F_{Hose}	Floating pipeline force (N)
F_{Flow}	Flow force (N)
F_{Winch}	Winch force (N)
F_{Wind}	Wind force (N)

1
Introduction

In recent decades, there have been many technological developments in the dredging industry. Larger vessels can be built in order to accelerate the whole process, by reducing the number of cycles. Thus, the equipment related to the process has become larger and heavier. One of the parts that require reconsideration is the bow coupling bend pipe, which has a double role. The primary role is as means of transport of the dredged material, while the second role is to provide structural support to other parts. At the same time, the very abrasive nature of the dredged mixture constitutes another challenging part for the dredging pipelines, with considerably more wear effect in curved pipes. It is therefore important to investigate whether the bow coupling installation can be constructed differently. Precisely, this thesis focuses on the bend pipe design, to understand the existing challenges and discover where improvements can be made.

The first section of this chapter discusses the background of this project, the second section explains its purpose, and lastly, the structure of this report is described.

1.1. Background

Dredging can be considered as an excavation activity or operation that takes place underwater, in shallow or deep water areas, to gather up bottom sediments. A dredge is a tool used for scraping or sucking the seabed, while a dredger is a boat or ship equipped with a dredge. For this project, the focus lies on the so-called 'Trailing suction hopper dredger' (TSHD). The basic features of TSHD vessels are discussed in the next paragraphs before moving to more specific characteristics of the bow coupling.

1.1.1. Trailing suction hopper dredgers (TSHD)

The trailing suction hopper dredger (TSHD) is the only dredging vessel that carries out its work while sailing. The dredged material can be used for land reclamation, coastal defence and other purposes. The vessel can also remove material, for example, to prepare for the subsea installation of pipelines and cables, or wind turbine foundations, or to deepen and maintain rivers and waterways. A trailing suction hopper dredger is suitable for dredging many types of soil. The dredged material depends on the vessel's design characteristics, but generally speaking, THSD can dredge clay, silt, gravel, and mainly sand, which is considered for the project. Figure 1.1 shows an overview of the vessel, where the dredging-related parts are indicated with red colour. A description of the main equipment of the vessel can be found in Appendix B.

Once the vessel arrives in the designated area, its speed drops and a suction pipe with a drag head is lowered to the bottom reaching the sea or riverbed. Then, the pumps suck the material, mixed with water, through the suction pipes into the vessel's hopper, where the dredged material is stored. Once the hopper is full, the suction pipes are lifted back on deck and the trailing suction hopper dredger sails to the project site, or offshore discharge area, where the vessel empties its hold. This can be done in four different ways, as presented in figure 1.2.

The first way is to simply open the bottom doors and release the material into the designated area on the seabed. The second option is to return the sand to the seabed through one of the suction pipes. This is a relatively new solution, which allows for the controlled release and accurate placement of the material and limits the turbidity of the surrounding water. The third technique is called 'rainbowing', with the hopper contents being sprayed directly from the vessel's nozzle to the reclamation area. Pumps and water jets liquefy the material in the hold. The final method is to pump the material through a floating pipeline. A discharge hose, connected to the bow coupling bend pipe, transfers the material to the reclamation site [1].

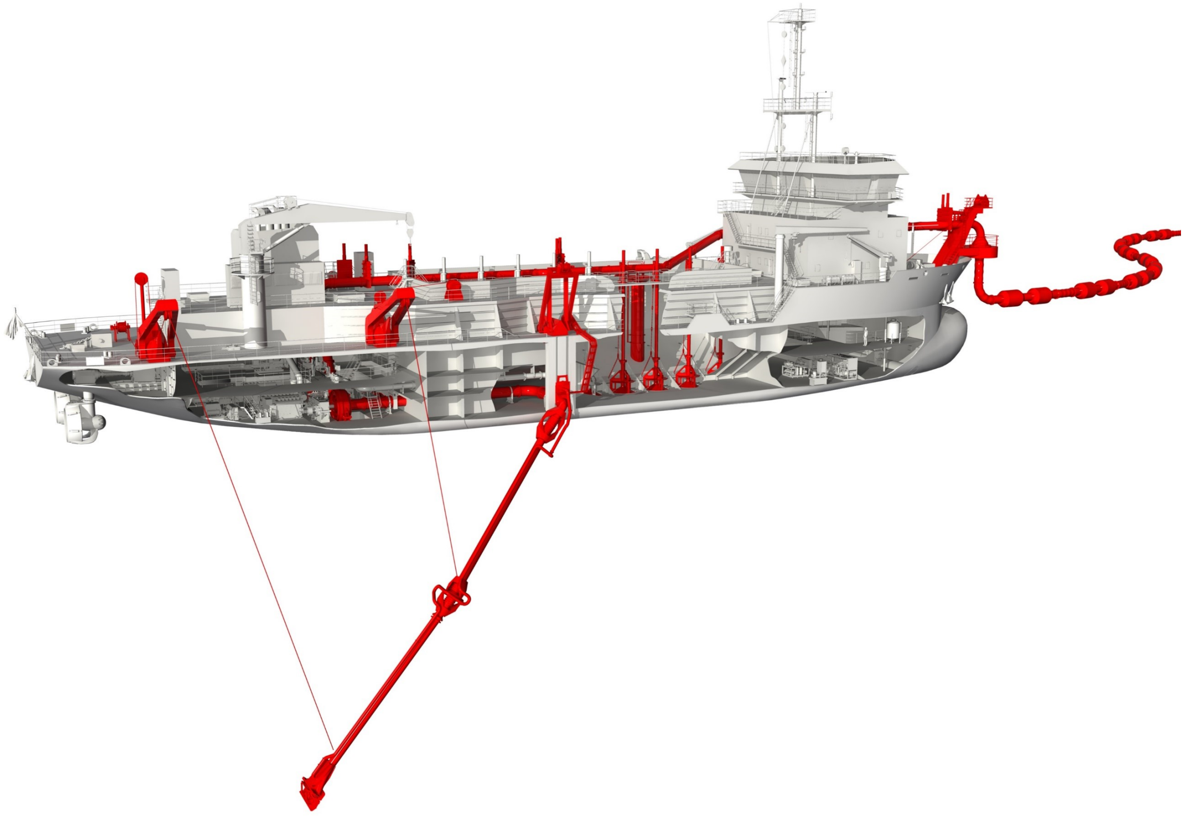


Figure 1.1: Trailing suction hopper dredger.

(a) Bottom doors (b) Suction pipe (c) Rain-bowing (d) Bow coupling

Figure 1.2: Discharging ways.

1.1.2. Bow-coupling bend pipe

This section focuses on the bow coupling bend pipe in order to provide adequate background information, as it is the part on which the entire research is centred.

Bow-coupling parts

The arrangement consists of a rectangular-shaped steel support tower, welded or bolted to the bow of the ship. The delivery pipe runs through the tower and ends at the bend pipe. The outlet side of the bend pipe is connected to the female connector part, also known as the "coupling mechanism" because it is the part that has the mechanism to couple the vessel with the floating pipeline. Specifically, the male connector part, which is at the end of the floating pipeline, is hoisted into the female part and locked. It should be mentioned that the male part constitutes a ball joint, which allows a movement of 15^o in all directions from the vertical axis. To lock the floating line to the coupling, the female part has a locking construction consisting of two half rings that can be closed employing a hydraulic cylinder. Also, the coupling is surrounded by a platform that leads the male part during connection and ensures accessibility for inspection and maintenance. A diabolo roller is fitted to guide the hoisting cable from the winch to the floating hose. The cable is used to hoist the floating pipeline into position.

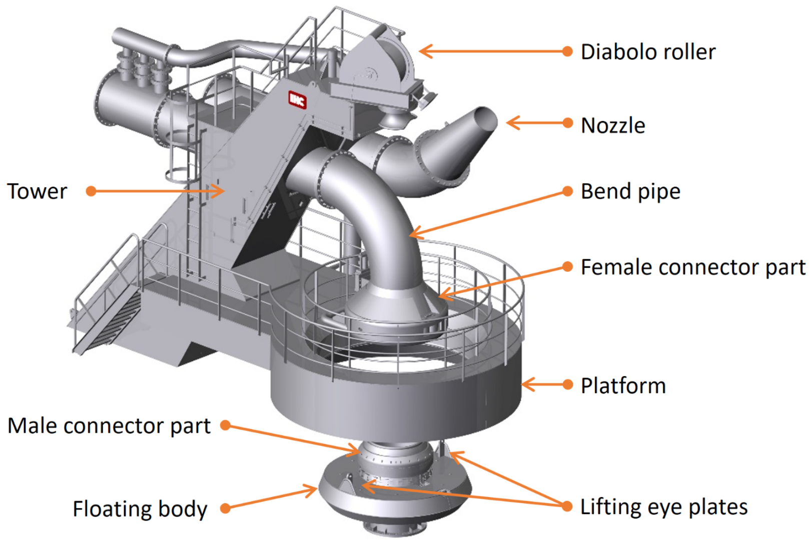


Figure 1.3: Bow coupling main parts.

Current design specifications

Figure 1.4 shows the main dimensions of the current cast bend pipe. As can be seen, the inner diameter of the bend pipe is 1000 mm and the bend radius is 1700 mm. The corresponding curvature ratio, defined as the bend radius divided by the inner diameter, is 1.7. Moreover, the thickness of the pipe is 35 mm, resulting in a total mass of 3001 kg, including the two flanges. The minimum thickness before replacement is 24 mm, meaning that the maximum thickness reduction due to wear is 11 mm.

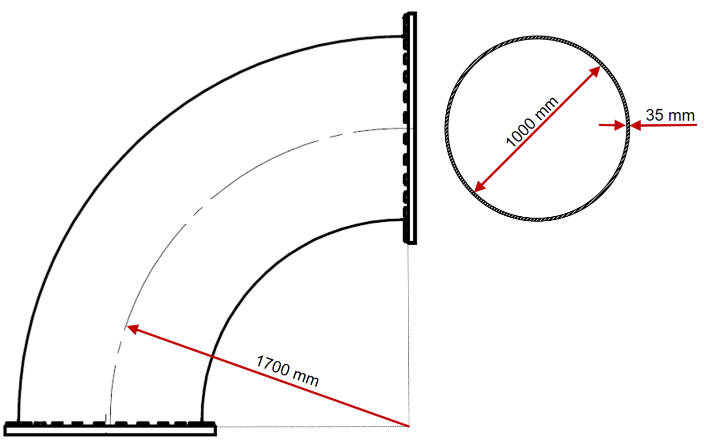


Figure 1.4: Current design dimensions.

Connection procedure

The first step is to approach the floating pipeline, which is located close to the shore. Then a work-boat attaches the winch cable from the ship to the lifting cable of the floating pipeline. The floating pipe consists of flexible pipelines which remain on the sea surface, having the floating body and the male part at the end. As soon as the winch and lifting cables are attached, the floating pipeline is pulled towards the bow of the vessel. A winch, located behind the tower, pulls the cable, which passes through the diabolo roller and then the platform to end up at the floating pipeline. Specifically, the last connection with the floating body is done by two chains, which are attached to the lifting eyes of the floating body. Due to this split, the male part is centred under the female part during lifting. The coupling mechanism comes in between the two chains, and as a result, the male part is pulled into the female part, which locks it. The female part, and consequently the bend pipe, hold the floating pipeline without any support throughout the discharging process.

Figure 1.5: Floating pipeline connection, [1].

Bow-coupling loading

As can be understood from the connection procedure and figure 1.3, the female part and the floating pipeline are only supported by the bend pipe, because it is the only part that is connected to the vessel. Thus, there are additional loads exerted on this specific bend pipe compared to the regular ones. Firstly, all curved dredging pipes have to withstand the high inner pressure and the flow forces, because of the redirection of the flow. However, this bend pipe has to support the weight of the female part as well as the high loads generated by the weight and the motions of the floating pipeline. Also, the environmental loads of wind and sea waves are applied to the system, although their influence is much lower compared to the floating pipeline load. The aforementioned loads and an accidental load that might occur during connecting the floating pipeline, are more closely reviewed in Chapter 5, where standardised load cases are applied to the system.

Replacement procedure

As the bend pipe is connected to the female part, both of them have to be disconnected and lifted. The first and most preferable way is to disconnect the bend pipe from the tower pipe and then lift it with the female part. The two parts can be disconnected at the shipyard in order to replace the bend pipe. The other solution is to first disconnect the female part and use the lifting cable from the diabolo roller to lower it. After that, the bend pipe can be disconnected from the tower pipe and then lifted.

Bow coupling designs: Van de Graaf

The bow coupling design, discussed in this section, is called "Van de Graaf", and it was developed by the eponymous company. Several different systems were designed in the past, such as "Single flex coupling", "Double flex coupling", "Stapel BV coupling", "Vosta Coupling", "E+S coupling", and “Amsterdam link". However, the benefits of "Van de Graaf" have made it the most applicable design, nowadays. At the same time, this design is divided into three configurations according to the position of the nozzle.

As figure 1.6 depicts, the first and most common option finds the nozzle next to the bend pipe, while the next two have a non-fixed nozzle. The second system has only one discharging pipeline and the nozzle has to be connected to the female part, the same way as for the floating pipeline. The last option has only one discharging pipeline as well. The distinctiveness of this arrangement is that both the nozzle and pipe can be mechanically moved and connected on the same pipe, depending on which discharging way is selected. More information about these three designs is provided in Chapter 4, as all of them take part in the conceptual design.

(a) Fixed nozzle (b) Coupling nozzle (c) Rotational nozzle

Figure 1.6: Bow coupling designs.

1.2. Research outline

1.2.1. Problem statement

As mentioned at the beginning of the chapter, the bend pipe has a double contribution to the system. The two functions that the current bend pipe has, lead to the increased thickness. The first and most important function is the flow of the dredged material. The flow comes with flow loading and slurry wear. The former concerns the high operating pressure of 25 bar, as well as the flow force due to the redirection of the flow in curved pipes. Also, the slurry transport, in combination with the small bend curvature ratio, leads to relatively high wear rates and consequently, rapid thickness reduction. However, the particularity of the case of the bow coupling bend pipe is the connection of the floating pipeline to the female part, which is attached to the bend outlet flange. Thus, the bend pipe does not just have to withstand its weight and the flow loading, but also the weight of the subsequent components and the complex behaviour of the floating pipeline. All those parameters lead to the increased pipe wall thickness of 35 mm, which should be replaced when reaching the minimum of 24 mm. The last 24 mm are necessary to avoid failure because of the additional loading under which the bend pipe should operate. As a result, more than half of the initial amount of material has to be replaced with a new cast pipe of about 3 t. A relatively frequent replacement of such parts has economic consequences due to the large cast model and the required fabrication before installation.

1.2.2. Aim

Considering all the aforementioned points, it is clear that the slurry wear and the loads exerted on the bend pipe constitute the main reasons for the frequent replacement of a large amount of material. The main objective of this work is to explore the potential of developing an alternative bow-coupling design that decreases the volume of material that has to be replaced while aiming at a longer lifetime. These two elements aligned with the replacement cost constitute the three aspects, which should be improved by the new design.

1.2.3. Research questions

This research focuses on the following main research question:

"To what extent can an alternative bow-coupling design reduce the replaced amount of material and extend its lifetime, in a cost-effective manner?“

In addition, the following sub-questions will be addressed in order to answer the main research question:

1. *What are the wear mechanisms for slurry transport in dredging pipes, and which of their parameters can be used to reduce slurry wear rates in the bow-coupling bend pipe?*

2. *What is the expected wear pattern of the bow-coupling bend pipe?*

3. *In which ways can a new design reduce the replacement material and prolong the pipe's lifespan?*

4. *What is the technical feasibility of the new design?*

1.3. Report outline

First, Chapter 1 provides the required background of the dredging industry and explains the problem of the current design. The second and third chapters focus on the wear in the bend pipe. Precisely, Chapter 2 starts with an introduction to slurry transport, in order to explain in the rest of the chapter, the slurry wear mechanisms and the parameters that affect them. Then Chapter 3, proceeds to a literature review on the wear profiles in bend pipes, which is used to determine the wear pattern of the bow coupling bend pipe. Moving on, to the second research area of the project, Chapter 4 describes the considerations and procedure of developing concepts, which are subsequently evaluated based on weighted factors. Chapter 5 has to do with the analysis of the new design where standardised load cases are applied to the system, and then the final parameters of the proposed design are compared with the current case. Finally, Chapter 6 summarises the conclusions of the project and provides recommendations for the company and future research.

It should be mentioned that the four sub-questions are answered at the end of Chapters 2, 3, 4, and 5, respectively, as well as in Chapter 6, which additionally states the answer to the main research question.

2
Slurry wear

In order to reduce the wear in bend pipes, first the factors that affect the wear rates should be addressed. Thus, this chapter explains the main characteristics of slurry transport and the wear mechanisms that occur while passing in a pipe. In the last section, all parameters that influence the wear rate in the bow coupling bend pipe are collected, and the way they affect the wear profile of the pipe is explained.

2.1. Slurry transport

One of the most important parts of dredging is slurry transport, a multiphase mixture that consists of liquid and particles. A slurry is a mixture of one or more different types of particles and a fluid, in this case mostly a liquid. It is a major field in dredging engineering, which has been researched for decades due to its complexity and significance [12, 13]. Slurries not only contain complex physical characteristics, but also a range of compositions, like various combinations of several solids and liquids. The size of the solid particles can vary from the scale of micrometres to centimetres [14]. Specifically, TSHD vessels are mainly used for the transportation of sand, whose size varies from 60 µm to 2 mm, see figure C.1. Particle size is one of the most significant parameters that affect the slurry's rheology, a dynamic process that determines the microstructure of the slurry. Additional parameters that influence the physical behaviour of the slurry are the density and viscosity in reference to the carrier liquid, and with regards to the particles, the density, size, and shape. Overall, the combination of the aforementioned factors results in several possible mixtures with different densities, viscosities, particles concentrations, particle size distributions and finally level of turbulence [14–16]. Thus, the slurries are categorized into different flow regimes, which describe the flow characteristic of the slurry in a straight pipe.

2.1.1. Flow regimes

Slurries can be divided into two main categories mainly based on the size of the solid particles, settling and non-settling slurries. The tendency of the solid particles to separate from the carrier liquid is low in non-settling slurries because the particles are sufficiently fine, light, or concentrated. However, this is not the case in settling slurries, and the tendency to separate needs to be taken into account when designing the slurry transportation system [16]. Still, under the umbrella of those two categories, more specific flow regimes can be found in literature, even though their definitions may vary sometimes. In this study, the reference will be made to the well-established classification reported in the handbook of IHC [4]. According to Berg [4], the flow of solid-liquid mixtures through a pipeline can be broken down into four different flow regimes.

Figure 2.1 shows an illustration of the four main regimes with their corresponding profiles in velocity and concentration distributions. The first profile is the mixture velocity, then the spatial volumetric concentration and the last one is the transport volumetric concentration. The spatial volumetric concentration is the volume occupied by the solids divided by the total mixture volume of a pipe segment. The transport volumetric concentration is the volume flow of solids divided by the total mixture volume flow. An explanation regarding the profiles of velocity and concentration distribution can be found in Appendix C.2.

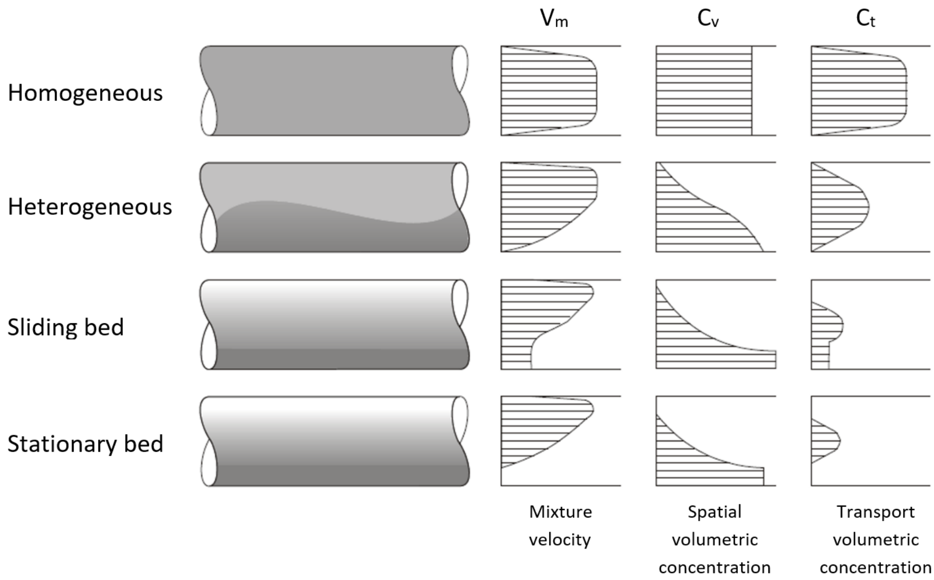


Figure 2.1: The four main flow regimes and their velocity and concentration profiles, [2].

Homogeneous flow

The first regime is the homogeneous suspension, which is a fully suspended flow. Notwithstanding, Berg [4] introduces another category of homogeneous suspension, the so-called, homogeneous non-Newtonian suspension. This area refers to slow- or non-settling slurries. These slurries are usually non-Newtonian e.g. suspensions of clay, fine ash, fine coal, raw cement and silt. Non-settling slurries flow in a pipe having a uniform distribution of particles across the flow section and an axisymmetric velocity distribution [4]. However, because of gravity, the concentration in the lower half of the pipe is higher than in the upper half of the pipe, so non-Newtonian homogeneous transport according to the definition never occurs [2], or at least it is extremely rare. Thus, Berg [4] considers as homogeneous suspension the slight variation of the concentration from the top to the bottom of the pipe although there is no difference in velocity between fluid and particles. That makes it a more realistic scenario, which usually occurs for fine materials and plastic materials with low concentrations [4].

Heterogeneous flow

Although all of the particles are in suspension, there is a significant concentration difference between the top and bottom of the pipe [4]. Other studies, divide the heterogeneous regime into two categories, one with fully suspended particles and the other with rolling and/or saltation of the particles [2].

Moving/Sliding bed

Particles in this flow are still moving along the pipe despite settling at the bottom. A portion of the solids particles are carried as a suspended load and the remainder is moved as a bed load. The velocity of the grains is lower than the fluid velocity and the concentration by transport is smaller than the concentration by volume [4].

Stationary/Fixed bed

Even though the slurry flow continues to take place, some particles settle down and remain as a stationary deposit on the bottom. In this area, the transport concentration differs strongly from the volumetric concentration. This is a dangerous situation because blockage of the pipe can occur [4].

2.1.2. Transition velocities

An important factor, which must be examined when pumping mixtures of soil and water, is the mixture velocity. Even though the particle size can affect significantly the flow regime of a mixture, velocity can also shift the mixture to other regimes by keeping the rest of the properties constant. For example, the so-called homogeneous suspension mainly concerns fine materials, but at the same time, even coarse materials can enter that regime at extremely high-speed flows. Those flow regimes are based on the concentration distribution in the pipe. The boundary velocities between the various regimes of flow are called transition velocities and may be determined by various equations [4].

By taking as a reference the four described flow regimes, the following velocities constitute their transition points, when the other factors remain unchanged [2].

1. No slurry transport -to- Stationary bed: Starting from zero and increasing the line speed, first the fixed or stationary bed will occur without suspension meaning that there are no particles above the bed.
2. Stationary -to- Sliding bed: Particles from the pipe bed start to erode. There is mainly saltation with limited particles suspended normally in the pipe. By increasing the velocity, the stationary bed will start moving. At the same time, the suspension rate increases with the line speed.
3. Sliding bed -to- Heterogeneous: At or above this velocity, the particles are fully suspended and move asymmetrically in the pipe.
4. Heterogeneous -to- Homogeneous: At very high velocities, the particles move symmetrically with the water in the pipe.

The transitional velocity, from the sliding bed to the heterogeneous regime, is the critical velocity because at this speed the hydraulic gradient appears to be at a minimum, which corresponds to minimum resistance exerted by the pipeline [2].

Critical velocity can be described, as the minimum velocity required for transporting a solid material through a pipeline without any particle deposition, which means that the particles of the soil in the mixture remain in suspension. If the velocity falls below this critical value, sedimentation occurs in the pipeline. The smaller the margin between the critical velocity and the actual velocity of the mixture is, the lower the resistance from the pipeline will become. This implies that the resistance is minimal at a velocity which lies just above the critical value. When the resistance in the pipeline increases, the required pressure for transporting the mixture increases, corresponding to more power.

MTI Holland [4] has developed a simple equation, using regression analysis, that is based on the results of experiments in the laboratory and measurements during dredging activities. The equation is valid for sand with a density of 2650 kg/m^3 and fluid with sea water density, 1025 kg/m^3.

$$v_c = 1.7 \cdot \left(5 - \frac{1}{\sqrt{d_m}}\right) \cdot \sqrt{D} \cdot \left(\frac{c_v}{c_v - 1}\right)^{\frac{1}{6}} \tag{1}$$

Where v_c in the critical velocity in m/s, d_m is the mean grain diameter in mm, D is the inner diameter of the pipe in m, and c_v is the volumetric concentration of solids.

For the current project, medium-size sand is considered and therefore the mean grain diameter is taken as 0.4 mm. The inner diameter is 1 m, as it is specified in the study requirements. The volumetric concentration considered has the maximum possible value, in order to minimize the risk of shifting to a lower flow regime [4]. TSHD vessels can reach a concentration of 0.48 during discharging, which corresponds to a mixture density of 1800 kg/m^3, with carrier liquid and particles' density, at 1025 kg/m^3 and 2650 kg/m^3, respectively.

The critical velocity according to Equation 1, under those conditions, is 5.63 m/s. As the operational line speed is about 6 m/s, that value comes to prove that the system falls into the desired heterogeneous flow regime, keeping also the hydraulic gradient at the lowest level.

Overall, the flow regime that is expected during operation is heterogeneous with all particles suspended, even though saltation and rolling might occur, as well. To remain in that flow regime, the critical velocity should be kept as the minimum line speed, below which the particles will start settling to the bottom. As shown the parameters that affect the critical velocity are the pipe diameter, particles' diameter, the volumetric concentration of the particles and lastly the densities of the two components of the mixture, solid and fluid. The determination of the flow regime is crucial for the further investigation of the wear in the bend pipe.

Thorough information about slurry transport, including flow regimes and transition velocities models and different approaches, can be found in these four studies [2, 17–19].

2.2. Wear mechanisms in dredging pipelines

The most significant factor reducing the bent pipe's lifespan is wear, which directly affects the part's mechanical performance. Wear can be defined as the progressive volume loss from a surface [20]. In the first subsection, the fundamental principles of slurry wear are explained.

To create effective countermeasures, scientists have conducted several investigations to comprehend the basic mechanism of wear. The multiphase flow allows particles to gain momentum and travel to impinge on the inside surfaces of pipes, fittings, valves, and other pumping equipment, wearing down these components. Because it poses massive issues for already-in-use equipment, notably pipeline systems used to transport slurries, researchers have recently paid close attention to slurry wear caused by solid particles.

Transportation of slurry through pipelines generally results in corrosion, abrasion, and erosion, with the last one being the most detrimental [21–26]. Slurry erosion typically happens when moving slurry impacts a surface, scars it, and removes material under conditions of turbulent flow. It should be noted that while abrasion and erosion are both mechanical wear processes that exhibit numerous similarities, they are occasionally misinterpreted for one another [27–29]. The key distinction between the two, however, is that erosion is the transfer of kinetic energy from the impinging particle to the target surface, whereas abrasion is the loss of material as a result of the passing of hard particles over the surface without impingement. In comparison to abrasion, the contact period between the erodent and the eroded surface is significantly shorter in erosion [30, 31].

2.2.1. Erosion - Impact wear

There are several definitions of erosion. According to Bitter [28], it is defined as "material damage caused by the attack of particles entrained in a fluid system impacting the surface at high speed", while Hutchings and Winter [32] define it as "erosion is an abrasive wear process in which the repeated impact of small particles entrained in moving fluid against a surface results in the removal of material from that surface" [33]. In any case, the general meaning and the way this word applies in this project is as the progressive loss, fracture, or displacement of material caused by the repetitive impingement of solid particles on a specific solid surface [34–36].

In the 1960s, Finnie [37] and Bitter [28, 38] systematically studied slurry erosion for the first time, proposing also erosion models like Neilson and Gilchrist [39], and Hutchings [40] [41]. Since then, various evaluation techniques and test methods have been used to assess erosion [31].

An investigation of previous erosion models by Meng and Ludema [42] revealed that 28 erosion models were linked to solid particle impingement, as well as 33 key parameters affecting erosion rate. However, robust models have not yet been developed for slurry erosion, which is a complex and understudied area. Certainly, this is the case for pipeline erosion, which is compounded by the fact that most literature on the wear of pipelines focuses on pneumatic conveying systems [43].

Erosion constitutes a very complicated phenomenon as a result of the interaction of numerous parameters, including the characteristics of the erosive particle, the properties of the eroded material, operating conditions, and erosion mechanisms. Generally, researchers in the literature refer to only two main mechanisms, regardless of the material ductility or brittleness, "cutting" and "deformation" as originally defined by Finnie [37]. These terms are not exactly what they are generally understood to mean metallurgically, especially when dealing with brittle materials. Therefore, a more accurate and detailed categorization of the erosion mechanisms is presented, as it was explained by Stachiowak and Batchelor [3], as shown in figure 2.2.

Different failure mechanisms can generally occur based on the particle's speed, the impingement angle, and the wall's material. Since the particle does not strike the surface or exert a significant force on it, there is essentially little wear for impingement angles close to 0^o [3]. When the impact angle is slightly larger than that, a wear mechanism that resembles the sliding wear's cutting process occurs, figure 2.2a. It is possible in situations when the shear stress caused by the impact exceeds the shear strength of the surface, such as mod-

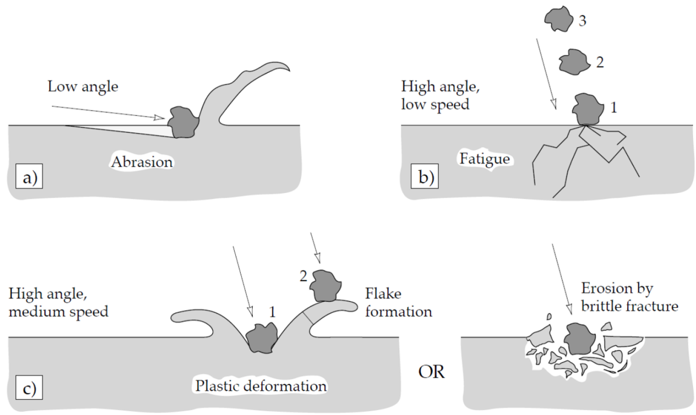


Figure 2.2: Impact wear mechanisms, [3].

est impact angles of relatively hard materials on ductile surfaces [18, 44]. However, everything changes when the particle strikes the surface from a higher angle. When the impact energy of the particles is inadequate to deform the material plastically, still the surface fails due to fatigue after repeated collisions, figure 2.2b. For higher velocity and consequently higher impact energy, plastic deformation of the surface is expected for ductile materials, forming flakes around the impact point, figure 2.2c [18, 44, 45]. For brittle materials, on the other hand, erosion fracture occurs, and as a result of subsurface cracks, the material is removed from the surface that is impacted, figure 2.2d.

As a result of those mechanisms, the pipe wall surface may develop a wavy pattern, perpendicularly to the flow direction. This effect is known as ripple formation or erosion ripple, and mainly depends on duration, particle properties and pipe material [43, 46, 47]. As impact wear is responsible for this formation and erosion is the dominant wear mechanism for bend pipes, erosion ripple is expected to gradually appear in bend pipes after a certain operational period.

Solid particle erosion depends on the ductility of the surface, acting differently on ductile and brittle metallic materials [48]. Firstly, figure 2.3 depicts the difference in the wear rates between ductile and brittle material according to the impact angle. The impact angle, which is the angle between the direction of the particle velocity and the target surface, can also affect the amount of slurry erosion. It is generally accepted that the maximum erosion rate occurs near normal impact for brittle materials. Nevertheless, ductile materials show maximum erosion at intermediate impact angles, with the exact number varying between 20^o and 50^o. For example, Al-Bukhaiti et al. [49] observed that the maximum erosion for steel AISI 1017 occurs between 40^o and 50^o. Still, Berg [4] and Patel et al. [20], stated that 20^o - 30^o is the range for the maximum erosion while Barkoula and Karger-Kocsis [50], Oka et al. [51] and Hufnagel et al. [52], concluded at 30^o, 30^o and 20^o, respectively.

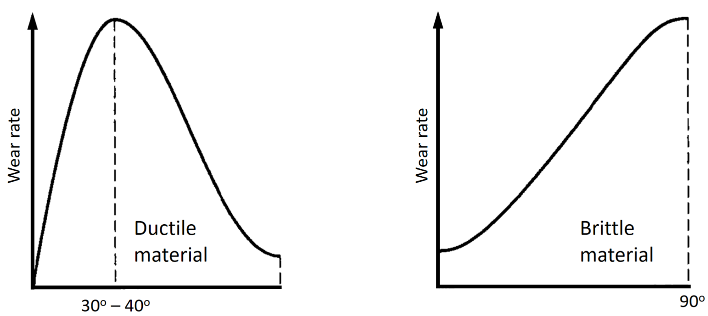


Figure 2.3: Impact angle for ductile and brittle materials.

However, if one added the two lines on the same graph, the top point of the ductile material would be way higher than the brittle one. Zolfagharnasab et al. [53] showed that the erosion rate of ductile materials can approximately be even ten times larger than brittle materials. Their excellent anti-wear performance led to the investigation of ceramic coatings to perform in slurry transport equipment [20, 54]. Finally, Zhang et al. [55] reported that the craters on a ductile surface are shallow and longer at low-impact angles, whereas at high-impact angles, they are deep wide and more circular.

Overall, as Finnie [37] discussed, for particles impinging at low angles on ductile materials [42], cutting wear dominates the erosion [56, 57], while deformation wear, or crack formation, becomes the predominate one for the brittle materials and particles impinging at high angles [28, 38, 58]. The two mechanisms usually act together to produce wear scars [59].

2.2.2. Abrasion - Sliding wear

As indicated before, abrasion is also known as sliding wear. According to Hutchings and Shipway [60], sliding wear can be defined as the loss of material during relative motion between two solid surfaces in contact under load. Following again the wear terminology from Stachiowak and Batchelor [3], there are four main mechanisms with which a sliding particle can remove material from a surface.

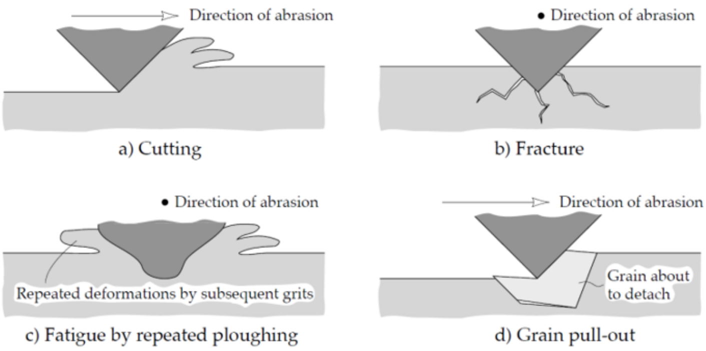


Figure 2.4: Sliding wear mechanisms, [3].

The mechanism in figure 2.4a, cutting, illustrates the basic principle in which a sharp object slides over a softer surface. Cutting deflects the material in the abrasion direction, forming a chip, which is removed from the surface creating a groove [61]. When the abraded material is brittle, it is likely that fracture will occur on the surface, figure 2.4b, and wear debris is a result of crack convergence. Conversely, deformation of the surface is more likely to occur for a ductile material, when blunt or rounded particles plough repeatedly, figure 2.4c. In this case, wear debris is the result of metal fatigue. In this instance, metal fatigue is the cause of wear debris. Grain detachment or grain pull-out is indicated by the final mechanism, shown in figure 2.4d. This process mostly applies to ceramics because of the comparatively weak grain boundaries. The whole grain is lost as wear debris in this process [3, 62, 63].

2.2.3. Corrosion

Chemical assault creates a film material in dredging pipes, a considerable wear that is essentially unavoidable. It is known as 'corrosive wear' and it is the process of chemical or electrochemical metal degradation [20]. However, the importance of corrosion in the dredging industry arises when it occurs with other wear mechanisms, erosion and abrasion in that case. In literature, the cumulative effect of both wear, chemical and mechanical, is perceived as "erosion-corrosion" [64–66], where erosion includes both impact and sliding wear in pipes.

It is proved that the total weight loss of materials during the erosion-corrosion process is generally higher than the sum of pure electrochemical corrosion and pure mechanical erosion due to the synergistic effect of erosion and corrosion [67–71]. Zeng et al. [67] designed a 90^o elbow to investigate erosion-corrosion of carbon steel, while Liu et al. [72] conducted erosion-corrosion testing utilising a carbon steel 90^o elbow to evaluate the influence flow velocity in the corrosive medium under single phase flow circumstances [66].

Corrosion can accelerate erosion in this synergistic action, and vice versa [73]. Erosion promotes corrosion by "cleaning" the surface, removing the corrosion product or film [67, 74]. Moreover, the repeated deformation of the surface from the abrasive particles forms a work-hardened layer [75], which is more anodic, a parameter that makes it much more susceptible to corrosion [76]. Lastly, the deformed surface from the erosion mechanisms increases roughness and therefore the effective surface area, which enhances to corrosion process [67, 68, 77]. Then, corrosion dissolves the work-hardened layer [75], increases roughness [68, 77], detaches all flakes and most importantly weakens the grains boundaries [67].

Nonetheless, as [67, 78, 79] stated, the contribution of pure corrosion at the bend pipe's total wear is much smaller than erosion, a situation that indicates that pure corrosion is not a dominant factor.

2.3. Wear parameters for bend pipes

In this section, the parameters that affect the wear, in the pipe are to be explained. These parameters can control the level of wear in a pipe and consequently, proper modification of them can lead to reduced wear. Nonetheless, by wear, erosion is mainly meant, and most of the parameters are related to that mechanism. Moreover, as the next chapter focuses on the wear pattern in the bow coupling and the maximum wear area in the bend pipe, the influence of each parameter is stated.

Generally, the parameters can be divided into four groups, concentrating properties and conditions of different components that are involved in the process. As shown in figure 2.5, the four categories are the particles, the slurry, the pipe wall and lastly the particle-wall contact. A detailed explanation of those parameters is given by Javaheri et al. [31], except for the parameters related to the pipe geometry.

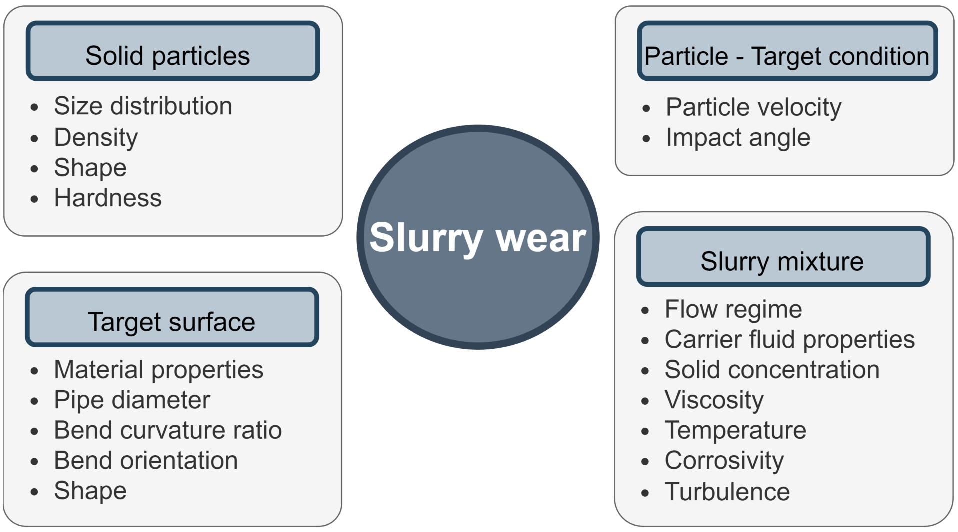


Figure 2.5: Slurry wear parameters.

2.3.1. Mixture velocity

The contribution of mixture velocity and consequently particle velocity is one of the most important parameters for the level of wear in the pipe. As the velocity increases, the particles hit the wall with higher energy resulting in more erosive wear on the target surface [80–84]. Even though, the wear rate, indeed, varies with mainly the particle size and impact angle, a mixture with high velocity will cause severe erosion regardless of the other parameters, a case that does occur the other way around. Additionally, Zhang et al. [85], stated that with a remarkable increase in slurry velocity, the puncture point location moved slightly towards the bend outlet.

2.3.2. Particles concentration

Another important parameter is the concentration of particles in the mixture. Considering that the wear of a surface depends on the frequency of collisions by particles, increasing the number of particles would undoubtedly increase the damage in the pipe [66, 86–89]. As the particle mass flow rate rises, Peng and Cao

[41] reported that the penetration and erosion rates rise linearly. At the same time, Li et al. [47], observed that after a certain point the wear magnitude tends to remain stable, despite the increasing number of passing particles in the pipe. As it was expected, the same trend was followed by the number of particle-wall collisions. An interesting note at this point is that while the previous two values were about to reach a plateau, the number of particle-particle collisions took an upward trend with the increase in concentration. This outcome can only be explained by the buffer effect [47, 90]. When particles collide, their energy is lost and their velocity decreases, resulting in the particles gathering near the wall surface and travelling slowly along it. As a result, subsequent particles first collide with previous particles, and a large number of particles decelerate near the wall surface, creating a buffer which damps the surrounding particles to directly hit the wall [47, 90]. Lastly, increasing the concentration of particles in a bend pipe, even though the maximum wear increases, its location remains the same and the eroded area expands [86, 91].

2.3.3. Particles size

Previous studies agreed that particle size will affect the erosion rate in the slurry flow. Generally, the increase in particle size will result in higher momentum, then higher impact energy and therefore increased erosion rate [92]. However, the relation of the particle size with the wear in bend pipes appears to be more complicated. Peng and Cao [41] stated that as the particle diameter increases, the wear rate decreases at the first stage and then increases, having minimum wear at 150 μm. Similar behaviour was observed by Ya et al. [91], who found the 60 μm particle size as the critical diameter for their model. A liquid's redirection at the elbows affects the particles greatly, especially small particles. This may be explained by the fact that the dominant force varies depending on the particle's diameter [41].

When the particles are larger and correspondingly heavier, the inertia force causes the particles to strike directly against the elbow wall, causing severe erosion on the outer wall. Small particles are considerably eroded when the secondary flow in the elbows is intense, causing them to impact the elbow side wall [41]. During the passage of a fluid through a pipe elbow, centrifugal and viscous forces interact to create a strong secondary flow normal to the axis of the pipe. This is a characteristic of fluid flow in elbows, and consists of two counter-rotating vortices, one in either half of the pipe cross-section [67, 93–96]. From a top view of the vortices at the exit of the bend, where the secondary flow is most probably fully developed, one can see that the particles from the inner wall travel through the centre of the pipe to the outer wall, where they split again to end up at the inner wall travelling from the sides.

Moreover, the density and viscosity of the fluid not only influence the secondary flow, but also the entire wear rate in the pipe. Zolfagharnasab et al. [53] analyzed the influence of density and viscosity of the carrier fluid for laminar and turbulent flow finding that with high density and viscosity, particles follow the flow streamline [97]. Despite the fact that both properties result in lower impact wear, the viscosity of the carrier fluid seems to have a better correlation with the erosion rates [83, 95]. The motion of the sand particles is determined by the relative velocity difference of the surrounding liquid phase through drag force. As the fluid becomes more viscous it restricts the movement and orientation of the particles, thus they are not highly erodent [98, 99]. Additionally, a liquid film at the wall surface may be expected to form, which becomes thicker as the liquid becomes more viscus, something which slows the particles' impact velocity and impingement energy [95, 99, 100]. Accordingly, the viscous fluid reduces the chances of the sand particles hitting the pipe wall with high energy, resulting in a drop in material degradation [99]. The single factor of fluid viscosity has no direct influence on the erosion behaviour, the main function of fluid viscosity is to influence the motion of the particles in the fluid, and then indirectly affects the wear process of the pipe [99].

Overall, for small particles the drag force is dominant and therefore the secondary flow vortices drive the particles to the outer wall and then the sides of the bend. As the secondary flow gets more intense approaching the bend outlet, that area and the downstream pipe are more likely to show significant wear on the outer wall and the sides. On the other side, for larger particles, the inertia force determines their movements, having a smaller influence from the fluid flow direction in elbows. Hence, the larger the particles are, the easier is to deviate from the fluid streamlines and impact the outer wall, where the erosion is expected to be maximum [41, 95].

2.3.4. Flow regime

Even though the flow regime describes the behaviour of the particles in the mixture passing in a straight pipe, it does not determine only the wear of the straight horizontal pipe but also of any subsequent pipe, the bend pipe in this case. Starting with the expected wear location in straight pipes, figure 2.6, depicts the pipe cross sections for the main four regimes.

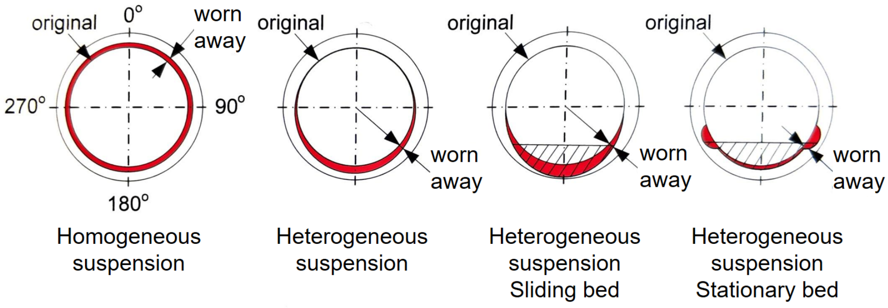


Figure 2.6: Flow regime wear profiles, [4].

For homogeneous suspension, the wear is distributed uniformly over the pipe wall, and the bottom wear does not significantly exceed top wear, like the rest of the cases. With a heterogeneous regime, where the highest concentration is at the bottom, the highest wear is in the lower part of the pipe, and specifically covers an angle of about 90^o to 120^o. By taking advantage of this, the pipes are turned over 90^o to 120^o in order to spread the wear, depending on the operational conditions and the inspection results. The wear due to a heterogeneous suspension with a sliding or saltation bed is concentrated at the bottom, by the larger particles moving over it. The angle of maximum wear is between 135^o and 225^o. A stationary or slow-moving bed can protect the bottom of the pipe from the faster-moving particles. The greatest wear is in the transition areas at 120^o and 240^o, between the stationary bed and moving mixture [4].

In reference to the bend pipe, the concentration profile of the mixture at the inlet plays an important role. Precisely, considering the heterogeneous flow regime, the particles travelling at the lower part of the pipe, will consequently move deeper in the bend, resulting in the first impact point close to the outlet [85].

2.3.5. Pipe diameter

It seems that the diameter of the pipe does not only affect the flow regime in the pipe, but also the wear caused by the slurry transport. Xie et al. [101] and Mohamad [83] found that the maximum erosion wear decreases with the increase of the pipe diameter. Kannojiya and Kumar [102] changed the diameter from 50 mm to 250 mm, and an about eleven times lower wear rate was observed. However, further enlargement of the pipe showed less influence on the wear rate which tended to stabilise, a situation that comes to agree with Peng and Cao [41]. They explained that the turbulence is more intense near the wall of a pipe with a smaller diameter, the particles gain more momentum and, strike it with greater impact and cause severe erosion [102]. In comparison to smaller pipes, larger pipes have a lower erosion rate due to a higher turbulence intensity. Because of the narrow channel that the multi-phase slurry flow goes through in the smaller diameter pipe, the solid particles are diverted from the stream and repeatedly collide with the pipe wall. Additionally, Kannojiya and Kumar [102] and Mohamad [83] showed that although the location of the maximum wear is more or less unchanged, the affected area expands significantly.

2.3.6. Bend curvature ratio

The bend radius in simple words determines how abrupt the change of the flow domain direction is, and consequently the length of the bend pipe and most importantly the location where the particles will hit the wall for the first time and under what circumstances. It is generally accepted that a sharp redirection of the flow causes more severe wear on the bend wall [83, 101, 103]. A large value of R/D ratio makes the bend curvature longer allowing for a more steady flow and therefore less collision with the wall [41, 102]. There are, however, significant differences between the impingement angle and impingement number per unit length. In comparison with the short radius bend, the long radius bend has a smaller impingement angle [103].

It is worth noticing that when Wang and Shirazi [103] increased the bend ratio from 2.4 to 7.8, the maximum impingement number per unit length was about one-third of the previous one. Similar results were found by Ya et al. [91], who observed that the wear rate in a 3D bend ratio was almost half as the one with a 1.5D bend ratio. They also stated that the main reason for that is the momentum and the angle of the first impact in the bend. For short bend radius, the particles hit the wall with tremendous energy and even though they cause significant wear they are carried away from the wall by the flow. On the other hand, if the particle hit the wall at a very low angle, it will not cause the same damage but it will hit the wall in more areas with the remaining energy after the first collision [91]. According to Wang and Shirazi [103], compared to the short radius bend, the long radius bend has fewer particle impingements per unit length, resulting in a larger particle impingement area. They also concluded that those two factors play a major role in reducing penetration rates in long-radius bends and elbows.

Finally, as one could expect the maximum erosion in the pipe, just like the first impact in the bend, moves deeper in the pipe, close to the outlet with the decrease of the bend radius, even though the impact velocity remains almost the same [103].

To sum up, the rate of impingement per unit area and the impingement angle vary with the elbow radius. The higher the curvature ratio, the longer the bend pipe is and consequently the more distributed the worn area is, with lower total wear. About the location of the maximum wear it is clear that the as the bend radius increases, it shifts closer to the inlet with a noticeably lower magnitude.

2.3.7. Bend pipe orientation

The pipe orientation can constitute an important parameter when gravity is applied on the system, in combination with a non-homogenous regime. The two contradictory orientations have a horizontal pipe upstream and a vertical pipe downstream, downwards and upwards, respectively. Peng and Cao [41] suggest that there is an insignificant difference in the wear rate pattern along the bend when the orientation was changed. Using particles of 0.2 mm in a bend pipe with an inner diameter of 40 mm and bend radius 60 mm, they found that the maximum penetration rate is always close to the bend outlet for all orientations.
However, Zhang et al. [85], using much larger particles (2 mm - 15 mm), with a 10 cm pipe size and 3.5 curvature ratio, found that the location of the maximum impact force varies with the bend orientation. As discussed at the beginning of the chapter, the flow regime mainly depends on the particle size, concertation and velocity. As a result, the particles were very close to the bottom of the upstream pipe before entering the bend, something which affected the location as well as the angle of the impact. By gradually changing the direction of the gravity acting on the system, Zhang et al. [85], observed that for horizontal-to-vertical-downwards, the maximum impact force is closer to the bend outlet with a larger angle, compared to the horizontal-to-vertical-upwards case. Similar results were occupied by Deng et al. [104], where despite the fact that the carrier fluid was gas, they concluded that the horizontal-to-vertical-downwards orientation is more likely to show higher wear rates. They base that on the fact that the particles travelling on the pipe bottom hit directly the wall with less chance to collide with other particles. A case like this reduces the inter-particle collisions, which correspond to increased erosive wear on the bend wall.

2.3.8. Pipe material properties

As mentioned before, the ductility of the target surface plays a major role in the wear mechanism and the resulting surface condition. This parameter is directly associated mainly with the impact angle and then with other impingement parameters, like velocity. In general, brittle materials show much greater wear performance, regardless of the other factors. The prevailing opinion is that hardness is one of the most significant factors, influencing the erosion behaviour of a wide variety of materials [105–112]. However, it should not be considered as the main parameter since there are cases for which material hardness does not constitute a reliable indicator to predict erosion rate [31, 105, 107, 110, 113, 114].

2.3.9. Discussion

This section discusses the parameters that affect the wear in a bend pipe. It is clear that any parameter that has to do with the particles and the carrier liquid, cannot be changed. On the other side, the operational conditions like velocity, and concentration, can be controlled but the only way is by lowering both of them, in order to keep the heterogeneous flow regime. However, this solution would result in slower discharging operations, which is not economically beneficial for the company. In reference to the pipe geometry, the pipe diameter and bend radius cannot be increased because of the space limitations, see Section 4.2. Also, an enlarged pipe in the same space would result in a lower bend radius, which increases wear, and secondly, the required power would be much higher.

Apart from those parameters, the shape of the bend pipe was also investigated, as a possible solution to reduce wear. Section C.4 discusses the influence of pipe geometry on wear reduction for elbows. There have been many studies done to reduce the wear of pipe wall surfaces by changing their shapes or adding extra members. However, most of the research concerns gas-solid mixtures travelling in small-scale pipes, without having any connection with the dredging industry in terms of flow conditions and sizes. At the same time, the special, and therefore expensive geometries proposed, require large-scale modifications, out of the allowable area according to the project's limitations. Notwithstanding, extra members in the pipe can be seen as sacrificial elements because they become more prone to erosion. Thus, their replacement should take place frequently, especially with slurry wear, leading to a high-cost solution considering the relatively frequent replacement, manufacturing and process interruption. Nonetheless, the effort for exploring alternative and promising solutions constitutes a positive sign for the future, but based on the current research level, none of them can be used as they are at very early stages, especially for the dredging industry.

Considering all of those parameters, the only one that can be changed is the material of the bend pipe.

2.4. Conclusion

First, in this chapter, the slurry transport was introduced, and the flow regime was determined. Based on the flow conditions of the system, a heterogeneous flow regime is expected, which means that all particles are fully suspended although they travel close to the bed of the straight horizontal pipe. The next section described the three slurry wear mechanisms that occur in dredging pipelines, erosion, abrasion and corrosion. Erosive wear was found to be the most dominant mechanism, during slurry transport and especially in bend pipes. The last step was to gather all parameters that affect the slurry wear rates as well as the wear profile in the bend pipe. The former is required to be able to investigate which parameters can be used to reduce the wear rates in the pipe, while the latter contributes to determining the wear pattern of the bend pipe in the next chapter. By considering all of the parameters, it can be concluded that the only possible solution that can improve the current situation is to use a different material, with better wear resistance. However, the challenging task of utilizing a material with relatively high brittleness is something that should be taken into account in the conceptual design.

3

Wear pattern in the bend pipe

The aim of this chapter is the determination of the wear pattern in the bend pipe, with a focus on the location with the maximum wear rates. Extensive efforts have been dedicated to predicting the erosion magnitude of a standard elbow both experimentally and numerically. Even though the majority of studies were carried out using gas as carrier fluid, the following section summarises numerous projects that investigated the bend wear with liquid. Overall, the sizes of the pipes that are equipped for not only experimental but also computational analysis are much smaller compared to the desired size for the current assignment. However, a good understanding of erosion in small pipes can also allow reasonable estimation for erosion in large dredging pipes. Thus, the outcome of the research is analysed in the second section of the chapter. There, the final wear distribution for the bow coupling bend pipe is determined, considering also a number of assumptions with respect to the wear parameters.

3.1. Background

This section presents some projects from the literature, which analyses mainly the erosive behaviour of particles flowing in liquid when passing through bend pipes. The number of papers that uses liquid is limited and none of them is specialized for dredging applications. The projects that are discussed hereafter are verified and validated by experimental results and other valid models. Most of them utilize water as carrier liquid in a 90-degree standard elbow pipe (R/D = 1.5), and they always have long straight pipes before and after the bend pipe. More information about the analysis procedure, validation and verification, results and reasoning for them is provided in the corresponding articles.

Peng and Cao [41] performed simulations for 34 scenarios, using different pipe sizes, bend orientations, bending angles, bending ratios, mixture velocities, particle sizes and flow rates. First of all, they compared predicted and experimental penetration rates for elbows for five erosion models with different particle-wall rebound models. The orientation of the bend was horizontal-to-vertical downwards and even though, there were significant differences regarding the magnitude of penetration rates for the models, all of them showed clearly the gradual increase of erosion along the bend pipe maximizing their values at the outlet region. A representative example used 0.2 mm particles and 0.2 kg/s mass flow, with 10 m/s mixture velocity in a 40 mm pipe diameter. Initially, there was a slight increase until 35^o, after which the wear rate remained almost stable until 60^o, where the significant increase started and despite the fluctuations, there was a peak at about 75^o and another one even higher just before 85^o.

The aim of Zeng et al. [67] was to investigate both erosion and corrosion in a 90^o elbow with a 50 mm inner diameter. The focus of this study was on the experiment, but the authors proceeded to computational fluid dynamics (CFD) analysis to understand better the behaviour of the mixture in the bend and then relate that to their experimental results. The mixture had 4 m/s velocity and was composed of water and sand particles of 450 μm diameter, 1.2% mass concertation and 0.235 kg/s mass flow rate. In the upstream straight pipe, the wear rates at the innermost side are greater than those at the outermost side, whereas, in a downstream straight pipe, the opposite occurs; the values at the outermost side are greater than those at the innermost side. The erosion at the elbow is significantly higher at the outer wall and escalates along the flow direction reaching the maximum value at the outlet area. In that area, the sand concentration is higher because of the inertia that leads them to the outer wall and the second reason is the secondary flow effect which is more intense near the outlet and drives the sand particles towards the outer wall. Both reasons result in a frequent impact of sand particles on the wall and consequently cause severe erosion there. The inertia is mainly responsible for the erosion along the axis of symmetry at the outer wall, increasing gradually towards the outlet. The secondary flow also increases the erosion at the middle of the outer wall but that happens after about 45^o, where the secondary flow strengthens until the outlet where is fully developed. As a result, the erosion does not only increase towards the outlet but also spreads to the sides of the bend, with a decreasing magnitude. Overall, the erosion appears to be higher at the bottom of the bend inlet, which decreases significantly by entering the bend. About the outer wall, there is a tiny increase until about 20^o, then a more important rise until about 60^o, where the erosion rate increases more dramatically until about 80^o, where the maximum rate was measured on the elbow. Also, after about 60^o, the erosion rate has a significant magnitude towards the sides of the outer wall. Finally, the measurement points at the downstream pipe showed significant damage on the outer wall and the sides, indicating that the wear pattern continues after the bend exit.

The outcome of the previous study was used as a validation model by Khan et al. [66]. To do so, they created similar operating conditions and compared their results, which seemed to have a great agreement, validating in that way their CFD model and experimental setup. For the CFD model, they used three different meshes, which predicted almost the same erosive profile of the bend outer wall. The maximum erosion is at the pipe outlet area, having though noticeable wear between 60^o and 90^o. Still, the plane of symmetry concentrates the highest erosion rates, while the intensity decreases gradually moving to the sides of the bend.

Mouketou et al. [95] used a standard elbow pipe with 40 mm inner diameter, inlet velocity of 30 m/s and 150 μm diameter particles with 0.2 kg/s flow rate. For the CFD simulation, the method Eulerian-Lagrangian was selected to model the multiphase flow of oil-water-sand in an elbow with horizontal-to-vertical downwards orientation. The bend pipe's outlet, particularly at 87^o, was observed to be the most eroded area. This finding is consistent with experimental data from Blanchard et al. [115], with which the authors compared their findings. It is apparent that, even though the maximum erosion was found at the outer wall, the sides of the bend after about 30^o were predicted to have erosion. The reason for that effect is the high viscosity of the oil in the mixture which greatly influences the trajectories of the particles towards the wall sides. This is a characteristic example where the viscosity and consequently the secondary flow determine the overall wear of the pipe. The significant spread of erosion on the side walls close to the bend exit can be explained in consideration of this observation. As a general rule, erosion damage happens mostly in two places on the elbow. Due to the direct impingement of the particle entrained in the liquid phase, the extrados located close to the bend exit is the main area. The second spot is on the downstream straight pipe's side walls, close to the bend exit, as a result of secondary flow driven on by centrifugal forces.

Khan [86] used a considerably larger pipe compared to the rest of the studies, finding again the maximum wear at the bend outlet. Specifically, the model was composed of a 510 mm diameter bend pipe with 10% sand particles of 250 μm diameter flowing with water at 2m/s velocity. In reference to the wear distribution in the rest of the pipe, the first about 25^o did not show any wear sign, but between 25^o and 65^o, the simulation predicted low wear while the area after 65^o had medium wear rates, to find the high wear at about 90^o.

Kannojiya and Kumar [102] made use of a 100 mm pipe diameter with 6 m/s velocity and 10% of 150 μm particles with 2200 kg/m^3 density. Both of their models, with 1.5 and 2 curvature ratios, found the maximum worn area to be on the outer wall showing also an increasing trend towards the outlet. As it was expected, keeping in mind the explanation from the previous section, the eroded area for a 1.5 ratio bend is more concentrated between 45^o and 90^o, while for bend pipe with R/D=2, the erosion rate starts at around 30^oat the centerline of the outer wall and continues on the outer wall, until about 90^o. For both curvature ratios, the erosion close to the outlet tended to affect the sides, covering approximately the entire outer wall and the high rates continued to the downstream pipe.

Additionally to those, there are more studies that found the maximum erosion at the outlet of the outer wall, with the wear gradually expanding to the sides of the pipe along the direction of the flow [83, 101, 116]. Chen et al. [35] used discrete element method (DEM) model and CFD techniques to evaluate the wear rate of different elbow configurations with 0.15 mm sand particles and found that the outlet was the most eroded one for all elbows configurations. Moreover, Zhang et al. [85], using a bend with 3.5 curvature, found the maximum impact force at the bend outlet and the next pipe, while the first 27^o of the outer wall was almost intact. Lastly, Blanchard et al. [115] claimed that the maximum wear location varies from 75^o to 105^o, and suggested that it should be expected at $85^o \pm 15^o$ around the bend outlet, on the outside surface on the line of symmetry, regardless of particle size and bend geometry.

In reference to studies with gas as a carrier fluid, they far outnumber the ones with liquid, giving a clear picture of the wear profile when air with particles travel in a bend pipe. As it was expected the properties of air compared to liquid change the wear pattern in a bend pipe. In contrast to gas fluids, liquid fluids have considerably higher density and viscosity. Thus, the surrounding fluid exerts a dominant drag force on the sand particles pulling them towards the outlet, and simultaneously the secondary flow can play a crucial role in their motion [67]. However, these two aspects do not apply to the same extent in gas fluids, and eventually, the sand particles can easily deviate from the fluid streamline and impact the outer wall. Specifically, simulations and experiments showed that for various flow conditions in bend pipes with mainly a 1.5 curvature ratio, the maximum wear is located between 45^o and 60^o [7, 8, 80, 87, 117–121]. This outcome could be expected bearing in mind that most of the particles follow the straight line coming from the upstream pipe and hit directly the wall. The location is also related to the centerline of the upstream, where the velocity is maximum and therefore the particles flowing at that level strike the wall with higher energy.

3.2. Wear zones

The aim of this section is to discuss the wear pattern of the bow coupling bend pipe, so it can be used in the conceptual design. Firstly, the outcome of the conclusion from the literature review is discussed, dividing the bend pipe into four wear zones. Then, a number of adaptation steps are explained in order to bring the literature result as close as possible to the real bend pipe. Since there is no information specified in the presented study, like analyses for these operational conditions, the bend pipe wear is determined by assumptions which are based on the literature. Figure 3.1 shows the definition of angles that are used in this section, θ and ϕ.

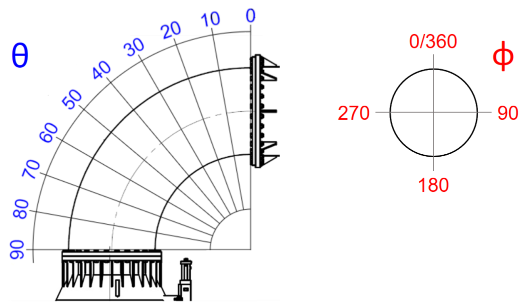


Figure 3.1: Definition of bend pipe angles, θ and ϕ.

3.2.1. Literature outcome

Based on everything discussed in this chapter and mainly the previous section, the wear pattern of a bend pipe is illustrated in figure 3.2. Precisely, this figure shows the outcome of the literature review if it is applied on a bend pipe with a small diameter and 1.5 and curvature ratio, like most projects used. It can be seen that there is a total of four wear zones, according to the expected level of wear.

The critical zones are separated based on the wear rate differences, without meaning that the wear rate in a zone is constant. The literature clearly showed that there is very low wear at the inlet of the pipe, the first about 25^o, and the entire inner wall. Then, a medium level of wear was found after 25^o at the outer wall, and finally, after 60^o, the erosion increased significantly, with the maximum rates being in the last 10^o, at the outlet area.

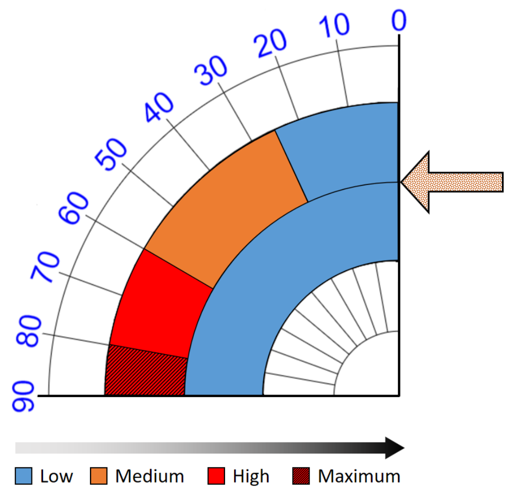


Figure 3.2: Bend pipe wear zones: Literature outcome.

3.2.2. Adaptation steps

A total of five steps were enough to adapt the literature outcome to the bow coupling conditions. Figure 3.3 shows how the first four steps influenced the wear zones, to end up with the final wear distribution, as shown in figure 3.4, after the last step.

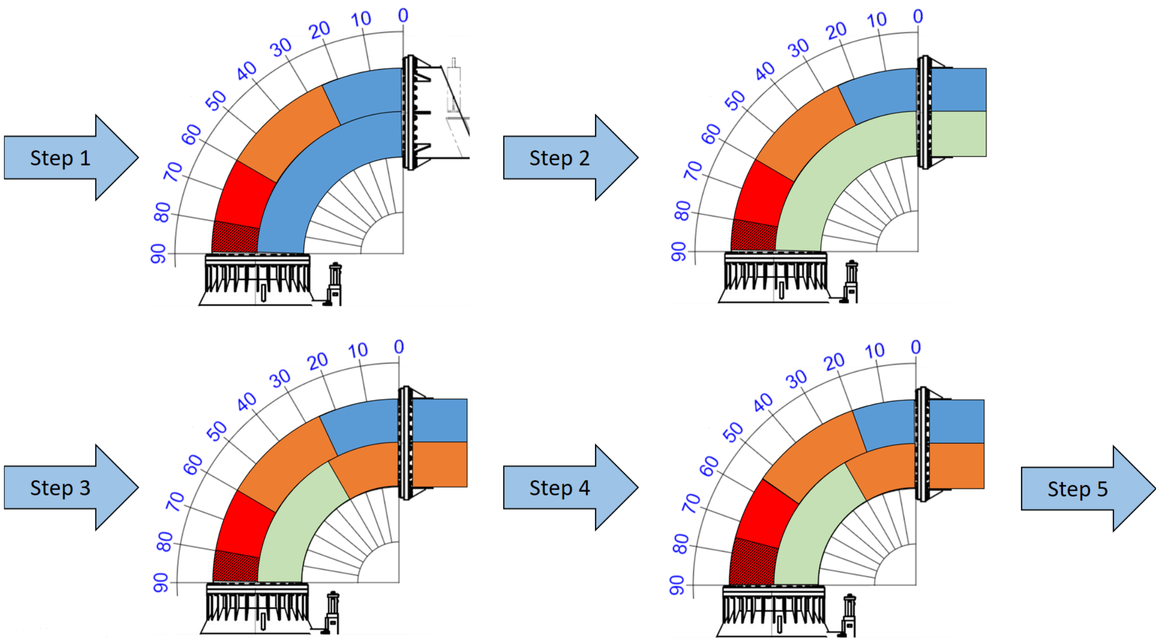


Figure 3.3: Bend pipe wear zones: Adaptation steps.

Step 1: Geometry

The first step was to apply this configuration on the bow coupling bend pipe, 1 m diameter and 1.7 m bend radius. The influence of geometry on the overall wear is described in the fourth step.

Step 2: Inner wall

This step is about the inner wall. Even though the literature did not show any sign of wear there, in reality, there is wear, at least at the inlet. This wear is related to the previous pipe, in which the heterogeneous regime is expected to cause wear at the pipe bed. Thus, it is expected that this erosion will continue at the inlet of the bend and then gradually die out towards the outlet. Moreover, low to medium wear can appear in more locations of the inner wall, especially close to the outlet as a result of the secondary flow. Apart from that, the flow profile that the slurry enters the bend pipe can play a significant role. In case a vessel has a very short upstream pipe another elbow close to the bow coupling, the concentration of the particles might not be at the pipe bottom while entering the bend. Therefore, considering also the chaotic flow streamlines, the mixture might cause significant wear at the bend sides. Consequently, the inner wall is considered separately, with green colour, whose wear rate is between the low and the medium wear zones. This is because the wear on the sides approaching the outlet as well as the inlet wear can be higher than the top side of the inlet. However, extreme operational conditions can change the wear profile of the previous straight pipe, and therefore the wear rate at the bend inlet.

Step 3: Inner wall - Inlet

The flow velocity as well as the concentration of the mixture can be controlled by the operator of the vessel. The former is by changing the pump rotational speed and the latter by liquidizing the mixture in the hopper. Also, the particle size distribution is hard to be predicted since various soil particles can be sucked by the drag head and that also depends on the dredging area. Considering a mixture with lower velocity, higher concentration and even larger particle size, the flow regime and consequently the wear profile of the straight pipe would be different. A flow with saltation or even a sliding bed would increase significantly the wear rate at the bottom of the pipe. Even though based on the calculation in Section 2.1.2, heterogeneous flow is expected, the undesired scenario of the sliding bed should be considered. As a result, it is assumed that the maximum wear of the upstream pipe can reach the rates of the medium wear zone. In reference to the bend pipe, the medium wear will continue for the first 30^o.

Step 4: Outer wall

The fourth step focuses on the outer wall and specifically, the parameters that can affect the wear zones and in which way. Starting with the geometry of the bend pipe, one of the most significant factors is the large diameter of the bend. The only way found that the pipe size affects the wear in the bend, is that the larger the diameter is the lower and more distributed the overall wear is predicted to be. Thus, the diameter does not seem to influence the location of the maximum erosion, but only the affected area, with much lower damage. A larger worn area with lower wear rates is the result of an increased curvature ratio, which can also shift the maximum wear slightly away from the bend outlet. Another part that is related to the geometry is the length of the upstream pipe, which as discussed in Step 2, can lead the particles to unexpected locations.

Moving on, the operational conditions can influence the wear zones of the outer wall. As mentioned in the previous step, velocity, concentration and particle size distribution, are three very important parameters, which can first affect the flow regime. A more homogeneous regime would result in a more distributed wear of the bend with a lower maximum rate. On the other side, a flow regime closer to the sliding bed would concentrate the wear towards the bend outlet, as most of the particles would enter the bend pipe from the bottom of the inlet. In any case, the location of the maximum wear is expected to remain the same. At the same time, generally, a high concentration of particles expands significantly the worn area, especially with smaller particles which are driven by the carrier liquid and can hit any location of the pipe. Small particles can also increase the wear of the outer wall and the sides close to the outlet and probably the downstream pipe, because of the secondary flow.

All those parameters show that the maximum wear can be found slightly before the bend outlet, while the area with noticeable wear can be larger. Considering that, it was decided to shift the medium and high erosion zones by 5^o towards the inlet. Thus, the maximum wear zone is increased by 5^o, whereas the low wear zone covers only the first 20^o.

Step 5: Downstream Pipe

The last step examines the influence of the subsequent, vertical pipe, connected to the bend outlet. All projects in the literature considered long straight pipes before and after the bend pipe, in order to analyze a bend pipe. However, in the case of the bow coupling, the next pipe is the female part, a complex part which has never been studied for slurry transport. In the female part, the end of the floating pipeline, the male part, is connected and the special connection allows the floating pipeline to operate with up to 15^o angle from the vertical axis. To begin with, the connection from the female to the male part is expected to generate intense turbulence in that area. Also, the orientation of the floating pipeline might have an effect on the path that the flow streamlines and the particles follow. By taking those points into consideration, it was decided to add 5^o to the maximum wear zone at the outer wall. In reference to the inner wall, the last 5^o are expected to show medium wear.

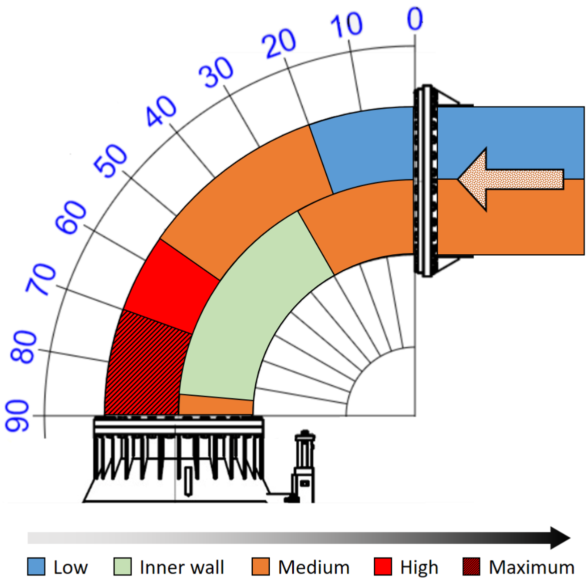


Figure 3.4: Bend pipe wear zones

To sum up, the first 20^o of the outer wall is expected to have the lowest wear of the pipe, after which the wear rates take an upward trend until 55^o. After that point, there is an even more significant increase in the wear rates, which reach a maximum in the last 20^o of the bend pipe. The inner wall is not expected to show significant wear rates, with the exception of the two bend ends, first 30^o and last 5^o, which are affected by the flow situation in the corresponding connected pipes. A more detailed description of the wear rates in the bend pipe is given in Appendix C.3. As one can understand, the lack of wear analysis for a project increases considerably the level of uncertainty. As there is no study specialized on the bow coupling bend pipe the decisions that had to be made, tended to be more conservative. That led to larger areas that were expected to have significant wear, compared to the wear zones as initially defined based on the literature. The following paragraphs explain the assumptions that took place in order to end up with the final wear pattern of the bow-coupling bend pipe.

3.2.3. Assumptions

Corrosion is uniformly distributed in the entire bend pipe

From the literature review, it was clear that slurry transport causes mainly erosion in the pipes, especially in bend pipes where the flow changes direction. Impact wear is the most common mechanism in the bend and for that reason, it is considered the main factor to determine the maximum wear in the bend. Abrasion wear is not a frequent phenomenon to occur in bend pipes with slurry transport and this is the reason most of the projects do not refer to that separately, and they just consider erosion. Lastly, corrosion was found to accelerate the overall wear process in the pipe when acting simultaneously with other mechanisms. As pure corrosive wear does not show significant damage, the assumption that it acts uniformly and increases everywhere the wear in the bend, was taken. That means that corrosion does not constitute an indicator to find the maximum wear in the bend, as it is important only working with the mechanical wear mechanisms, specifically erosion.

Application of the literature result on the bow coupling bend pipe

Despite the large size difference between the research bend pipes and the bow coupling bend pipe, it is assumed that in both cases the wear pattern would be similar. In general, there are neither simulations, experiments, nor inspection measurements for large-scale bend pipes that transport slurry. As a result, this assumption had to be made in order to approximately define the expected wear in the bend. On the positive side, a lot of projects with different pipe diameters found very similar results.

Previous straight pipe has heterogeneous flow regime and medium wear

The wear of the previous straight pipe is applied based on the assumption that the flow is fully developed with a heterogeneous flow regime. In reality, the previous pipe is relatively short compared to models from the literature, because the discharging pipe coming from the hopper, usually ends at the bow coupling from the side of the vessel. If the distance between the bow coupling bend pipe and the previous elbow is short, a different wear profile might occur in the straight pipe, probably damaging the sides as well. For a heterogeneous regime, the maximum wear is at the pipe bed and decreases towards the top side. The bottom of the pipe can reach a medium wear rate, under special circumstances, as described in Step 2.

Additionally, Berg [4] mentioned that bend pipe wear may be twice as much for straight pipes. Most studies in literature did not find any significant wear in the upstream pipe but the reason for this can be that the particle concentration they use is much lower than in the dredging industry and some studies did not even consider the straight pipe wear to minimize the computational effort.

Medium wear for the first 30^o of the inner wall

The wear of the previous straight pipe continues in the bend pipe for the first 30^o, $\theta = 0^o - 30^o$, with a decreasing trend and having the maximum rate at $\phi = 180^o$. This assumption was taken based on the expected direction of the particles, as it was presented by Zeng et al.[67] and Zhang et al. [85].

Maximum wear shifts from inner to outer wall at 25^o

For $\theta = 0^o - 25^o$, the maximum wear is at $\phi = 180^o$, and for the rest of the bend pipe, $\theta = 25^o - 90^o$, at $\phi = 0^o$. It was decided to have the shifting point between the beginning of the medium wear at the outer wall, $\theta = 20^o$, and the end of the medium wear in the inlet wall, $\theta = 30^o$. Thus, $\theta = 25^o$ is the middle of the overlap of the two medium zones.

Wear rate increases along the outer wall

Wear rate increases throughout the bend pipe at $\phi = 0^o$. That means that from the inlet towards the outlet every next point has higher wear than the previous. This trend starts from the inlet, $\theta = 0^o$, and ends somewhere in the maximum wear zone, $\theta = 70^o - 90^o$, where there can be more than one peak. Apart from that, according to Peng and Cao [41], Zeng et al.[67], and Khan et al. [66], the wear rate of the outer wall is not linearly related to the bend angle. It is assumed that a similar pattern is followed for the bow coupling bend pipe. The wear continues to increase in the corresponding zones, and in every zone towards the outlet, the increasing trend is slightly higher. Specifically, at $\phi = 0^o$, each zone has a different slope of the wear rate, with the low wear zone having the most shallow one, and the maximum wear zone the steepest one. Thus, by taking one step back, one could say that the wear rate at $\phi = 0^o$ increases almost exponentially, from $\theta = 0^o$ to $\theta = 90^o$.

Maximum wear zone is five times as the low wear zone

As an average value from Peng and Cao [41], Zeng et al.[67], and Khan et al. [66], it was found that the maximum wall thickness reduction of the maximum wear zone is at least five times the maximum of the low wear zone. This comparison concerns only the wear at $\phi = 0^o$ and it is assumed that the same thickness difference will be applied to the bend pipe.

3.3. Conclusion

This chapter plays a vital role in the final design as it determines the wear pattern that can be expected for the bow coupling bend pipe. In the first section, a literature review takes place about the wear profiles in various bend pipes. Even though the projects used different flow conditions every time, the results tend to be similar. The general outcome of the literature was that the wear in the pipe is mainly on the outer wall, whereas the inner wall showed insignificant damage. Specifically, the wear rate of the outer wall increases along the direction of the flow, reaching the maximum wear rate at the outlet area. A more detailed approach is given by separating the bend pipe into wear zones, according to the expected wear rate. However, the fact that there is no available research on slurry wear in large-scale bend pipes led to the second section. There, the knowledge obtained from the previous chapter and the first section of this chapter were combined to adapt the literature outcome to the bow coupling bend pipe. Some special characteristics of the bow coupling and the lack of valid information in some cases made necessary the use of some assumptions. All in all, the chapter ended up showing the predicted wear pattern of the bow-coupling bend pipe, concluding that the first and the last 20^o of the outer wall will have the minimum and maximum wear, respectively.

4

Concept development

This chapter describes the process of generating concepts and then compares them to end up with the optimum one, according to the project's goals. As already described, currently the bend pipe serves two functions, flow and support, which lead to the high wall thickness of the bend pipe. The aim of this chapter is to find solutions that can reduce the replaced amount of material as well as extend the lifetime of the bend pipe. To achieve that, the number of loads acting on the bend pipe should be limited to the necessary ones, internal pressure and flow force, while other loads should be carried by other components. For this reason, it was decided that the two functions will be served by different parts. By applying this, all parts can be designed in a more dedicated way around their function, like using different materials, as the conclusion of Chapter 2 suggested. At first, the requirements and the limitations of the project are stated. Then, by keeping those in mind, various solutions are described for both functions, from which the best options take a place in the morphological chart. From there, various concepts are developed approaching the project from different perspectives, and then compared based on weighted factors. After evaluating the concepts, the best one is further discussed before proceeding to some design modifications that improve its performance.

4.1. Fixed requirements

This section explains the requirements, that should be covered by the new design.

1. Safety

All concepts should have as a priority the safety of the people, not only during operation but also during maintenance, inspection and replacement of any part.

2. Reliability

The system should be able to function under loading, without failure for the expected lifetime. The capability of the system and precisely the structural components to support the floating pipeline throughout the process is a requirement of great importance. The high pressure is another factor that all concepts should be able to deal with, for the solid parts and especially the connections among them. The connections can be either via bolts or a locking system, but not welded.

3. Flow redirection: 90-degrees

The bend pipe should redirect the flow from the horizontal axis coming from the tower, to the vertical axis leading to the female part/floating pipeline connection.

4. Inner diameter

The inner diameter of both pipes connected to the bend pipe, straight pipe from the tower and female part, is 1000 mm. Thus for a smooth pipe switch at least the ends of the bend pipe should be 1000 mm.

5. Separated flow and structural function components

The pipe components that are responsible for the flow of the dredged material should not be considered on the support function or affected by the support demands of the system.
Vice versa, the structural parts should not be considered for the flow function and they should not be affected by that.

6. Same hoisting and connection procedure

The connecting procedure with the floating pipeline remains unchanged so the existing floating pipelines can be attached. The new design should not affect any stage of the connection, from the hoisting cables/chains to the connection mechanism with the male connector part. Specifically, the modification of the female part is possible as long as it does not affect the locking of the male part, which will remain the same so all already existing floating pipelines can connect to the new system. Thus, if it is needed a different female mechanism can be suggested to lock the same male part.

4.2. Limitations

The limitations of the new design concern the available operational space. The first one is mainly about the position of the female part, which consequently restricts the position and the length of the bend pipe. The length is a very important parameter because as explained in Chapter 2, a larger bend curvature ratio could reduce the wear of the pipe significantly. The second limitation is there to make sure that there are no obstacles in the area that the hoisting cables pass, in order to lift the floating pipeline.

Limitation 1: Connection points

Particularly this limitation shows the distances and the orientations of the two ends of the bend pipe, see figure 4.1. These dimensions are standardized, based on the company guidelines and some of the reasons are explained. Firstly, the origin of those dimensions is the distance of the female part from the sea surface. This distance is determined by the characteristics of the floating pipeline, and specifically the maximum allowable bending stress. The connection to the female part on the one side and the buoyancy force on the other tend to bend the last part of the floating pipeline. As a result, the minimum distance of 7.6 m was should be applied for vessels with a 1 m pipe diameter. This distance is for a full vessel which is about to start discharging. This brings the second reason, which is the ease of connecting the male to the female part, as at least the floating body from where the pipeline is lifted should be vertical. Lastly, the larger the bend radius, the minimum the reaction forces on the bend are, and the lower the wear in the floating pipeline is.

Consequently, the position of the bow coupling is relatively high with respect to the deck level, depending also on the vessel design. This situation limits the height of the connection of the bend pipe to the tower, which cannot move higher. Initially, a higher tower with heavy pipes on top would raise the COG of the tower resulting in higher moments on the deck. Additionally, the discharging pipe approaches the bow coupling from the sides of the vessel, usually at the deck level. Thus, increasing the height would result in sharp angles on the previous pipes. In that way the problem is not solved, it is just moved to another position. One last point is that a higher tower at the bow of the vessel would also limit the visibility of the captain on the bridge. To sum up, the maximum bend radius is 1700 mm, for the highest and lowest connection to the tower and the female part, respectively.

Limitation 2: Hoisting cables space

The second limitation is about the space that should be free for the hoisting cable and chains. Figure 4.2 shows that with the pattern lines areas, the side views at the bottom and on top, the top view at the platform's level. Starting from this, at that level, the free space around the female part should be about 200^o, as the lifting pad eyes on the buoyancy body are exactly at 180^o. Moving to the bottom drawings, it can be seen that from the platform's level going up to the roller, the available space converges to the centerline of the female part and the diabolo roller. The final shape of this area is similar to a hollow cone cut in half vertically.

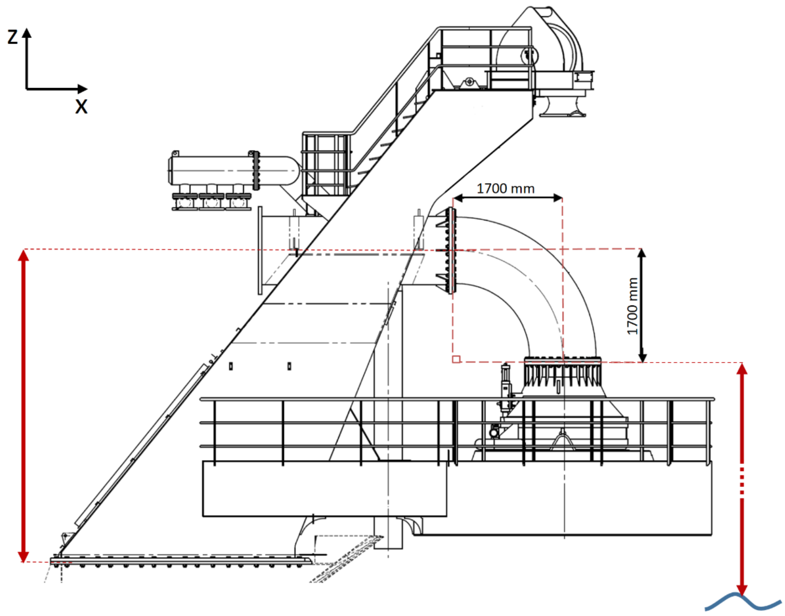


Figure 4.1: Limitation 1: Connection points.

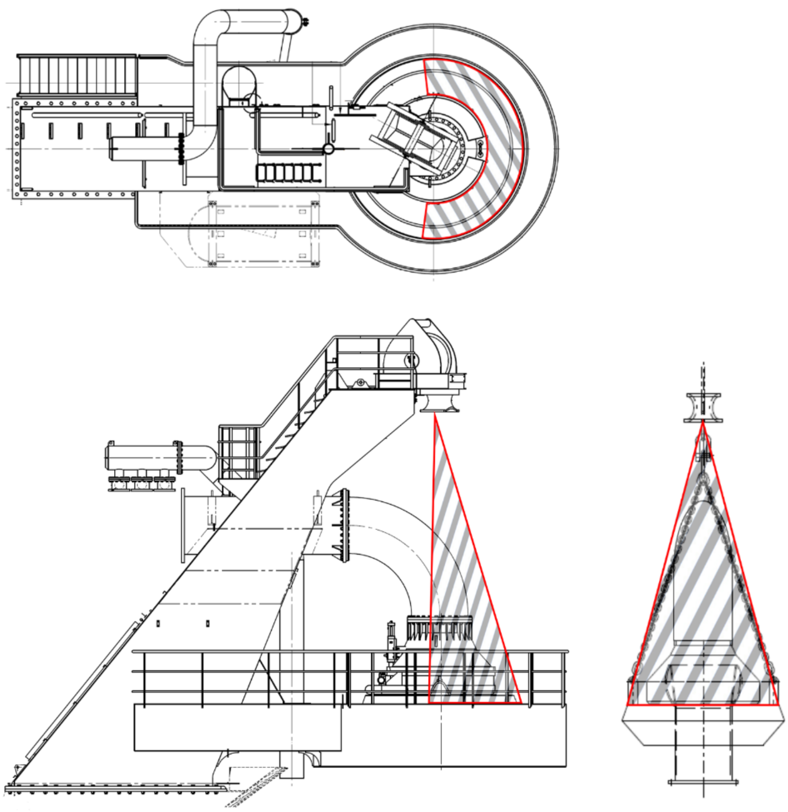


Figure 4.2: Limitation 2: Hoisting cables space.

4.3. Functions

For the conceptual design, the model is divided into two main functions. The support function is composed of permanent parts, that have to support the flow function parts, and they determine the overall operational structure of the model, and they are responsible to provide sufficient strength and stiffness to the system. On the other side, the flow function parts are replaceable and this is because they are responsible for the redirection of the flow and they are exposed to the extreme wear caused by the passing dredged material.

In the following subsections, all possible solutions are presented. Each sub-function can have various solutions, approaching the problem from a different angle. However, as each solution might have more than one options, a brief comparison is made to determine the most appropriate one for the morphological chart.

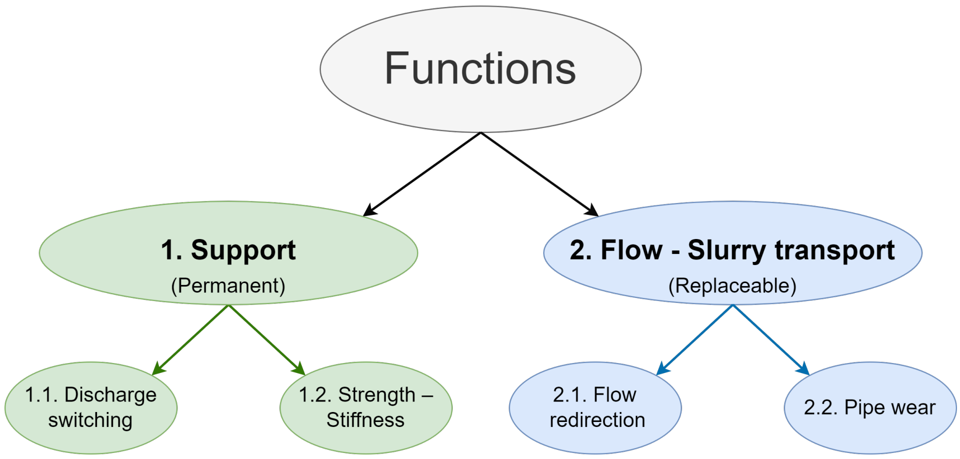


Figure 4.3: Functions.

4.3.1. Support structure

The first function concerns the overall structural design of the system, which at the same time consists of permanent components.

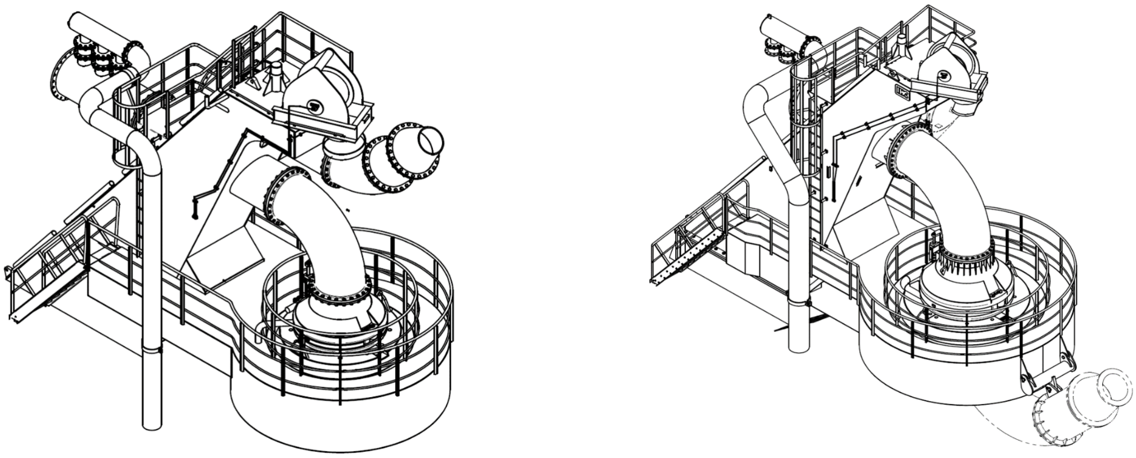
1-1: Support structure - Discharge switching

The first sub-function is of vital importance for the conceptual design because it determines the overall operational side of the system. Precisely, as shown in Chapter 1, there are two basic models for bend pipes, one with fixed connection and one with reconnectable bend pipe. In both cases, the bend pipe is bolted at both ends, with the tower pipeline and the female part, respectively. The difference between the two approaches is that the fixed bend pipe is always connected to the tower pipeline while the reconnectable connects only when should be used. The reason for that is that the latter arrangement requires the connection of the bend pipe and the nozzle on the same tower pipeline. Thus, always one of the two discharging ways is connected for discharging while the other one is in a sea-fastening position.

1-1-1: Fixed bend pipe:

As shown in figure 4.4, the fixed bend pipe can be used in two different arrangements, in terms of the nozzle position. The first option, which is the most simple and commonly applied design, has the nozzle in a fixed position, attached to the side of the tower, next to the bend pipe. For this concept, the discharging pipeline has to be divided into two pipes when approaching the tower to reach the nozzle and the bend pipe separately. Each of these pipes has a valve, through which the switching between the nozzle and the bend pipe discharging can be achieved.

The second option that can be used with a fixed bend pipe is a reconnectable nozzle. What is meant is that, even though the bend pipe is always in place and ready to discharge, the nozzle constitutes a separate body which is connected only when needed. This time the nozzle is locked in the female part with a similar connecting procedure as for the floating pipeline. The assistance of a secondary boat is necessary to lift and connect properly the nozzle to the female part, or the vessel can be prepared at the shipyard before going to the designated dredging area. This special nozzle is much larger than the regular one in order to redirect the flow by 135^o, instead of just 45^o. Also, the male part at the other end to fit in the female part makes the structure even heavier, thus additional connection to the platform is necessary to support it and reduce the resulting forces on the bend pipe. However, since the nozzle is connected to the female part, it requires only one pipe to reach the bow coupling and consequently connect the bend pipe.

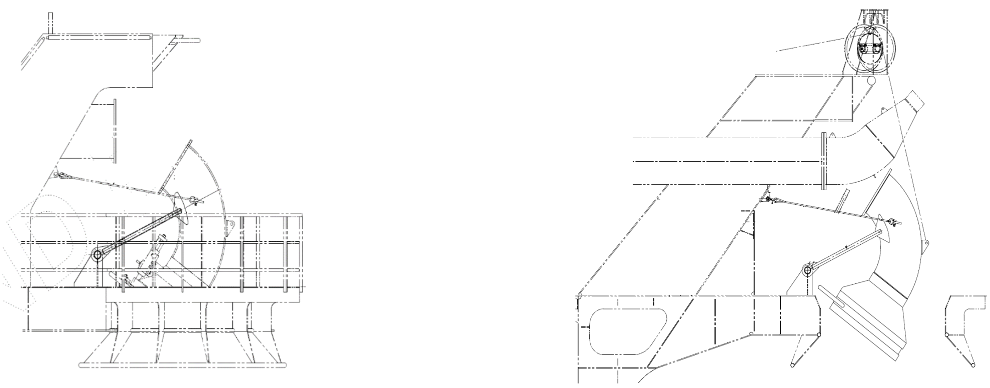


(a) Fixed nozzle (b) Coupling nozzle

Figure 4.4: Fixed bend pipe (1-1-1).

1-1-2: Reconnectable bend pipe:

Figure 4.5 shows two side view drawings of the reconenctabe bend pipe as it is applied on the Beagle series vessels of IHC. The left-hand side drawing depicts the sea fastening position of the bend pipe with the supporting cables coming from the tower, just below the tower pipe. The weight of the system bend pipe-female part keeps the tension of the cable, which in combination with the rotating arm forms a triangle to keep the system in place. The rotating arm is welded at the bend pipe at one end and at the other is connected to the platform with a large pin, and it is there to just lead the rotation of the bend without providing strength. The second drawing is a cross-section of the same view, which also includes the connected nozzle and the lifting cable coming from the diabolo roller on top. The lifting cable is connected to the pad eye at the outer surface of the bend pipe and counteracts the weight of the bend pipe-female part system to control the rotation. Hence, by keeping the COG away from the rotational axis the rotation of the bend pipe does not require an additional mechanism but just the already existing lifting cable. In reference to the nozzle, it has a pin connection close to the tower pipe flange which allows its rotation about the vertical axis to end up at the side of the tower where is locked for sea-fastening.



(a) Side view (b) Cross section with nozzle

Figure 4.5: Reconnectable bend pipe (1-1-2).

A deeper discussion and comparison of the three options is given in Appendix D. In any case, the location and operation of the nozzles are not affected by the conceptual design. To be more specific, for both concepts the focus is only on the bend pipe and only the modification of that is allowed. In general, the first sub-suction is divided into the fixed bend pipe and the reconnectable bend pipe and these options have a place in the morphological chart.

1-2: Support structure - Strength/Stiffness

This sub-function is responsible for the strength and stiffness of the system. The components and their arrangement should provide support for the bend pipe and generally all components of the other main function. There are no degrees of freedom allowable and thusly the strength and stiffness should be sufficient in all directions and loadings. Detailed information about loads on the bend pipe is given in Chapter 5. In any case, the inner pressure and the impact loads from the material flow cannot be avoided, but the rest of the external loads should be served by permanent support. The first two solutions are compact and support uniformly the bend pipe, while the last three options support the inner pipe wall, the outer pipe wall and the female part, respectively. These are basic concepts, while geometry and in general support arrangement are to be decided at the analysis phase.

1-2-1:Two pieces outer shell

This concept consists of two shells, which are connected via a flange connection, vertically or horizontally. The two pieces form exactly the outer shape of the bend pipe to keep the design compact but mainly to provide uniform support to the inner pipe. This solution has already been applied by IHC, at three pipe joints like T and Y branch pipes, which have high erosion rates and the special shape makes its manufacturability complex and expensive. The double-walled concepts are composed of an inner and an outer section, aiming to separate wear from the product's strength. In practice, the inner pipe in that case has the role of a liner, which is usually made of hard cast material and serves the wear factor. The two shells constitute the outer section, which is also mainly cast but with more ductile material since it provides strength to the system without coming into contact with the abrasive-erosive nature of the slurry transport.

For that concept, either cast bend or section bend or frame can be considered, keeping the same operational principles. The frame option has the important advantage of inspection, while the sections bend can offer a reduced cost. However, the cost of bend pipe composed of sections requires high manufacturing costs and a lot of working hours, and still, the accuracy of the cast model is very difficult to be achieved. It needs at least eight plates that should be initially cut, then curved and finally welded in order to add at the end all flanges and their bolt holes. The cast model only requires the last step of that fabrication process. Also, another important point is the theoretically pointy contacts on the inner bent pipe, which should gain be very precise in order to fit the inner pipe. These contacts act force at the bend pipe and the bend pipe may also cause a problem to the section bend since the contact point is exactly at the centre of the curved plate. Lastly, the non-uniform shape of the section bend leads to higher stresses and therefore thicker wall, which increases the required amount of material. The more the sections are, the better the support and the lower the stresses are, but with a much higher cost. The stated reasons led to the decision not to consider the section bend for that concept.

Generally, it is a simple and compact design which provides protection to the inner bend pipe from the external environment, like the hoisting cables and the chain which always slide on the bend pipe. Also, there is no risk of leakage in case the inner pipe comes to failure at an unexpected time. Lastly, it has similar loading conditions as the current design, which makes the calculation process easier and easily applicable on existing vessels with minor vessel modifications. On the downside, the total mass of the design is expected to be higher and together with the probably larger outer diameter are actually the two factors that might demand slight modifications on both pipe connections. The weight in combination with the support at the inlet flange makes the strength and stiffness goal more challenging. The main disadvantage of this concept is the lack of inspection access since the inner pipe is entirely covered, and therefore the bend pipe should be disconnected for inspection, otherwise, a new procedure should be proposed. Another drawback concerns the replacement, which requires the extra step of disconnecting the two pieces and placing the inner pipe, precisely.

- *Special point about horizontal connection:* The sides flange may obstruct the hoisting cables because it widens the pipe range.
- *Special point about vertical connection:* The bolt flanges on top and bottom contribute to the bending loads. An additional part at the top of the flanges can provide even more strength. Thus, when the two

flanges are connected the letter *T* is shaped. The vertical pieces are for the bolt connection while the horizontal top part is the very outer point of the bend pipe and it adds to the strength and stiffness of the system.

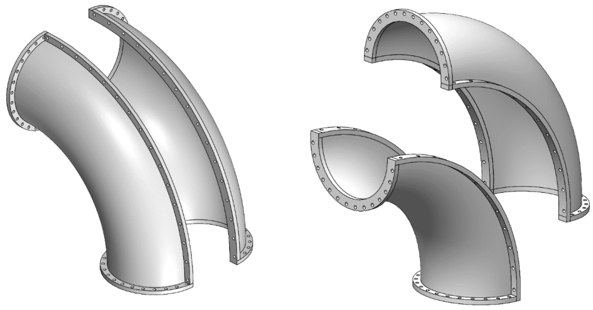


Figure 4.6: Two pieces outer shell (1-2-1).

1-2-2: Frame

The solution of the frame arises as a light design, which can provide the desired strength and stiffness to the bend pipe, allowing pipe inspection at the same time. The special part of this idea is that it can be applied in two ways, keeping in both of them the basic concept. An important point before moving to the possible options is the inability of the frame to be used with an inner pipe having the role of a liner, like for Concept 1.2.1. When using a liner, the connection with the other pipes is difficult to fit perfectly, thus there can be a local leakage. The use of outer walls prevents that phenomenon but the open frame design cannot do that. As a result, the use of a frame requires the connection with the other pipes of being by the bend pipe's flanges.

The first option is to not have the frame as a separate part, but as one solid part with the bend pipe. Specifically, the frame beams can be designed and considered during the casting of the bend pipe. Then the cast model is like the regular bend pipe with additional stiffeners on its outer surface. The positive side of this solution is the no need for additional support that has to be connected to the bend pipe. However, a cast bend pipe with extra stiffeners will increase significantly the cost because of the extra material and the complex geometry.

Another option would be to fit and then weld or bolt the bend pipe in the frame, which could be a solid, permanent part. However, the necessity of using the bend pipe's flanges makes the application of these concepts infeasible. This limitation brings the second option of using the frame, which is to split it into two pieces (like 1.2.1), thus the flanges of the bend pipe are not an obstacle, anymore. Each frame is composed of welded curved beams and there are two flanges used for the connection of the two frames. The frame can be made with relatively low cost and simple manufacturing ways. Another solution is to cast the frame to ensure the required accuracy. Lastly, the strength and stiffness depend on the arrangement of the beams, their sizes and their material.

To sum up, the only way to apply the frame solution is in two pieces, like the previous concept, with the difference of having the pipe connection flanges part of the bend pipe and not part of the support.

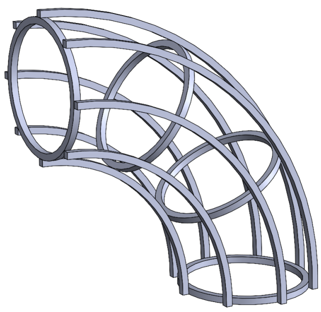


Figure 4.7: Frame (1-2-2).

1-2-3: Support the inner wall of bend pipe

The support of the inner wall is the first subfunction that considers the connection of the bend pipe to the surrounding members. The first idea considers the connection only with the tower whereas the next one extends that connection to the platform, as well. The support should most probably be welded on the tower and the platform while the connection to the bend can either be welded or bolted. If the bend pipe is welded, the crew has to cut the welded point and then machine the support to make it ready for the next one. However, for a bolt connection, only a simple plate structure has to be welded on the bend pipe in order to bolt the support on that. When it has to be replaced only the unbolting of the bend pipe is enough to remove it and then bolt the new one, with the welded plate. The challenging part is to find the ideal connection point or points to the bend pipe in order to minimise the stress on that. A larger connection is better to minimise the stresses on the bend pipe but it affects the inspection access to the pipe. In any way, the loads on the bend pipe are unavoidable with this concept and the best it can be achieved is to remove loading between the support and the tower, the tension before and the bending at the support. Even in that ideal scenario, the strength between the support and the female part relies only on the bend pipe. The fact that the maximum erosion, and consequently the maximum thickness reduction, is expected to be at the bend outlet generates a lot of issues, which should be checked in the calculation stage if it is selected.

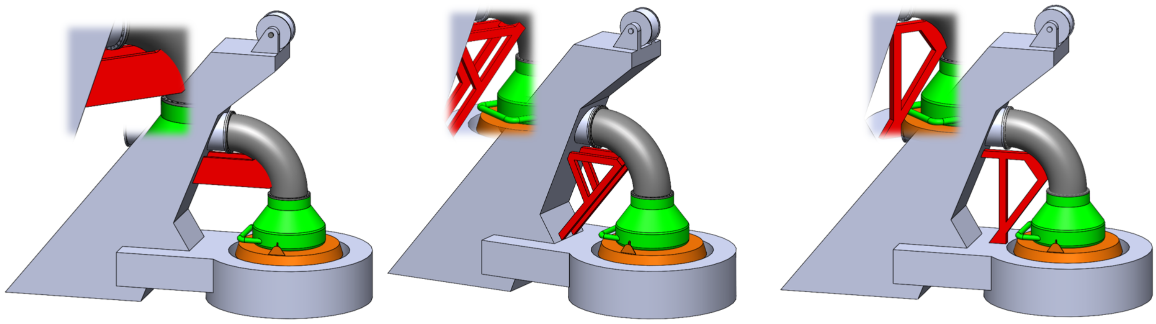


Figure 4.8: Support the inner wall of bend pipe (1-2-3).

1-2-4: Support the outer wall of bend pipe

A similar approach to the previous solution, with the difference of supporting the bend pipe from the top side, and therefore the support is under tension instead of compression. The first idea uses just a plate, which connects the tower to the bend pipe. Both connections have the option of being either bolted or welded, with the latter one making the replacement extremely difficult, time-consuming, dangerous and costly. The other two options from figure 4.9 share the same operational principle, a connection arm under tension with two pin connections at both ends. The second option is made of plates and the last one is made of rods or even steel wires. Even though for under-tension structures the cross-section area plays the most important role, the steel wires solution might bring installation difficulties, in case pretension is required. In terms of loading, as mentioned before, a large connection area causes lower stresses on the surface but the inspection is more difficult. For all concepts, the bend pipe is exposed to a lot of loads, especially the tension after the connection point until the female part. Also, the concentrated pulling from the supports should be considered. In reference to the last two options, the position of the pin-joint at the bend pipe determines the load that the lifting will induce on the pipe surface. If the hypothetical extension line of the connection arm does not pass between the two ends of the support pad base, a moment is generated, which tends to rotate the pad eye anticlockwise, for the presented case. Moreover, the high vertical load that the connection arm has to support, generates a great horizontal force component at the connection point with the bend pipe which compresses the part between the connection point and the tower.

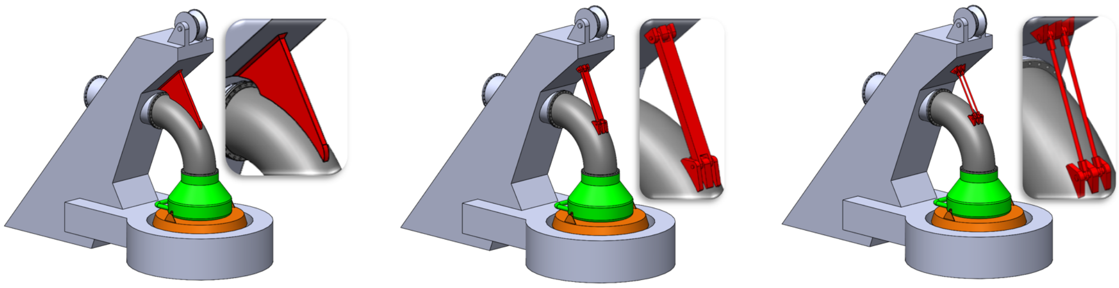


Figure 4.9: Support the outer wall of bend pipe (1-2-4).

1-2-5: Support of the female part

The connection to the platform looks like an expected solution that comes with a lot of advantages, but it has challenges, as well. Starting from the positive side of this solution, it would certainly offer good support to the bend pipe since it supports the very end of that, the female part. Even though there is no direct connection between the platform and the bend pipe, the support of the female part in terms of strength and stiffness, leads to limited loads left on the bend pipe. To put it in a more simple way, the side view of the system now looks like a triangle whose two ends are not connected, and then this connection comes to close the triangle and make it stiff. A good connection can provide strength and stiffness to the bend pipe, considering that the bend weight, female part weight, hoisted pipeline weight and any other forces generated from the floating pipeline are held by the new connection.

On the downside, the available space for such a connection is limited since the floating component of the male part floater has a very large diameter and its lifting pad eyes are at exactly 180^o. As mentioned in the limitations of the project, only the 160^o on the tower side can be used. The two connections have the additional role of preventing the lifting cables and chains to go further behind, towards the tower. The positive sign of that is the protection of the bend pipe from the hoisting chains, which usually slide on the bend pipe's sides. Also, the lowest point of the female part is higher than the top point of the platform, so that the crew can watch the connection.

However, there are two points that lower the practicality of that solution. First of all, the platform may not have adequate strength to support the weight of those large parts, especially at that great distance from the tower. Currently, the platform is only being used as lead for the floating pipeline to end in the female part and the top side is for the crew to have access to the bow. The level of required support will be determined in the calculation process if this concept is selected. The second disadvantage of that concept is the interruption of the walking corridor at the platform, because of the limited space. For new vessels, this can be considered during designing the vessel and simply expand slightly the walking space locally.

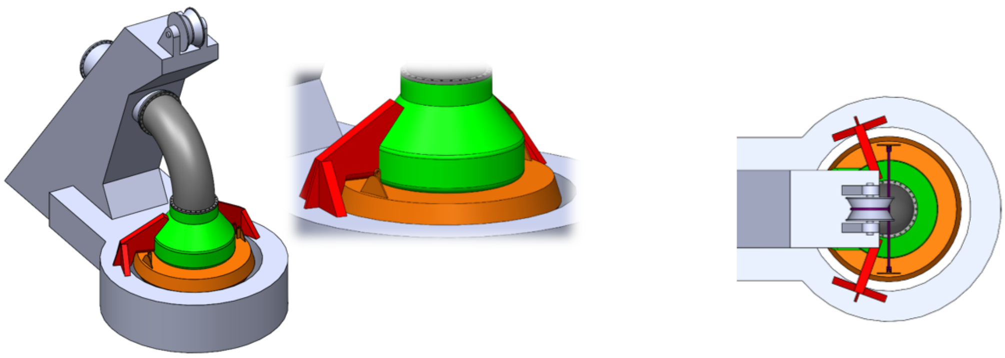


Figure 4.10: Support of the female part (1-2-5).

4.3.2. Flow

The second function refers to the transport of the dredged material from the pipeline coming through the tower to the female part and subsequently the floating pipeline. As explained in the previous two chapters, the direct contact of the bend pipe components with the slurry transport makes them vulnerable to wear. For that reason, they are expected to require much more frequent replacement compared to the structural parts, and therefore they are considered replaceable parts. The first sub-function discusses the possible solutions for the basic design of the bend pipe, while the second sub-function focuses on the wear in the pipe.

2-1: Flow redirection

The first sub-function aims at the redirection of the flow, from a horizontal straight pipeline, 90^o downwards with the bend outlet to be concentric with the female part. There are two basic options for that purpose, the regular cast bend pipe and the sections bend pipe.

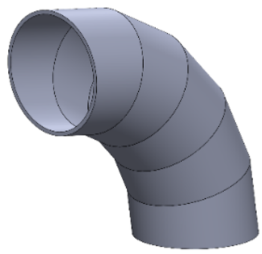
2-1-1: Regular bend pipe

The first solution, is the regular cast bend pipe, just like the current design. The cost of the bend pipe depends on the wall thickness, the material, the geometry complexity and any fabrication at a later stage if needed, like flange holes or support connection. In general, cast components are expensive due to the accuracy and the ability to create complex solid parts whose manufacturing would be very difficult.

2-1-2: Sections bend pipe

A bend pipe composed of several steel sections is an alternative solution to use for the redirection of the dredged material flow. Some of the main characteristics of sections bend have already been discussed for the support function. Here, it is proposed as a mode of redirecting the flow, which means that it has direct interaction with the slurry transport and its behaviour during flowing.

(a) Regular bend pipe (2-1-2)



(b) Sections bend pipe (2-1-2)

Figure 4.11: Flow redirection options

2-2: Flow - Pipe wear

This is a crucial sub-function because it has to deal with the wear in the pipe, which is the main reason that leads to the frequent replacement of the pipe. The three solutions that have a place in the morphological chart are described in this section. The first two can be used in the form of a single bend pipe, like the current design, or as a liner in another pipe. Liners are the inner pipes whose function is mainly for wear protection and are not considered in the strength of the system. Usually, liners do not have any load acting on them and can reach zero thickness, before being replaced. Inevitable loads for the bend are the internal pressure and the impact load from the flow, which can cause its failure at a very low thickness.

Moreover, another category of wear solutions that could be included in the morphological chart is the one with a divided bend pipe. To put it in another way, this category considers a bend pipe consisting of more than one part, and when one of them reaches the minimum wall thickness is replaced, while the rest continue operating. Additionally, this approach can be more specific and allow the replacement of only one part which is expected to be eroded the most and therefore has a more sacrificed role. For that, a very concentrated erosion area is required, which is not the case for the current project as shown in the previous chapter. Nevertheless, even though this kind of concepts saves material, they can only have successful results under very specific circumstances. Their main disadvantage is the non-uniformity of the wall thickness on the bend pipe, after every replacement. The new part will have more thickness compared to the non-replaced neighbouring ones, which most probably have a much lower thickness, especially at the connection with the new part. As a result, the connection areas will generate intense turbulence of the flow, which correspond to a higher wear rate.

Overall, any solution that requires the replacement of a specific part or parts of the pipe, is not considered in the morphological chart.

2-2-1: Wall thickness arrangement

The first solution aims at the wall thickness arrangement according to the wear rates in the pipe. Thus, the conclusion of the literature review plays a vital role in that solution. As discussed in the previous chapter, the maximum wear in the pipe is expected close to the outlet, on the outer wall, whose inlet area is foreseen to have the least thickness reduction. Considering this situation, a concept would be the increase of the thickness along the bend pipe, from the inlet to the outlet. However, as the inner wall did not show any significant damage, like maximum or minimum, it is preferable to not involve it in the wall thickness arrangement, to avoid the extra complexity of the bend pipe shape. This part can only be cast to achieve the desired wall thickness distribution, and the more complex the geometry is the higher the cost of the cast bend pipe will be. Even though the main indicator for the cost of a cast bend pipe is the required amount of material, because it is a very large part, also the geometry has an influence up to a certain degree. Hence, the inner wall will have a constant thickness and the outer wall will have the same thickness at the inlet and with an increasing trend, the outermost point of the outlet will have the maximum thickness. For example, a bend pipe with an inlet thickness of 20 mm and a maximum of 35 mm at the outlet of the outer wall, can save about 35% of the amount of material that the current design requires, which corresponds to roughly 1 t. Alternative arrangements are possible but deeper analysis with thickness will take place if it is selected.
Figure 4.12 shows an illustration of how this solution works, indicating the thickness difference along the outer wall, where the initial thickness t_{1b} is higher than the rest of the pipe.

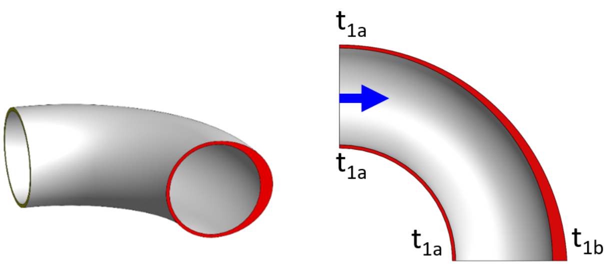


Figure 4.12: Wall thickness arrangement (2-2-1).

2-2-2: Flip bend pipe

This solution considers a regular cast bend pipe with constant thickness, just like the current case. The special point about this idea is the simple way of contributing to the better management of the wear in the pipe, by decreasing the amount of replaced material and increasing the pipe's lifetime. The answer to that is to flip the bend pipe after a certain point, instead of replacing it with a new one. The idea arising from the outcome of the literature is that the maximum and minimum wear in the bend pipe is expected to be measured at the outer wall, close to the outlet and close to the inlet, respectively. The two extreme situations at the outer wall seem to complete each other and the flip of the bend pipe will bring the inlet with the minimum thickness reduction to the outlet, where the wear is maximum. Conversely, the outlet which will have the maximum thickness reduction will be placed at the inlet where the wear is the lowest of the pipe. Still, this solution comes with some challenges, like the connection points with the corresponding pipes and the determination of the new expected maximum wall thickness reduction. The rotation might cause significant wall thickness differences with the previous and the next pipe while reusing an already eroded pipe might bring some uncertainty in reference to the final minimum thickness location. Nonetheless, the severity of those challenges depends on the final concept and therefore further consideration is to be given to the entire concept, in case it is selected for this sub-function.

Figure 4.13 presents the steps that the bend pipe should follow, showing at the same time the thicknesses at the ends of the bend pipe, where $t_1 > t_{2a} > t_{2b}$. Starting from the left side, this is the initial pipe with uniform thickness, while the next step shows the reduction of the wall thickness in the pipe. When the minimum thickness is reached at the bend outlet (t_{2b}), the pipe should be flipped, so that the outlet area with the lowest thickness, is connected to the upstream pipe. The thickness t_{2b} constitutes the flipping thickness and its value plays a vital role in the total thickness reduction and the lifetime of the pipe. It can be seen that after flipping the pipe there are a lot of similarities with the previous concept, in reference to the increased thickness along the outer wall.

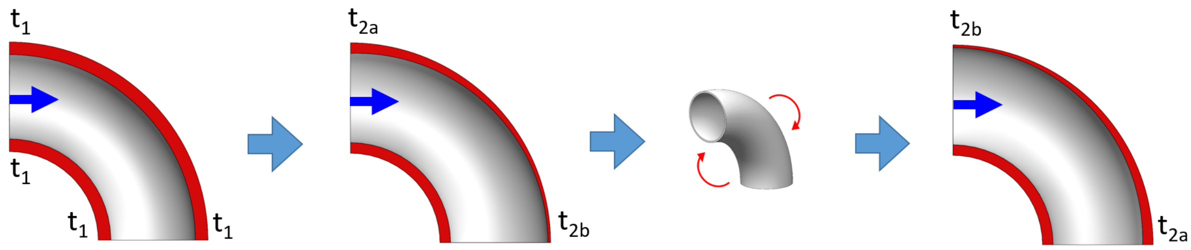


Figure 4.13: Flip bend pipe (2-2-2).

2-2-2: Inner wall covering (chocky bars)

The third alternative is to cover the interior of the wall, either with coating or chocky bars. The latter is much more applicable in the dredging industry compared to the coatings, which are rarely even mentioned as a solution. The reason for that is the incapability of the coatings to be a cost-effective solution for the wear reduction of the slurry transport pipes. However, it is a promising technology and there have already been steps towards that direction, developing various materials and deposition methods, which might find a widen application in the future[122–126].

In reference to the chocky bars, they are utilised in a number of applications to give an extra layer of protection in highly abrasive and erosive environments. It is a solution that exclusively focuses on the lifetime extension of the surface on which it is welded. The working principle for these bars is to act as a harder material than the actual product is made of. They are made of bimetallic material produced by metallurgically bonding a highly alloyed chromium molybdenum white iron to a mild steel base plate. The alloy's hardness provides excellent wear resistance, while the steel base plate absorbs the high impacts and enables simple installation and use [127].

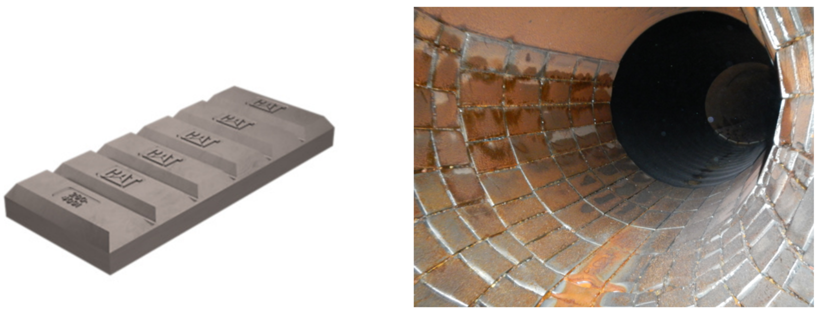


Figure 4.14: Inner wall covering - chocky bars (2-2-2), [5].

4.4. Morphological Chart

Figure 4.15 presents the Morphological chart, which gathers all solutions discussed in the previous paragraphs. The aim of this chart is to group all options that can be used to generate a concept. The solutions are separated according to the function and sub-function they serve. In order to build a concept, at least one solution from each row has to be selected, allowing for several possible concepts.

Before developing the concepts, it was decided to approach the two functions separately. What is meant is that the solutions for the two sub-functions of the flow function are compared and their best combination of flow redirection and pipe wear solutions, is applied to all concepts. The reason behind that decision was to focus on the support function where much more solutions can be unfolded, considering that some of them can even be combined. On the other side, the flow function has limited proposed solutions, among which some have significant advantages over others, that may even be infeasible under certain circumstances. Consequently, the use of the same flow function solution for all concepts results in a more objective comparison of the support function solutions.

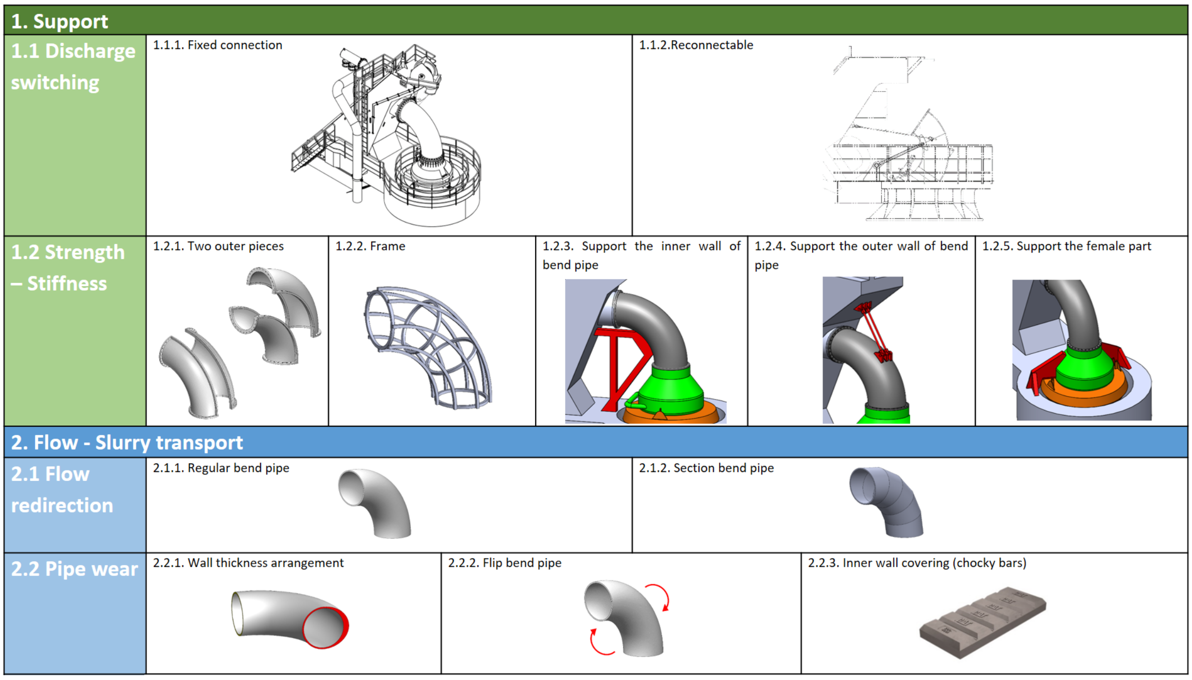


Figure 4.15: Morphological Chart

The solutions of the two sub-functions are briefly compared below, but a detailed explanation is provided in Appendix D.2. The main parameters for their comparison are the production cost, service lifespan, material and recycling.

Flow redirection: The two options are the regular cast bend pipe and the sections bend pipe. Firstly, the sections' bend pipe affects the flow and specifically worsens the wear rate in the pipe, and the location of the maximum thickness reduction cannot be predicted. Also, the cast bend pipe can be made of harder material for better wear performance, something which is difficult for a sections' bend. Brittle materials would have a high risk of failure under all those fabrication steps, and therefore special treatments had to be adopted to achieve the desired shape. The only positive side of the sections' bend pipe is the lower manufacturing cost but the aforementioned drawbacks outweigh that parameter. Thus, the regular cast bend pipe is preferred over the sections' bend pipe.

Pipe wear: There are three solutions about the wear in the pipe, the increased thickness, the bend flip and finally the chocky bars. To begin with, the last solution is a straightforward method for extending the lifetime of the bend, as the bars are worn down before reaching the pipe wall surface. Chocky bars are made of very hard and expensive material which also needs to be welded in the pipe. Welding of straight bars in a bend pipe requires numerous working hours and it does not result in a smooth surface, which is expected to affect the flow in the pipe and probably the wear profile. Also, the replacement requires a lot of machining of the inner surface of the pipe to make it appropriate for the welding of the new ones.

The other two solutions are significantly cheaper compared to chocky bars and they can always be recycled, as well. The first solution increases the thickness of the outer wall along the bend pipe while the second option flips a regular bend pipe at a certain minimum thickness, so the inlet side goes to the outlet and vice versa. However, the casting of those pieces requires a minimum wall thickness of 25 mm. Also, one of the assumptions for the pipe wear pattern was that the outlet has five times the thickness reduction of the inlet. The last two points make the first solution practically infeasible because the maximum thickness at the outlet of the outer wall should be exceptionally high, to achieve a uniform thickness reduction for the outer wall.

Overall, the optimum solution for the flow function is the regular cast bend pipe (2.1.1), which has to be flipped after a certain point (2.2.2). This combination is applied to all concepts.

4.4.1. Concepts

After the composition of the morphological chart, it is possible to define the different concepts by combining the partial solutions. While materialising a certain combination of partial solutions into a concept design, engineers often discover unexpected advantages, problems, and possibilities. As explained in the previous paragraphs, all concepts use the same solution for the flow function. Thus the generated concepts differ only in the solutions of the support function, and in that way, more support solutions can be considered. Given the fact that more than one solution of the sub-function "Support-Stiffness" can be combined, there is an endless number of possible concepts. However, the combination of solutions should be done wisely, so that all concepts have fundamental differences. In the end, eight different concepts were developed, which approach the system from a different perspective. There are also projects that although share similar principles, they focus on different areas leading to different strong and weak points. Only the last two concepts consider the option of reconnectable pipe, while the first six concepts use the fixed bend pipe. From those, the first three concepts focus on simplicity and applicability, whereas the next three provide strong and stiff structures.
In the following paragraphs, the eight concepts are described, providing also the part solutions from the morphological chart.

Concept 1

The first concept is one of the most simple and compact solutions. It is a fixed bend pipe, where the support is provided by half shells. In this case, the bend pipe has the role of a liner, meaning that it is only used for the flow and the resulting wear. As a liner, the bend pipe is not connected to any other pipe, but only at some contact points with the support shells. The main advantage of this concept is the simplicity, safety and lastly that it can be easily applied even on existing vessels, with minor modifications. However, the inspection of the bend pipe is an important drawback, and the initial cost of the two cast pieces can be high. High strength and stiffness can be achieved by increasing the thickness of the support shells, which on the other side will increase the total weight. The replacement procedure is not very complex, but it requires the extra step of disconnecting the shells, in relation to the current case.

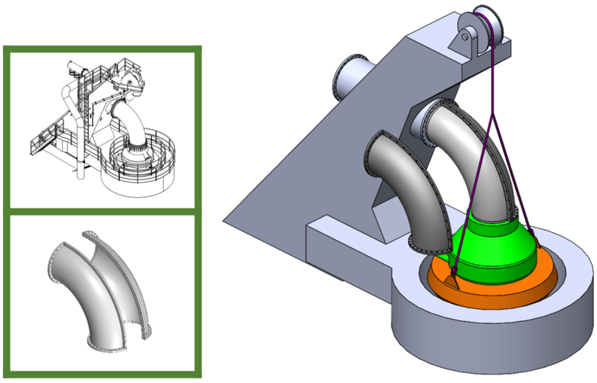


Figure 4.16: Concept 1

Concept 2

The second concept uses external support for the bend pipe, specifically from the top side. This might be the cheapest solution, as the support can just be composed of wires or beam elements that are constantly under tension. The installation might be a problem as precise pretension should take place before connecting to a new bend pipe. Apart from that, for every new bend pipe, some fabrication should take place in order to connect it to the support, regardless of the connection type. The fact that the bend pipe is fully exposed to the outer environment is good for inspection purposes but risky from a safety perspective. In contrast to the first concept, the bend pipe is not a liner and clearly has significant loads exerted on it. Additionally, the stiffness of the support is very low, especially under loading acting on the side of the bend.

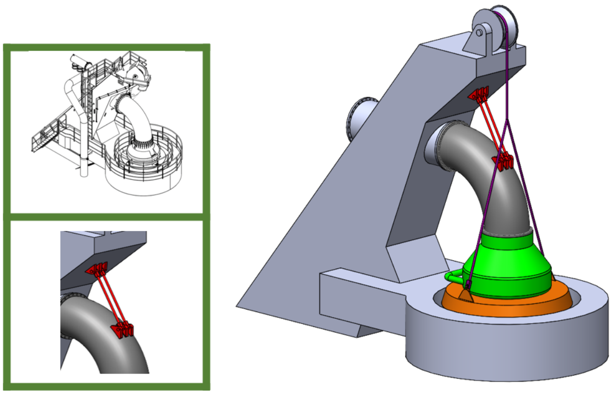


Figure 4.17: Concept 2

Concept 3

As can be seen, the third concept uses the same main support as the previous one, combined with two more solutions from the "Support-Stiffness" sub-function. The thought behind the new concept was to provide additional strength and stiffness to the system. The frame was selected as an option which could still allow the inspection from the outside. However, as explained before, the frame can only be applied as two connected pieces, like the first concept. This concept offers better and more uniform support on the bend pipe, but at the same time increases the cost and complexity of the system. The fact that the flanges are part of the bend pipe, means that the bend pipe is the only connecting link between all components and consequently has significant stresses. Lastly, the replacement procedure is similar to the first concept and the bend pipe.

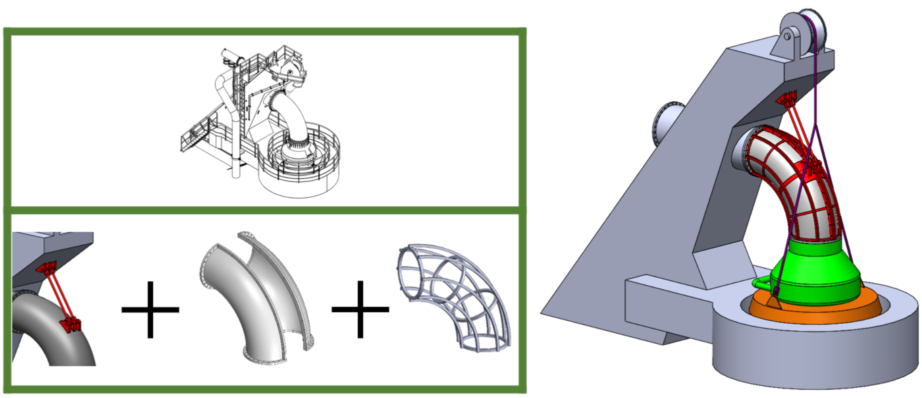


Figure 4.18: Concept 3

Concept 4

The general idea of this concept is similar to Concept 2, with the difference of supporting the bend pipe from the side, the inner wall. Thus, most of the comments about the second concept apply to this concept, as well, and only major differences are mentioned here. Compared to that, the larger and more expensive support structure can result in a relatively stiffer and stronger system. Also, the connection area can be larger as the inner wall is expected to have relatively low wear and therefore its inspection is not essential.

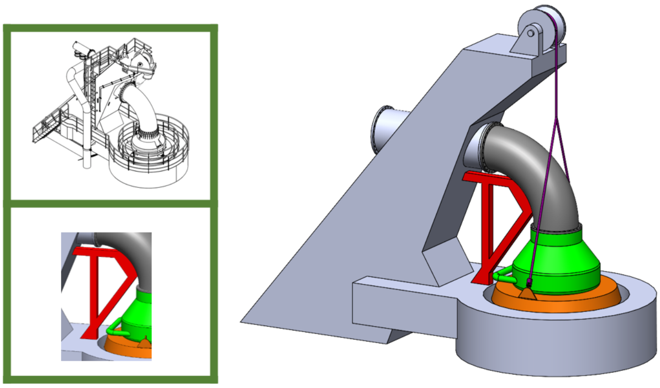


Figure 4.19: Concept 4

Concept 5

For Concept 5, two sub-function solutions were combined, the support of the inner wall, like the previous concept, and the two outer pieces connected horizontally. The reasons that led to this concept are safety, bend pipe as a liner and lastly the replacement of the liner. Firstly the outer shell prevents leakage in case the liner fails and secondly the liner does not provide any strength to the system, and there are limited loads exerted on that. The selection of horizontally connected pieces has to do with the replacement procedure. Specifically, the bend pipe can be replaced by removing only the top piece of the support. The same procedure was considered for the inspection of the liner, as the bend pipe's outer wall is the important one to be measured. However, the scenario of disconnecting the top piece of the support should be analysed to check whether it is feasible or not. Otherwise, temporary support for the female part should be applied. In general, this concept is considered to offer the most strong support for the bend pipe. The strength of the individual supports is reduced compared to Concepts 1 and 4, in order to avoid unnecessary costs, as it is already quite expensive.

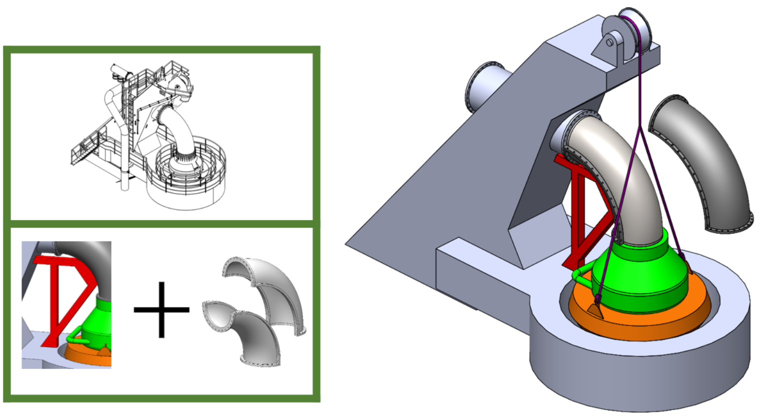


Figure 4.20: Concept 5

Concept 6

This is one of the two concepts that support the bend pipe, by supporting the female part. It is expected to be the stiffest concept because it supports directly the female part, on which the loads from the floating pipeline are applied. Moreover, it is easily approachable for inspection at any point of the bend pipe, since it is one of the concepts that do not have any connection to the bend pipe. Also, the connection of the female part makes the replacement of the bend pipe very convenient, because the female part can remain on the vessel and remove only the bend pipe. Yet, like Concepts 2, 3, and 4, there is no safe to prevent leakage, and the bend pipe connects the female part with the tower pipe. Another issue is the connection with the platform, a part which is designed to lead the floating pipeline in the female part and to allow people to walk around the female part. Thus, the platform might require to be redesigned to withstand the loads at the connection points, which have a great distance from the tower, where the platform is supported. Furthermore, there is limited space for the two support structure and as a result, a part of the walkway should be used. In order to make the platform a safe working environment for the people, the platform should be modified accordingly.

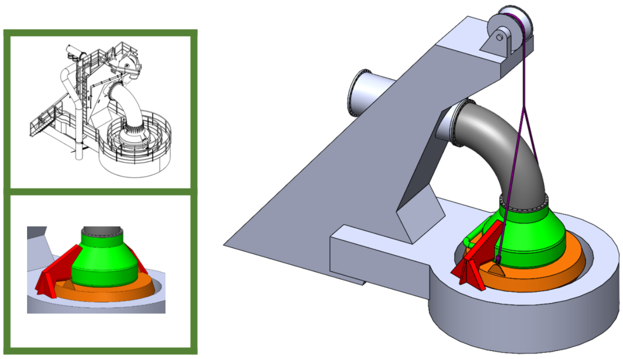


Figure 4.21: Concept 6

Concept 7

As can be observed, Concept 7 has a lot of common points with the previous concept, with their main difference being the use of reconnectable bend pipe. An explanation of the differences between fixed and reconnectable bend pipes is provided in Section 4.3.1 and Appendix D. The rotational connection results in more loads exerting on the bend pipe compared to the previous concept, which had a fixed connection. Apart from those characteristics, the seventh concept has some additional challenging parts that require further investigation. Precisely, the problem with this design is that the centre of gravity of the system bend pipe-female part is very close to the axis of rotation. As a result, a mechanism should be added to the system in order to control the rotation of the bend pipe about the rotational axis at the connection with the platform. The installation of a mechanism increases dramatically the overall cost of the concept. A possible solution would be to install a hydraulic cylinder on the tower and connect the other side to the bend pipe. Deeper consideration is to be given in case this concept is selected.

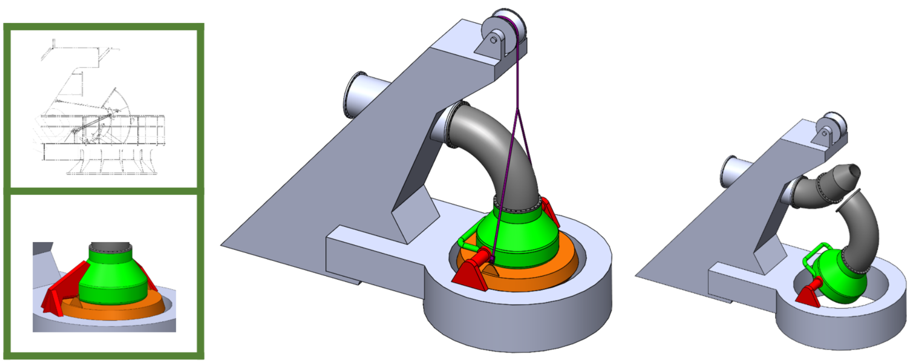


Figure 4.22: Concept 7

Concept 8

The last concept adopted the operational principle of the original reconnectable solution and added the two pieces shell for support. The reason behind the horizontally connected pieces is the same as for Concept 5, replacement and inspection, something that should be analysed to check its feasibility, or the support of the female part is necessary. In contrast to the previous concept, this design does not require any additional mechanism to rotate the bend pipe, but only the gravitational force and the hoisting cable. This concept has a lot of similarities with the first concept, and their main difference is about the switching between the two discharging ways, bend pipe and nozzle. Other than that, the pieces are connected horizontally and the bottom piece is connected to the lead rotational structure, which is not considered in the strength of the system.

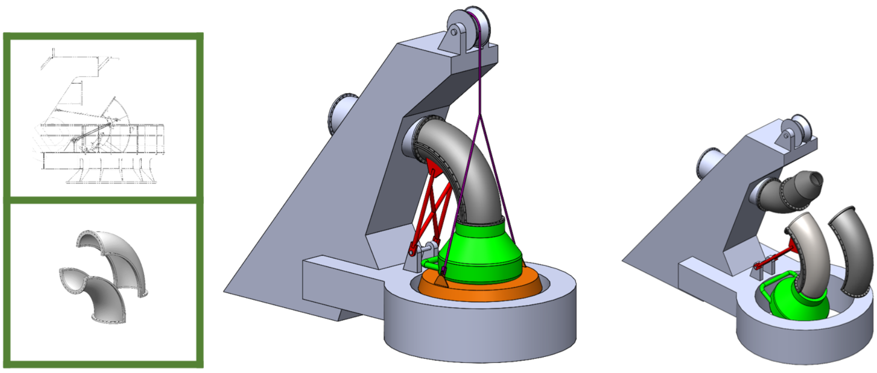


Figure 4.23: Concept 8

4.5. Concept Selection

After generating those concepts in the previous section, the next step is to compare them to end up with the optimum one, according to the project's priorities. In this section, the variable requirements are explained, based on which the eight concepts are compared.

4.5.1. Variable requirements (weighted factors)

The weight of every requirement was decided after a discussion of the author with company experts, who stated the priorities of the company for the project. The variable requirements help to make the differentiation between the eight concepts. With the variable requirements, different criteria are defined. Not all criteria matter in an equal way to the re-design. So, they are placed in order of importance and a score is given to each criterion.

Cost of flow/replaceable parts (weight: 5)

The first variable requirement refers to the cost of the flow function components. As has already been explained, those parts are to be replaced on a more frequent basis because of the erosive nature under which they operate. Specifically, the cost is determined by the amount of material, manufacturing process and additional fabrication to make the parts appropriate for installation. At his stage, the comparison is under the assumption that all bend pipes have the same material.

The cost of the replaceable part, as it was expected, is the most important variable requirement for the project and therefore it has a maximum weight of 5.

Ease of replacement (4)

The replacement procedure of the replaceable parts constitutes another very important parameter for the selection of the final concept. The number of steps, the required equipment, the fabrication that has to take place and the ease of connection are the main parameters for a replacement. One could say that all these are translated to cost and that they should be included in the first weighted factor. However, it was decided to keep them separately for a more in-depth comparison and for that reason the cost of emplacement was not considered in the cost of replaceable parts.

Even though the replacement constitutes a very important parameter for the new design, it has a weight of 4, as its influence is closer to the next weight group.

Cost of structural/permanent parts (4)

This is the cost of the support function of the project, permanent parts that are responsible for the support of the replaceable parts. Similarly to the first requirement, the main parameters are the amount of material, the manufacturing cost and lastly the type of support, including any special characteristics of that. Moreover, the structural complexity as a whole, new and existing structure, plays an important role in the cost, since entirely new standard models have to be developed and analyzed, thus the simpler the better.

The fact that these parts are permanent and no regular replacement of them is needed, put this requirement on a lower level of importance, 4.

Loading on bend pipe/replaceable part (4)

The aim of this requirement is not to highlight the concepts with which the loading on the replaceable bend pipe is the lowest possible, but how easily this can be achieved. The separation of the two functions is already one of the two solutions that this project follows to decrease the replacement cost, thus the final concept will do that to the greatest degree. As explained before, the primary reason for the separation is to replace less amount of material, and that can only be achieved by minimizing the loads on the bend pipe so that less thickness is required. The higher the strength and stiffness of the support, the less the minimum allowable thickness. For example, the current design has a minimum thickness of 24 mm, for the reason that there is no support contribution from any other part.

Moreover, at the same time, there is another reason which makes this variable requirement even more critical, and this is the material type. As discussed in Chapter 2, harder materials proved to have a longer lifespan under impact and sliding wear. Nevertheless, in order to use a more brittle material, the loads on that should be minimized greatly because of the dynamic loading that is exerted on the system. The ideal scenario would be the combination of both advantages of isolation of the bend pipe, resulting in a low minimum thickness, with an abrasion/erosion resistance material.

The reduction of the bend pipe loading will be examined at a later stage. Here the focus is on how easily the isolation of loads on the bend pipe can be achieved. The benefits of this requirement make it highly important and it is represented by the weight of 4.

Bend pipe inspection accessibility (3)

The inspection of the bend pipe constitutes another important parameter as the bend pipe is exposed to a severe environment, which is practically the root of the frequent replacement issue. The extensive wear of the bend pipe necessitates its inspection on a regular basis, or at least more regularly than other pipes with lower wear rates. Consequently, access to the bend pipe is important so that the inspector can measure the wall thickness of the bend pipe properly, via an ultrasonic sensor. The measuring device should directly contact the bend pipe, perpendicularly to the wall, to provide accurate results. Lastly, the inspection process takes place only when the vessel is out of service, at the shipyard or dock. They avoid inspection on board for safety purposes since in most cases external platform or crane is needed to lift the inspector to the desired position or to disconnect the inspected part, respectively.

For the inspection of the bend pipe, the use of external modes is necessary since the limitation of the hoisting cables deprives the installation of an inspection platform close to the bend. Another reason for that is the replacement of the bend pipe, which should be lifted from that area. Thus, these two limitations are the reasons that only a temporary platform should be able to connect/disconnect easily. However, in order to move a platform the use of external lifting means is necessary and for that reason, this concept is rejected at this level.

Even though it is an important task for the smooth functioning of the bend pipe, and consequently, the entire system, its less contribution to the main project goal places it at a lower importance level with a weight of 3.

Required support on the existing structure (3)

As discussed in the requirements of the project, even though the focus is on the bend pipe, the surrounding parts can also be included in the new design, to connect on them the bend support structure, if needed. However, only slight modification of them is allowed in order to adjust them on the new design, without though violating the limitations. The aim of this variable requirement is to give attention to the strength of the existing structure, to which the bend support is connected. According to the connection part and the location of that part, different levels of support might be necessary. Despite the fact that this requirement could be considered in the cost of the structural parts, it was decided to keep it separated to emphasize only the existing structure. Hence, the overall cost of the required support, like extra material, fabrication and new model design and analysis, is considered only for this requirement.

The weight of this requirement is 3, as it is less important than the previous ones, considering that it is a new design that is expected to bring changes to the current concept.

Safety - Protection (2)

The requirement of bend pipe protection covers more than one area. It has to be clarified that safety is already one of the most important fixed requirements of the project and in the end, the system should definitely be safe enough for people. In this case, safety has to do with safe walking areas for people, moving parts of the system and whether people are involved in general. At the same time protection is important mainly to prevent leakage since an unexpected failure of the bend pipe can put the crew's life in danger. At the same time, if the leakage risk is minimized, also less minimum thickness of the bend pipe can be an option for an extended lifetime. The last factor that comes with the word "protection" is of minor importance and concerns the protection of the bend pipe from the outer environment, and precisely the hoisting cable/chains that slide on the bend pipe sides.

The weight of this variable requirement is 2, because, for safety, solutions and operational instructions can be proposed while the regular inspection of the bend pipe makes pipe leakage an extremely rare scenario.

Ease of applying on existing vessels (1)

The ease of applying a concept on existing vessels is the last variable requirement. The interesting of this requirement is that it more or less sums up the scores from previous factors like 3, 6 and 7. For an already operational vessel, the cost of the permanent parts is more important and the challenge of adapting the vessel for the new design can be very high. Hence, modifications on the vessel to make it operational and the overall cost of the new design determines whether it is applicable to existing vessels.

However, the weight of this requirement is the lowest, 1, because the aim of the assignment is to find a solution for a better and more cost-efficient future, and therefore its applicability on already operating vessels is just another positive point.

4.5.2. Concepts rating

The best concept is the one that ends up with the highest score compared to the other seven concepts. the previous section explained the eight factors that represent the priorities of the new design. Thus, a weight value was given to each one to determine the different levels of importance. At this stage, the concepts are evaluated for each of those defined factors, on a scale of one to five. Score "5" corresponds to the highest performance of a concept for a specific factor, while "1" is the lowest. The next step is to multiply all scores with the corresponding factor weights. The final rate for each concept is the summation of those eight values. As there is no detailed analysis of the concepts for each requirement, the ratings are mainly based on estimations. In order to reduce the evaluation uncertainty, it was decided that experts in the field had to be involved in this procedure. Their experience and knowledge of the subject provide an objective and practical perspective on the concept selection. Hence, four professionals from the company Royal IHC, provided their own scores for each concept. Table 4.1 presents the final rates, which were taken as the average of the five different people that scored the concepts. The five individual tables can be found in Appendix E, as well as the reasoning behind every score mainly from the author's point of view, considering also the experts' feedback.

The first row of the red boxes shows the summation and consequently the final score for each concept, while the last row of the table compares those rates with the optimum score, 130. As can be seen, Concept 1 had the highest score, followed by Concept 6 and Concept 5, respectively. Before discussing the features of the final concept, sensitivity analysis takes place, since the difference between the first three concepts is very low.

Table 4.1: Concepts' average rates.

Variable Requirements	Weight	Concepts 1	2	3	4	5	6	7	8
Cost of replaceable parts	5	5.0	3.2	3.2	2.6	5.0	3.2	3.2	5.0
Ease of replacement	4	3.4	3.6	2.4	2.8	3.4	4.6	4.4	2.8
Cost of structural/permanent parts	4	3.8	4.8	3.0	4.2	3.2	3.8	2.8	2.2
Loading on bend pipe/replacebale part	4	3.8	2.4	3.6	3.2	4.8	4.2	3.8	4.2
Bend pipe inspection accessibility	3	1.6	4.2	4.2	4.2	1.6	5.0	5.0	1.8
Required support on existing structure	3	5.0	3.8	3.8	3.4	3.4	2.8	2.6	3.2
Safety - Protection	2	4.8	2.8	4.0	2.8	4.2	3.0	2.4	3.4
Ease of applying on existing vessels	1	5.0	4.0	3.4	3.2	3.0	2.0	1.8	2.2
Summation:		103.4	92.8	87.4	85.4	97.0	97.8	89.4	85.8
		79.54 %	71.38 %	67.23 %	65.69 %	74.62 %	75.23 %	68.77 %	66.00 %

4.5.3. Sensitivity analysis

A brief sensitivity analysis was performed in order to define how the final decision is influenced by the change of weight for some factors. Such a scenario can occur in case the company changes its priorities. The first step was to increase the factor, where the final model has the lowest score. By increasing the inspection's weight from 3 to 5, then Concept 1 comes second and Concept 6 has the lead with the tinny difference of 1.2. Another scenario was to switch the first two requirements so that the replacement is more important than the cost of replacement. Even though Concept 6 scored better than Concept 1 for the ease of replacement, the first concept remained the best one, with 101.8, 2.6 more that Concept 6. In general, Concept 6 received high scores at factors that the first concept scored low and vice versa. Specifically, both concepts had the same score for the cost of the permanent parts, while for the rest one of them was better and sometimes with a significant difference. As a result, a higher weight of a requirement, in which Concept 6 got a better score, can possibly show it as the best one as well. This is what happened in the case of "Bend pipe inspection", but as proved for "Ease of replacement", this is not always the case.

On the other side, although, Concepts 5 and 6 have almost the same final score, only Concept 6 seems to be able to overcome Concept 1. The reason for this phenomenon is that Concepts 1 and 5 share the same strong and weak points, and with only the exception of "Loading on bend pipe", Concept 1 has higher scores for every requirement. However, even when the gravity of "Loading on bend pipe", becomes 5 instead of 4, Concept 5 remains in the third place, while Concept 1 just reduces slightly the space from Concept 6.

Overall, by changing up to two values at a time, the only scenario that finds Concept 6 as the best one, is when the weight of "Bend pipe inspection" increases to 5. Other than that, the two concepts were found to have the same score when "Required support on existing structure" was reduced to 1, while "Ease of replacement" increased to 5.

4.6. Selected concept

Among the eight proposed concepts, the first one concentrated the highest score. Specifically, Concept 1 is composed of a fixed bend pipe that uses two half cast pieces for support. For the flow function, a cast bend liner is used which has to be flipped once, before being replaced. Figure 4.24 shows the final concept with the selected options from the morphological chart. The support pieces are disconnected and the liner can be seen with pink colour.

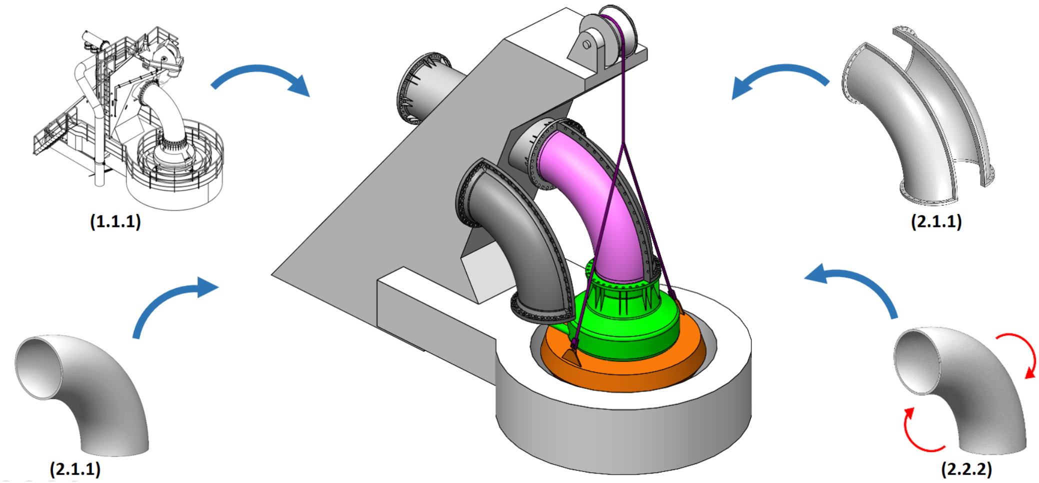


Figure 4.24: Final concept.

Generally, this concept has a lot of similarities with the current design, with respect to the loads on the surrounding components. Both systems find support only on the discharging pipe coming from the tower. On the positive side, the analysis of the system is almost the same, but on the other side, the support has a great distance from the female part, where most of the loads are applied. Also, compared to the current system, the new design is expected to be heavier. Considering the last two points, the resulting stresses at the connection of the discharging pipe with the support pieces, are expected to be high. At the same time, the two flanges that connect the two pieces, contribute significantly to the strength and bending stiffness of the new design. Furthermore, this system provides remarkable safety since the liner bend pipe is practically isolated, protecting in that way the liner from the outer environment, and the crew from leakage in case the liner pipe comes to failure. The use of liners is a common practice for dredging pipelines in order to protect the structural parts from slurry wear. As already discussed the liner is a regular cast bend pipe, which should have a minimum thickness of 25 mm. This is already a thick liner compared to the thicknesses that are usually preferred. A larger liner does not only increase the mass of the liner but also of the entire support system, which has to be enlarged in order to fit the liner. Another point for thinner liners is to avoid noticeable thickness differences in the pipe, which might affect the flow conditions. Therefore, there is no need for a thicker liner, and 25 mm is the final thickness for this cast part. The same thickness is applied for the other two cast pieces which are responsible for the support of the system. However, the final geometry and materials of both models are determined in the next chapter, where the analysis takes place using finite element methods (FEM). The analysis will also show the minimum operational thickness of the liner, which constitutes a vital parameter for the replacement as well as the flipping of the pipe. Thus, a detailed explanation of the flipping process is provided in the next chapter.

4.6.1. Inspection plugs

As shown from the rating table, the inspection of the bend pipe constitutes the main weakness of the selected concept. The solution of inspection plugs increases the cost of the permanent structure, affecting the final score of the selected concept. Nonetheless, even if the corresponding variable requirement decreases by one point, Concept 1 remains in the lead, and even increases the difference by adding points to the factor about the inspection. The support pieces deprived the bend pipe of being inspected from the outside, like the previous design. Thus, the only solution is to disconnect the bend pipe from the vessel the proceed to its inspection at the shipyard or any other maintenance point. There, experts can inspect from the inside or the outside by disconnecting the support pieces. Meanwhile, the inspector can have access to the bend pipe every time the female part or previous straight pipe is being replaced, and when the bend pipe should be flipped. It must be noted that the inspection of the connected parts is also from the inside because they use liners, which indeed are covered the same way. Only the female part has the advantage of being the last part of the pipe system and as a result, someone can inspect from the inside without disconnecting any other pipe. Thus, by taking advantage of that convenience, at least the bend pipe outlet can be inspected properly while the rest of the pipe can be visually inspected. However, the aim of inspection plugs is to provide access from the outside at certain critical points so the disconnection of the bend pipe is not always required. The number, location and size of the inspection holes are analysed in Appendix D.3.

Overall, after flipping the critical points are much more than the initial orientation. A total of eight inspection plugs was decided, with only one of them being at the inner wall, and precisely at 0^o after flipping. The outer wall requires at 70^o and 80^o before flipping and after that at $0^o, 10^o, 20^o, 36.67^o, 53.33^o, 70^o, 80^o$. The only non-symmetrical spots are at the inlet after flipping. Therefore, in order to avoid disconnecting the support pieces during rotation, the side with the two extra holes will be bolted to the female part first. Fig. 4.25 shows the final locations of the inspection plugs before and after flipping the bend pipe.

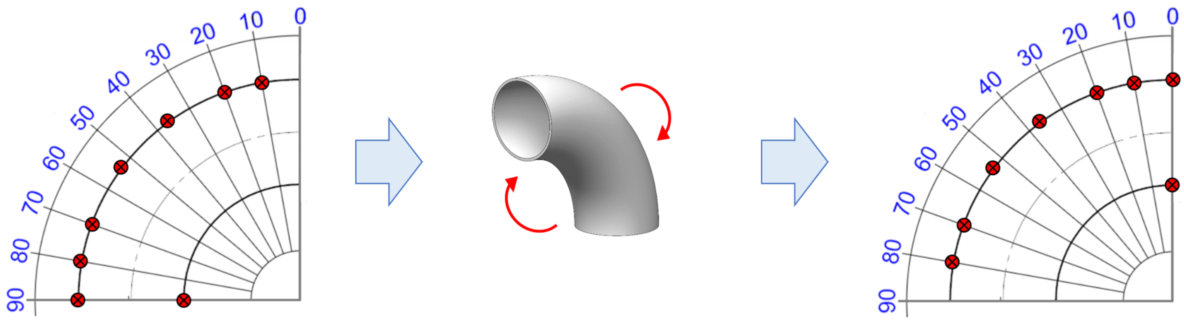


Figure 4.25: Inspection plugs: Final locations before and after flipping.

4.7. Conclusion

The aim of this chapter is to develop a new design of the bow coupling bend pipe that focus on the reduction of the replaced amount of material and the extension of its lifetime. Initially, the limitations and the requirements of the bow-coupling application were explained. One of the requirements was to split the system into two models, which consider the flow and the support functions separately. Specifically, it means that the components that are responsible for the flow of the dredged material should not be considered on the support function or affected by the support demands of the system. Based on the two functions and their sub-functions several options were suggested, from which the best ones were included in the morphological chart. Then, eight different concepts were developed, which were rated according to their performance in several variable requirements of different gravity. Finally, the highest score was concentrated by Concept 1, which is composed of a fixed bow coupling system with two support pieces, in which the bend pipe is placed and has the role of a liner. The chapter ends with the determination of eight inspection plug locations, in order to have access to the liner for inspection.

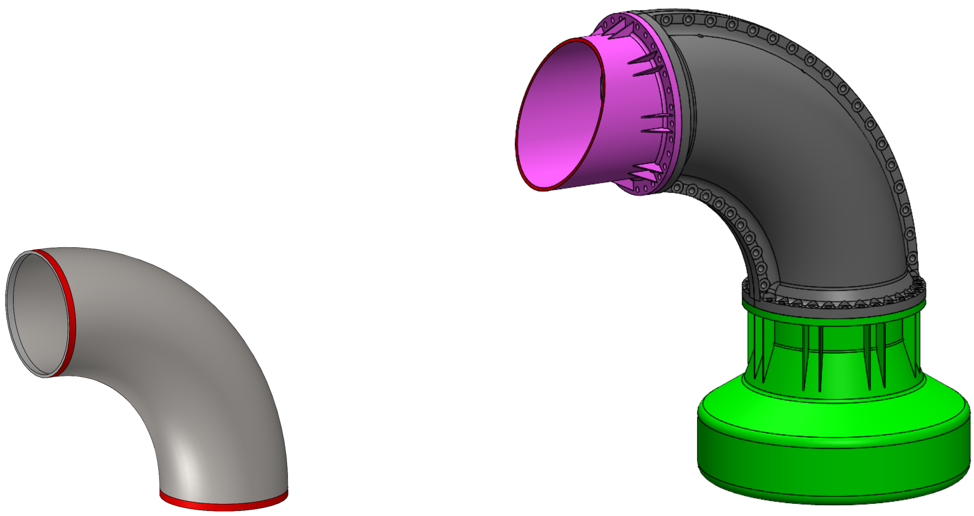
The separation of the two functions has a double impact on the project. First and foremost, the minimum operational thickness of the pipe can be significantly reduced because the bend pipe does not need to support additional high loads anymore. That corresponds to a decrease in the replaced amount of material, which is one of the main objectives of the project. The second benefit of this solution is that a harder material is possible to be used in order to reduce the wear rates in the pipe and therefore prolong its lifetime. At the same time, the last category of the morphological chart collected solutions, which can extend the lifespan of the pipe. After comparing the three options, it was decided that the most effective and practically feasible solution is to flip the bend pipe once at a certain moment before replacing it. This solution takes advantage of the previous chapter's conclusion, that the minimum and maximum wear in the bend are expected at the inlet and outlet of the outer wall, respectively. Overall, the separation of the two functions contributes to the reduction of the replaced amount of material, while a more wear-resistant material and the flipping solution can lead to an extended lifetime.

5
Proposed design

This chapter discusses the feasibility of the new design in terms of its capability to withstand real-life loading scenarios as well as its performance against the project's goals. The first section concerns the analysis of the new design, and it begins with the two models that are considered for the simulations. Then, all loads exerted on the system are explained, and their final values and applied locations are shown for each model. Before proceeding to the analysis using FEM, the required load cases are stated according to standards. Lastly, having the final parameters of the new design, a comparison with the current design takes place, in reference to the three aspects of the thesis objective.

5.1. Design analysis

At first, the new design is divided into two models as shown in figure 5.1. The first model is the inner bend pipe, the liner, and the second one is the support of the bend pipe. The aim of the first model is to determine the minimum operational thickness before the bend pipe is replaced. On the other hand, the second model consists of the straight pipe, coming from the tower, the two support pieces and finally the female part. Despite the fact that only the support pieces are of interest to the project, the previous and next parts are included to create more realistic conditions.



(a) Model 1

(b) Model 2

Figure 5.1: Analysis models

Even though the two models have some common loads, their support is different. Starting with the first model, the only contact that the liner has is with the support pieces. Generally, when a liner is installed in the support pipe, small plates are placed around the liner, between the two pipes in order to align it, see Appendix F.4. These small plates, around the straight part of the inlet and the outlet of the bend pipe, constitute the support of the liner. The final support areas are shown in figure 5.1a with red colour, which is constrained in all degrees of freedom.

With regard to the second model, it finds support on the only part that is connected to the vessel, the straight pipe which passes through the tower. Hence, the end of this pipe is selected as the constraint point with no degrees of freedom, see figure 5.1b. Another worth mentioning point about the second model is the connection between all parts, which is defined as a bolt connection on the corresponding flange's holes. In that way, the stresses and deformations show a more realistic result, compared to the approach of assuming the whole system as one part.

5.1.1. Loads on the system

In general, the loads acting on the system are due to gravity, internal pressure, external loads from the floating pipeline and lastly environmental loads from wind, current and waves. The following paragraphs explain the loads and where they act on each model.

Pressure

The pressure in the pipes would ideally be applied only on the inner bend pipe, providing that there is no leakage which requires its application on the support pieces, as well. However, the connection of liners between two different pipes is not designed in a way to prevent the flow of the mixture, to reach the support pipes. When this happens the first scenario is that only a small amount of the mixture will pass through the connection gaps and block them immediately. Otherwise, if the space is large enough, the leaking will continue until the mixture is spread everywhere, between the liner and the support pipe. Considering the first situation, the pressure should be applied only on the liner, whereas for the second scenario, the most possible one, the pressure should be applied only on the support pieces. Nonetheless, both situations are considered and therefore a total pressure of 25 bar shall be applied on the inner wall of both models. Apart from that, in Model 2, the pressure should also be applied at all grooves, which are necessary to place the o-rings along the flange connection, see figure 5.2. The special polymer material is compressed between the two flanges to prevent pressure loss.

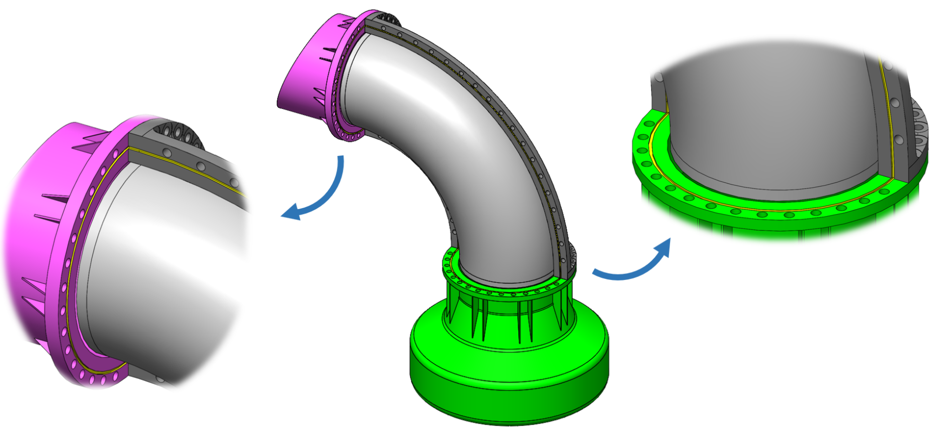


Figure 5.2: Connection grooves.

Floating pipeline force

The connection of the floating pipeline on the female part constitutes the most significant load of the bend pipe, and it is actually the reason for the increased thickness of the bend. For the finite element analysis (FEA), it was decided to simplify the system by analyzing only support pieces, the tower straight pipe and the female part. To do so, some loads have to be applied on the female part to represent the hose behaviour. Firstly, the load of the floating pipeline hanging on the bow connection acts in the centre of the ball joint.

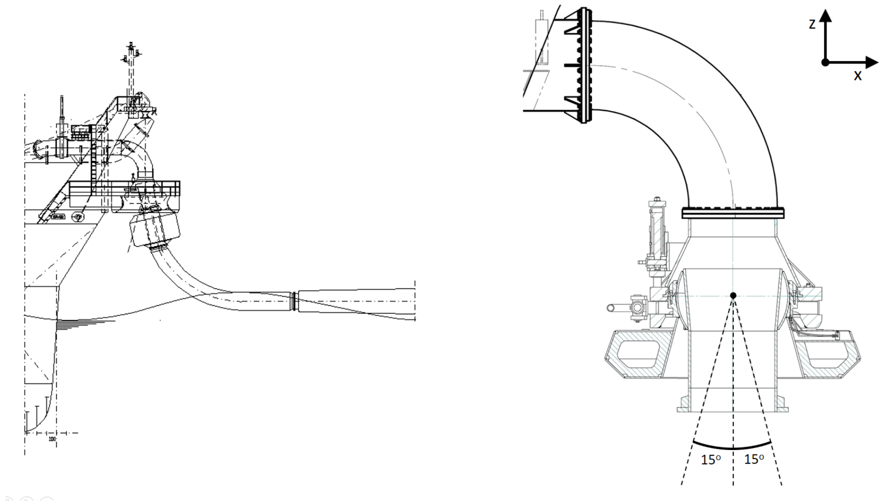
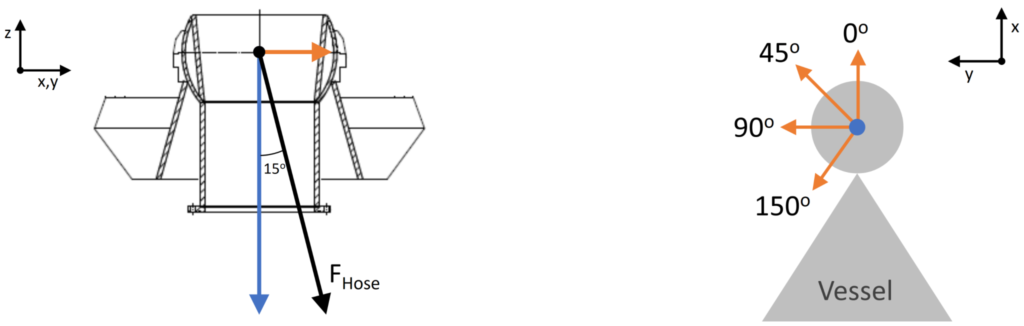


Figure 5.3: Ball connection.

The magnitude of the load depends on the properties of the floating pipeline and the environmental conditions during the operation of discharging to shore. Specifically, this dynamic load fluctuates with the vessel motions, the waves and current acting on the hose, and lastly the buoyancy, due to its capability to remain on the sea surface even when it is full. Thus, the company determined a load as a rule of thumb, according to the internal diameter of the bow connection. The F_{Hose} for 1 m diameter is 1 MN, and this load should be applied in various directions, as the ball connection allows for a free movement of 15^o from the vertical axis. By taking advantage of the symmetry about the z-x plane, four directions of the floating pipeline are considered to be enough to prove the bend's strength capabilities. Figure 5.4 shows the side and top view of the bow coupling, with the four angles to apply the F_{Hose}. The maximum angle is 150^o as the bow of the vessel is about 60^o.



(a) Side view

(b) Top view

Figure 5.4: Floating pipeline force

In reference to the area where the F_{Hose} is applied, there are two parameters that have to be taken into account. The first one is the location of the centre of the ball connection, while the second one has to do with the locking system of the male part. In practise, the F_{Hose} is supported by the two half rings that are used to lock the male part into the female part. However, usually, there is a tiny distance between the top side of the half rings and the centre of the ball connection, depending on the female part design. Thus, this difference can be neglected, and specifically for this design, the load is applied 1010 mm below the bend pipe outlet. Furthermore, a special part was designed on the female part to represent the two half rings, on which the F_{Hose} shall be applied as distributed load, see figure 5.5.

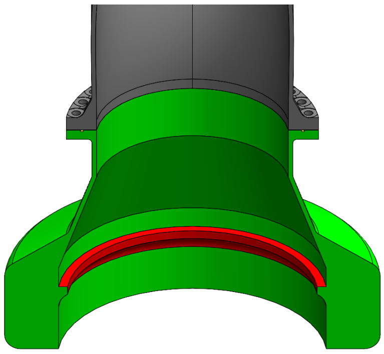


Figure 5.5: Application area of floating pipeline force.

In the same way, as for the bend pipe, the flow of the dredged material in the hose generates a vertical and a horizontal load, see next paragraph. Their magnitude is the same as for the bend pipe, provided that the angle is 90^o and the inner diameter is 1000 mm. In that case, the horizontal load is in the opposite direction as the pulling force of the hose which is indeed much higher, and therefore the horizontal component can be neglected. The pulling force is responsible for the hose angle at the connection with the female part. In relation to the vertical component, as it is collinear with the one acting on the bend pipe, but in the opposite direction, both loads could be deleted as they cancel each other out. However, since the F_{Hose} includes all loads from the hose, including the vertical component of the flow force, the vertical flow force on the bend should be applied as well.

Flow force

The next load for the system is generated from the redirection of the flow, resulting in a momentum force exerting on the bend. Based on the conservation of momentum, the rate of momentum change of a body is equal to the net force acting on the body. For a flow in a 90^o bend with a constant cross-section area, the change in momentum is caused by changing the velocity's orientation, rather than its magnitude. Thus, two force components are resolved, one for each direction, meaning that for the current case, the two forces are perpendicular. As shown in figure 5.6, the horizontal component, coming from the inlet, hits the outer wall at about 41^o, while the vertical component, at about 49^o. It should be noted that the forces are distributed and not pointed, and thus one could imagine them as the straight extension of the previous pipe for the horizontal force and the next pipe for the vertical force. For the calculation of the force exerted by the flow on the bend's outer wall, the mixture properties are required. As discussed in Chapter 2, the maximum concentration and the corresponding density of the mixture during unloading is 0.48 and 1800 $\mathrm{kg/m^3}$, respectively. At the same time, the volumetric rate for a velocity of 6 m/s in a pipe of 1m inner diameter results in a total of about 4.7 $\mathrm{m^3/s}$. With this information, the flow force can be calculated as shown below. It should be noted that the thickness reduction due to wear is not considered so as to keep the maximum velocity to result in a higher force.

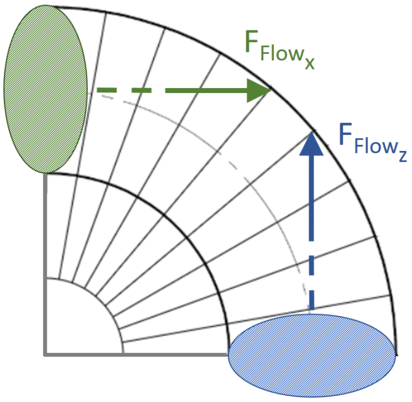


Figure 5.6: Flow force.

- General equation based on Newton's second law, for constant mass flow rate and flow area:

$$F_{Flow} = m \cdot a = \dot{m} \cdot (v_1 - v_2) = \varrho \cdot A \cdot (v_1 - v_2)^2 \tag{2}$$

$\dot{m}$ = mass flow rate (kg$/s$)
ϱ = fluid density (kg/m^3)
A = cross section area of the pipe (m^2)
v_1 = inlet flow velocity component (m/s)
v_2 = outlet flow velocity component (m/s)

- Horizontal flow force:

$$F_{Flow_x} = \varrho \cdot A \cdot (v \cdot cos(\theta_1) - v \cdot cos(\theta_2))^2 = +5103541N \tag{3}$$

- Vertical flow force:

$$F_{Flow_z} = \varrho \cdot A \cdot (v \cdot sin(\theta_1) - v \cdot sin(\theta_2))^2 = +5103541N \tag{4}$$

θ_1 = Inlet angle = 0^o
θ_2 = Outlet angle = -90^o

As was expected the two perpendicular forces are equal, because of the 90^o redirection of the flow. Also, those two loads shall always be applied together, and for that reason the term F_{Flow} is used, which represents both of them. Figure 5.6 illustrates the direction of the two forces acting on the bend pipe. Both loads are distributed on a circular area of 1 m diameter, like the inner diameter, and they should be applied on the outer wall. Even though, the flow forces act directly on the liner, these loads are considered for both models as the only support of the liner is the outer shell.

Winch force

During connection and disconnection of the floating pipeline, the winch has to lift the male part in the female part at a higher level than the operational so that the two half rings in the female part can close and lock the male part. Hence, if the operator accidentally lifts the floating pipeline more than it requires to be locked, the floating body will push the bottom of the female part upwards. The maximum possible force is the difference between the upwards winch force and the downwards, empty floating pipeline weight. The maximum pulling force from the winch is 450 kN, while the floating line mass is about 10 t.

$$F_{Winch} = 450\,\text{kN} - 10\,\text{t} \cdot 9.81\,\text{m/s}^2 = 351.9\,\text{kN} \tag{5}$$

The resulted upwards force is 351.9 kN, and it shall be applied as a distributed load at the bottom surface of the female part, see Figure 5.10b, because this is the area that the floating body will push.

(a) Winch force direction

(b) Winch force application area

Figure 5.7: Winch force.

Environmental loads

An operation in the sea comes with environmental loads as well, like wind, current and waves. The former acts directly on the bend pipe while the latter two cause motions of the vessel that consequently affect the bend pipe, like all other vessel components.

1. **Wind**
   The wind pressures and the corresponding velocities are calculated in accordance with "DNVGL-ST-0378". The operational and extreme values were calculated for a level of 10 meters above the sea surface. For the calculation, the drag coefficient is 1.2, while the side projected area is 4 m^2.

$$F_{Wind} = A \cdot P \cdot C_d \tag{6}$$

   Where A is the projected area, P is the wind pressure and C_d is the drag coefficient. For the same area and drag coefficient, the occasional and extreme wind loads have the following values.

   - **Occasional wind**(24 m/s; 360 Pa): 1728 N
   - **Extreme wind**(44 m/s; 1200 Pa): 5760 N

   These forces shall be applied as distributed loads on the side of the structural parts.

2. **Vessel motions**
   Even though, the sea state is usually calm, it was decided to consider the vessel motions for the non-regular load cases. The "DNV GL rules for classification of ships RU SHIP Pt.3 Ch.4 Sec.3", was used for the calculations and thus, those are the extreme values for ship accelerations. As the project is not specified for one vessel, average values for a large vessel were used, as shown in table F.1.

   Table 5.1 shows the three combinations of the vessel accelerations, which are computed for the relative position of the bend pipe on the vessel. These accelerations are the result of all ship motions, heave, sway, surge, yaw, pitch, and roll. The gravity is not included.

Table 5.1: Load combinations for vessel accelerations

Load combination	α_x	α_y	α_z
Beam sea	0	$\alpha_{y_{Beam}}$ = 6.09 m/s^2	$\alpha_{z_{Beam}}$ = - 4.59 m/s^2
Oblique sea	$\alpha_{x_{Oblique}}$ = 0.80 m/s^2	$\alpha_{y_{Oblique}}$ = 3.65 m/s^2	$\alpha_{z_{Oblique}}$ = - 4.92 m/s^2
Head sea	$\alpha_{x_{Head}}$ = 1.33 m/s^2	0	$\alpha_{z_{Head}}$ = - 4.92 m/s^2

5.1.2. Load cases

All the aforementioned loads are applied on the system in different magnitudes and combinations with respect to the condition of the system. Conditions are divided based on the probability of occurrence of one load, and thus the possible combinations will arise. For the load cases the standard "DNVGL-ST-0378", is adopted due to its similarities to the environment under which the bow coupling operates. According to DNVGL-ST-0378, a total of three load cases should be considered, for which the required safety margins differ, in order to make the nominal safety based on the probability of the loading. Due to some obvious differences between the bow coupling and the offshore lifting appliances that the standard specifically refers to, some adjustments take place where needed.

- Case I: Regular operational conditions
- Case II: Occasional operational conditions
- Case III: Exceptional conditions

Case I

The first case includes all loads that act on the system under regular operational conditions. The only loads during normal operation are the gravitational load, pressure, flow force and lastly the F_{Hose}. Case I is the only one that applies to both models, although the first model does not include the F_{Hose}. As stated in Chapter 4, one of the variable requirements is to isolate the liner, without transferring any load from the outer environment. Thus, it is assumed that the liner receives only the loads exerted by the flow, like flow force and pressure, and the gravity. In practice, any deformation of the support pieces would result in loading on the bend pipe.

According to the standard, the vertical loads on the crane should be multiplied by a dynamic factor, which represents inertia forces and shock, from the vertical motion of the floating unit. However, such a factor is not taken into account as Royal IHC already took care of that with the magnitude of F_{Hose}, the only external load that is affected by the vessel motions. Also, the maximum dynamic factor is 1.5 based on the standard, while the F_{Hose} is more than twice as the applied load, considering the extreme scenario of 40 tons for the floating pipeline.

Apart from those, the standard suggests considering the angles of the vessel, from the horizontal plane. Specifically, it is recommended to use 5^o for the heel and 2^o for the trim. These angles can be considered only for the direction of the gravitational acceleration, since the other external load, F_{Hose}, is already applied at a large angle. Also, even though the contribution of those angles could be considered negligible, it was decided to apply them only in the same direction as the F_{Hose}. When the hose is at 45^o, the heeling angle is considered because of the higher horizontal component, while for 150^o the trim angle is considered as the aim of that load case is to check the stresses when the hose approaches the negative x axis.

Based on all these, there is a total of five load combinations (LC) for Case I, one for Model 1 and four for Model 2, see Table 5.2.

Table 5.2: Load Case I: Regular loading

(a) Model 1

Model	SF	Gravity	Pressure	F_{Flow}
1	1.5	-z	1	1

(b) Model 2

Model	SF	Gravity	Pressure	F_{Flow}	F_{Hose}
2	1.5	-z< 2º < +x	1	1	0º
2	1.5	-z< 5º < +y	1	1	45º
2	1.5	-z< 5º < +y	1	1	90º
2	1.5	-z< 2º < -x	1	1	150º

Case II

Case II essentially consists of the same loads as Case I, with the addition of environmental loads. The standard applies only the regular wind load at this stage, but due to the low contribution of that load, it was decided that regular vessel motions should be included. However, since the calculated vessel accelerations are for extreme conditions, 30% of them was considered appropriate to be applied. The vessel accelerations are used the same way as the gravitational force in Case I, at the same direction as the F_{Hose}, with the exception of 45^o and 150^o. However, the gravity remains parallel to the vertical axis as the vessel accelerations consider all vessel motions.

As shown in Table 5.3, there is a total of four load combinations, which are all applied to Model 2.

Table 5.3: Load Case II: Occasional loading

Model	SF	Gravity	Pressure	F_{Flow}	F_{Hose}	Wind	Vessel accelerations		
						Occasional	α_x	α_y	α_z
2	1.33	-z	1	1	0º	+y	0.3 $\alpha_{x_{Head}}$	0	- 0.3 $\alpha_{z_{Head}}$
2	1.33	-z	1	1	45º	+y	0	0.3 $\alpha_{y_{Beam}}$	- 0.3 $\alpha_{z_{Beam}}$
2	1.33	-z	1	1	90º	+y	0	0.3 $\alpha_{y_{Beam}}$	- 0.3 $\alpha_{z_{Beam}}$
2	1.33	-z	1	1	150º	+y	- 0.3 $\alpha_{x_{Head}}$	0	- 0.3 $\alpha_{z_{Head}}$

Case III

Any exceptional load companions shall be applied in Case III. For this study, there are two exceptional loads, one with extreme environmental conditions and one with an accidental load. It should be clarified that there are no operational loads, as the system is out of service.

1. **Case III - Extreme environmental loading:**
   For this case, only the extreme values of the environmental loads are applied, with the gravity. As these loads are very small compared to F_{Hose}, which is not involved, only one combination is considered. Both, wind and vessel accelerations are in the same direction, the side of the bend because there they sum up to the maximum load.

Table 5.4: Load Case IIIa: Extreme environmental loading

Model	SF	Gravity	Wind	Vessel accelerations		
			Extreme	α_x	α_y	α_z
2	1.1	-z	+y	0	$\alpha_{y_{Beam}}$	- $\alpha_{z_{Beam}}$

2. **Case III - Accidental loading:**
This case concerns the accidental load that can happen during connecting the floating pipeline to the female part. As explained before the operator might lift the hose more than needed and consequently push the female part upwards. Based on the standard, no environmental loads shall be applied for accidental loads and as a result, this load combination is composed of the F_{Winch} and the gravity.

Table 5.5: Load Case IIIb: Accidental loading

Model	SF	Gravity	F_{winch}
2	1.1	-z	1

Load combinations

Table 5.6 concentrates all load combinations as defined in the previous sections. It can be seen that there is a total of eleven scenarios that are to be analysed with FEM.

Table 5.6: Load combinations for all load cases

LC	Case	Model	SF	Gravity	Pressure	F_{Flow}	F_{Hose}	Wind		Vessel accelerations			F_{winch}
								Occasional	Extreme	α_x	α_y	α_z	
1	I	1	1.5	-z	1	1	0	0	0	0	0	0	0
2	I	2	1.5	$-z < 2^{\circ} < +x$	1	1	0°	0	0	0	0	0	0
3	I	2	1.5	$-z < 5^{\circ} < +y$	1	1	45°	0	0	0	0	0	0
4	I	2	1.5	$-z < 5^{\circ} < +y$	1	1	90°	0	0	0	0	0	0
5	I	2	1.5	$-z < 2^{\circ} < -x$	1	1	150°	0	0	0	0	0	0
6	II	2	1.33	-z	1	1	0°	+y	0	$0.3\ \alpha_{x_{Head}}$	0	$-0.3\ \alpha_{z_{Head}}$	0
7	II	2	1.33	-z	1	1	45°	+y	0	0	$0.3\ \alpha_{y_{Beam}}$	$-0.3\ \alpha_{z_{Beam}}$	0
8	II	2	1.33	-z	1	1	90°	+y	0	0	$0.3\ \alpha_{y_{Beam}}$	$-0.3\ \alpha_{z_{Beam}}$	0
9	II	2	1.33	-z	1	1	150°	+y	0	$-0.3\ \alpha_{x_{Head}}$	0	$-0.3\ \alpha_{z_{Head}}$	0
10	III	2	1.1	-z	0	0	0	0	+y	0	$\alpha_{y_{Beam}}$	$-\alpha_{z_{Beam}}$	0
11	III	2	1.1	-z	0	0	0	0	0	0	0	0	1

5.1.3. Simulation

Table F.2 shows all load cases with the actual values as they were calculated and subsequently applied to the system. Before moving to the final results for the load cases, some remarks about the simulation procedure and the materials, have to be mentioned.

Model 1

Initially, Model 1 was composed of only one part, the liner bend pipe, which was supported at both ends. The aim of that analysis was to find the minimum operational thickness, and therefore a number of simulations had to be completed in order to determine the thinnest liner that could operate under the applied loads. However, as the wall thickness was already very thin an additional factor had to be considered, known as the "membrane factor". Thus, the final factor, that applied to the yield strength of the pipe, was the multiplication of the safety factor (SF) with the membrane factor, which was taken as 1.5, resulting in a total of 2.25.

In reference to the material used for the liner, there is a series of five different cast materials suitable for abrasive and erosive environments. The current design uses the first one because of the high loads exerted on the bend pipe. Due to the fact that the new design manages to minimise the loads on the liner, another material with better wear resistance could be selected. However, the inevitable loads on a bend liner do not allow for the selection of materials with very low impact value. Consequently, the second material of that series was considered appropriate. *Wearmet S2* is a slightly more expensive material compared to *Wearmet S1*, providing though almost twice material hardness, one of the most important wear parameters. The yield strength is 850 MPa, meaning that the allowable stress is 377.78 MPa.

Model 2

For the second model, four different parts were used, the previous straight pipe, the female part, and the two half pieces to support the bend pipe. The geometry of the flanges was according to company guidelines, based on the size of the pipe. However, as the model is cast, during the simulations, some modifications took place in the areas with high stresses on the support pieces. Thicker flanges and strengthened connections between the flanges and the pipe's outer wall reduced the stresses to the allowable range. For the previous pipe and the female part, the original geometry was kept, despite the fact that the previous straight pipe showed high stresses and deformation. That was the reason, the previous pipe was included in Model 2, instead of applying constraints at the bolt holes of the inlet flange, for the deformation. Less stiff and strong previous pipe results in a more extreme but at the same time realistic scenario, to analyse the support pieces. Moving on, all parts were connected via *M*48 bolts, which were designed as beam elements and a pretension of 732.3 kN was applied to all of them. Moreover, the liner mass of each pipe was added as well as the mass of the mixture in the pipes, for the operational load cases. With regard to the orientation of the pipe, it was decided to simulate all load combinations with the flipped bend pipe. In that way, there are two inspection holes very close to the inlet flange, where the maximum stresses were expected.

In contrast to the liner, the structural parts do not require special materials. For the straight pipe and the female part, the structural steel *S355J2* was used, while for the two cast pieces, the *G28Mn6 (+QT1)* was used, with 450 MPa yield strength.

Results

Figures 5.8 and 5.9 show the results from the FEM analysis. The maximum value of the label corresponds to the maximum allowable stress according to material yield strength and the safety factor of the load case. Specifically, figure 5.8 shows the liner with the lowest allowable thickness, for which the highest stresses appeared close to the two support areas on the inner wall.

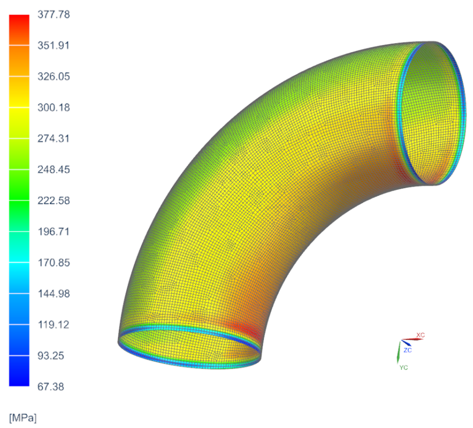


Figure 5.8: Simulation result: Model 1 (LC1)

On the other hand, figure 5.9 depicts LC5 which caused the highest stresses on the second model and therefore played a crucial role in the final geometry. The results of all load cases for Model 2 can be seen in Appendix F.3. In that case, the maximum stress concentrates in the short flange of the inner wall and bolt areas. The former is mainly because of the floating pipeline force, while the latter is a result of the high pressure. As one would expect, initially, the flange and the bolts at the top side of the inlet, showed significantly high stresses, but some careful modifications managed to reduce them. All load cases had similar stress profiles on Model 2, except the scenarios of Case III, whose maximum load was just the pretension of the bolts. Finally, it should be mentioned that the reaction forces of Model 2 are below the maximum allowable forces for the tower, as they have been standardised by the company.

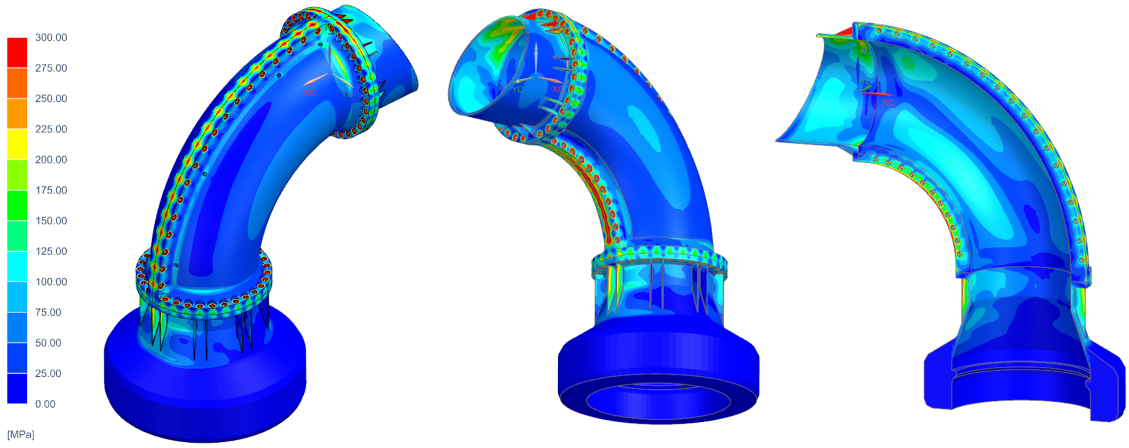
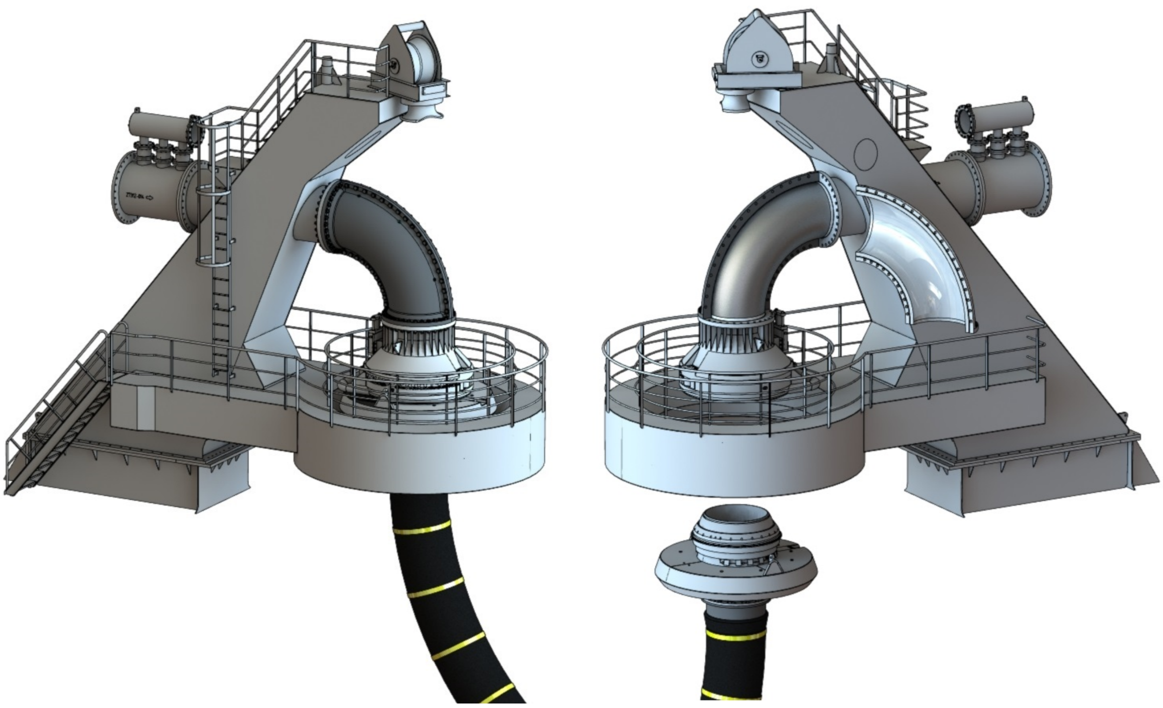


Figure 5.9: Simulation result: Model 2 (LC5)

5.2. Final design parameters

The new design is composed of three cast parts, the two support pieces and the liner. All of them will have a 25 mm wall thickness, which corresponds to a total of 3.62 t for the support and 1.75 t for the liner. Furthermore, the support pieces are made of *G28Mn6 (+QT1)*, while for the liner the *Wearmet S2* was selected out of a series of wear-resistant cast materials. Figures 5.10 depict the final design with the floating pipeline. Moreover, figure 5.11 shows the cross-section of the model, where the flow path is indicated with red color and the liners in the corresponding pipes with yellow color. The basic dimensions of both models can be found in the final drawings in Appendix G.



(a) Starboard side

(b) Port side

Figure 5.10: Final design

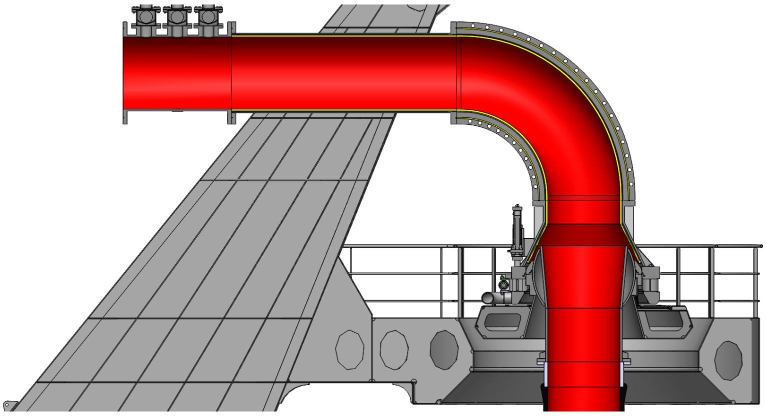


Figure 5.11: Final design: Flow path

The special point of the concept is that the liner has to be flipped after a certain point, as explained in the previous chapter. Based on the analysis of the liner, the minimum allowable thickness is 4 mm. Before flipping, the maximum thickness reduction is expected at the outlet's outer wall. However, in Section 3.2.3, it was assumed that the wear at the outer wall increased almost exponentially from the inlet towards the outlet. This means that after flipping, the inlet and outlet areas of the outer wall are expected to show the maximum loss of material. The only way to control which side will reach the wall thickness limit first is the flipping moment. An early flipping of the pipe ensures that the new outlet will have the minimum thickness of the pipe, while a late flipping will transfer the minimum thickness to the new inlet. Moreover, the earlier the flipping is, the lower the total wear thickness will be. Regarding this dilemma, it was decided that the total wear thickness is more important than knowing on which side the minimum thickness will be. For that decision, the fact that the two defined zones already cover relatively small areas of the bend pipe was taken into account, as well. At this point, another assumption from Section 3.2.3 comes to play a crucial role, and this is the wear rate difference between the inlet and the outlet of the out wall. The wear rate and consequently the wall thickness reduction of the maximum wear zone is five times higher than that of the low wear zone. It must also be noted that the total wall thickness reduction is defined as the summation of material loss before flipping and before replacing the pipe at the "maximum wear zone". That being the case, the thickness of the outer wall at the outlet area determines the total reduction, and consequently, the minimum thickness of 4 mm in that area shall constitute the reference point. Considering this and the wear rate relation between the inlet and outlet, the flipping thickness should ensure that the final thickness at the inlet outer wall does not go below 4 mm. Eventually, the bend pipe has to be flipped when the wall thickness at the outlet reaches 8 mm. The final expected thickness is 4 mm and 4.48 mm for the outlet and inlet of the outer wall, respectively. This difference is negligible and the two thicknesses can be considered as equal, 4mm, expecting the minimum allowable thickness at the same time. Finally, the total wall thickness reduction of the outlet outer wall is 34.6 mm.

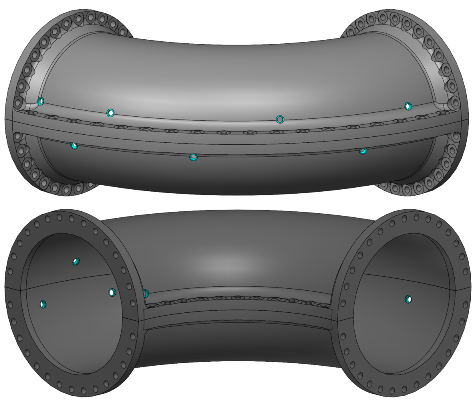


Figure 5.12: Inspection plugs: Final configuration on the model.

In relation to manufacturing, the cast model does not involve the inspection holes, the bolt holes and the area around them. However, during the analysis, it was necessary to add material at the connections of the flanges with the support pipe's wall. As a result, the fabrication of the two cast parts might have some challenging tasks, especially for the bolt holes close to the connection of two perpendicular flanges. Additionally, due to that extra material, the inspection holes had to be shifted away from the flanges, compared to the initial locations as defined in Section 4.6.1. Specifically, the inspection plugs are placed at $\phi = \pm 15^o$ to each side of the pieces' connection flanges and the two plugs placed close to the circled-shaped flanges are positioned at $\theta = 4^o$ and $\theta = 8^o$ of the outer and inner wall, respectively. Finally, the eight inspection holes are divided equally into the two support pieces and their final configuration is presented in figure 5.12.

5.3. Comparison with the current design

The aim of the project is to develop a cost-effective design that extends the lifespan of the bend pipe and reduces the amount of material that has to be replaced every time. Thus, the proposed design is compared with the current one in those three aspects to evaluate the effectiveness of the new design. The first paragraph focuses on the lifetime, whereas the second and third ones discuss the amount of material and the expected cost, respectively.

To begin with, the lifetime can be estimated according to the total thickness that the two designs can lose due to wear. The current design has an initial thickness of 35 mm and considering the minimum operational thickness of 24 mm, the maximum thickness reduction is 11 mm. On the other side, the new design is expected to reach 34.6 mm as a total thickness reduction. That case corresponds to about a three times longer operational lifetime. Furthermore, the fact that the new design can be made of a more wear-resistant material prolongs, further, its lifetime.

With regards to the replaceable amount of material, the mass of the current design barely exceeds 3 t, while the new design weighs about 5.37 t, 3.62 t for the support pieces and 1.75 t for the liner. Although the new design initially requires about 2.4 t more, for its first replacement only 1.75 t will be needed, because only the liner has to be replaced. This corresponds to 58 % of the amount of material that the current design has to replace every time.

As there is no detailed analysis of the cost, the comparison was made based on estimations, for the support pieces and the liner. Starting with the support pieces, although they need slightly more amount of material, the price for that material is significantly lower than the current one, because there is no need for high wear resistance. Additionally, the support requires more machining because of the flanges and the number of bolts, and lastly, the fact that they are two pieces increases the casting cost. Considering those four points, it is estimated that the support pieces will cost 1.5 times as much as the current design. Regarding the liner, the new material is slightly more expensive than the current one but its production requires 1.25 t less amount of material. Also, it is a simple cast design and there is no need for afterwards machining. It is predicted that the liner will cost half of the current design. Based on those two estimations, it can be assumed that the new concept costs twice as much as the current one. At the same time, based on the allowable thickness reduction, the service lifespan of the new design is more than three times as the current one. This means that the current model will already proceed to two replacements before even the new model reaches its first one. With respect to financial profit, the new design depreciates the extra initial cost before the first replacement. The long-term economic benefit of the new design is obvious, based on the results of this study, even though a number of assumptions had to be taken. In other words, if the company produces three of the current design bend pipes, in practice, the first two cover the initial cost of the new total system, support pieces and liner, while the third one pays the expenses for the next two liners of the new design.

To sum up, the new design has the potential to triple the operational period, requiring less than 60% of replaced material and saving about 50% for every replacement.

Lastly, the fraction of the total thickness reduction over the initial wall thickness can be used to compare the efficiency that the two designs have in managing their original material. The current design has the initial thickness of 35 mm and as the minimum thickness is 24 mm, the maximum thickness reduction is 11 mm, resulting in 31.43% of the initial thickness. On the other hand, the new design, having the summation of before and after flipping reduction, reaches 34.6 mm. Considering the initial thickness of 25 mm, it means that the thickness reduction is 138.4% of the initial thickness. The new design is able to lose more total thickness than its actual initial thickness. This is not an unusual situation for the dredging industry because as already mentioned, it is common to rotate the straight pipes and have a more distributed thickness reduction.

5.4. Conclusion

The completion of this chapter comes to prove the technical feasibility of the new design, based on all those considerations from the previous chapters. At first, the loads exerted on the bow coupling were explained and realistic magnitudes were applied to them. The loads were then combined according to standards and guidelines in order to create the scenarios for the FEA, where the final geometry and material were determined. Precisely, the support pieces and the liner have 25 mm wall thickness, while the minimum allowable thickness for the liner is 4 mm. Furthermore, the liner has a harder material than the current design, but the support pieces' material is less wear-resistant because it is not affected by slurry wear. Based on its final parameters, the proposed design was compared with the current one showing that the lifetime extends at least three times, while the replaced amount of material and the replacement cost reduced to about 60% and 50%, respectively.

6 Conclusion

The aim of this research study was to design a new bow-coupling bend pipe, which could prolong its lifetime while decreasing the replacement material and cost to the greatest extent. To begin with, an elaborate analysis of the slurry transport and the wear mechanisms for dredging pipelines was made, defining also the parameters that affect the slurry wear rates in bend pipes. Furthermore, after conducting a literature review on bend pipe wear profiles, the expected wear pattern for the bow-coupling bend pipe was determined, using also the wear parameters and assumptions. The obtained information regarding the slurry wear in the pipe and the double contribution of the bend pipe to the system constituted the main pillars on which various concepts were developed, with respect to the priorities of the project. After concluding on the optimal concept, the main focus was placed on the technical feasibility of the new design, considering realistic load cases according to standards and guidelines. Lastly, the evaluation of the proposed design was performed on the three aspects of the thesis objective.

The research sub-questions and finally the main question of this thesis report are individually addressed in the next paragraphs, while the last two sections provide recommendations for the company and future research work.

6.1. Research question

1. **What are the wear mechanisms for slurry transport in dredging pipes, and which of their parameters can be used to reduce slurry wear rates in the bow-coupling bend pipe?**

   The slurry wear constitutes the main reason for the frequent replacement of the current design. The abrasive nature of the slurry transport in combination with the bend pipe shape results in rapid wall thickness reduction, necessitating the replacement of the pipe. There are three wear mechanisms that occur in during slurry transport in dredging pipelines, erosion, abrasion and corrosion. Erosive or impact wear, is when particles impact against a surface at an angle, while abrasive or sliding wear, is the loss of material by the passage of hard particles over a surface. Corrosive wear is the process of chemical or electrochemical metal degradation, and it can accelerate the total wear during the so-called "synergistic action", which is the cumulative effect of all mechanisms together. However, for slurry wear and especially in bend pipes, erosion was found to be the most dominant mechanism.

   At the same time, the parameters that influence those mechanisms can be used to reduce the wear rates in the bend, paying more attention to the erosion factors because it is the most important mechanism. The parameters can be divided into four categories, the four components involved in the erosion process, the solid particles, the slurry mixture, the target surface and lastly the particle-target condition. As explained in Chapter 2, only the target surface parameters can be changed and specifically the type of material. Brittle materials show much better wear performance compared to ductile ones, and thus the use of a more wear resistant-material can reduce the wear rate of the pipe. One condition that comes with the utilization of materials with relatively high brittleness is that any dynamic or impact loading on them should be minimized.

2. **What is the expected wear pattern of the bow-coupling bend pipe?**

   The determination of the wear pattern constituted one of the most challenging tasks of the project because there has been no specialized wear analysis for this bend pipe. Also, when reviewing the available literature, one may notice that the majority of the papers are dealing with much smaller bend pipes and the carrier fluid is gas. Nevertheless, studies that investigated slurry transport with liquid as a carrier fluid, showed significant common elements about the wear profile in the bend pipe, despite the wide variety of flow conditions and pipe sizes. The general outcome of the literature review was that the wear in the pipe is mainly on the outer wall, on which the wear rate tends to keep increasing along the direction of the flow, reaching the maximum wear rate at the outlet area. Considering this conclusion and the slurry wear parameters from Chapter 2, the literature outcome was adapted to the bow-coupling bend pipe operational conditions. Peculiarities of the bow coupling and the lack of valid information in some cases made necessary the use of assumptions to complete the expected wear pattern.

   The bend pipe was divided into smaller critical zones and a range of five wear levels was considered to describe them. Specifically, the bend's minimum and maximum wear areas are expected on the outer wall, in the first and the last 20^o, respectively. The inner wall is not expected to have significant damage, with the exception of the two bend ends, which are affected by the flow situation in the corresponding connected pipes. Since no study has been conducted on bow coupling bend pipes, the level of uncertainty was relatively high and therefore the decisions made tend to be more conservative.

3. **In which ways can a new design reduce the replacement material and prolong the pipe's lifespan?**

   One of the special characteristics of the bow coupling is that the bend pipe currently serves two functions, flow and support, which lead to the increased wall thickness of the bend pipe. As a solution to that situation, it was decided to split the system into two models, which should be dedicated to only one function. Thus, the components for the flow function should not be involved in the support of the system, and the support parts should not be considered for the slurry transport and therefore not be affected by the slurry wear. In that way, only the inevitable loads of internal pressure and flow forces are exerted on the flow function members because the support components are responsible for the additional loads.

   This case benefits the project in two ways. Firstly, the high minimum operational thickness, which was necessary for support purposes, can eventually be reduced and consequently result in a notable cut of the material volume that should be replaced. The second advantage of specifying components for each function has to do with the type of material that can be used to improve the performance of the corresponding parts under their operational circumstances. As Chapter 2 concluded, more wear-resistant material is the only parameter that can be adopted to decrease the wear rates in the bend pipe. The isolation of the bend pipe from the high external loads allows the utilization of a more brittle material whose hardness contributes to lower wear rates and therefore to a longer lifetime.

   Apart from that, there is another feature of the proposed design that aims at the lifespan. Considering the expected wear pattern as defined in Chapter 3, the bend pipe has to be flipped at a certain point and then replaced. After flipping the bend pipe, the outlet will be connected to the upstream pipe and therefore the last 20^o of the outer wall, where the highest wear rates and the minimum wall thickness are, will become the first 20^o of the outer wall, where the wear rates are expected to be the lowest. Vice versa, the area with the least thickness reduction will become the area with the maximum wear rates of the bend pipe.

   To sum up, separating the components, according to their functions, results in a reduction of the replacement material, while a more wear-resistant material and the flipping option can extend the lifetime of the bend pipe.

4. **What is the technical feasibility of the new design?**

   In order to evaluate the technical feasibility of a new structural design, its behaviour should be analysed under real-life loading scenarios. The bend pipe–liner and the support pieces were considered separately since the loads exerted on them are different. Based on calculations and company guidelines the magnitude and the applied locations of the loads were specified. Finite element analysis took place, applying eleven load combinations, which were developed in accordance with standards. The yield strength and the safety factor of each case determined the maximum allowable stress for each component, a value which is not reached using the final geometry and materials.

- Main research question

  To what extent can an alternative bow-coupling design reduce the replaced amount of material and extend its lifetime, in a cost-effective manner?

  The main objective of this project was to explore the potential of improving the current bow-coupling design in three aspects, the pipe lifetime, the replaced amount of material, and the cost. Thus, a comparison with the current design takes place in relation to each of those three areas in order to evaluate the extent that the new design accomplishes the study's goal.

  To begin with, the first indicator to estimate the lifetime is the total thickness that the two designs are expected to lose due to wear. The liner of the new design has a 25 mm original wall thickness, and it should be flipped at 8 mm, before being replaced at 4 mm, which is the minimum allowable thickness for this part. By adding 17 mm before and 17.6 mm after flipping, a total thickness reduction of 34.6 mm can be achieved. Considering that the current design has a maximum thickness reduction of 11 mm, it means that the new design can achieve about a three-times longer lifetime. Moreover, the second indicator has to do with the material, which can reduce the wear rate and consequently the corresponding material loss. The fact that the new design can be made of a more wear-resistant material, with higher hardness, extends further its lifetime.

  Regarding the amount of the replaced material, the mass of the two designs is used for the comparison. The mass of the current design is about 3 t, while a new design weighs about 5.4 t, 3.62 t for the support pieces and 1.75 t for the liner. Although the new design initially requires 2.4 t more, for its first replacement only 1.75 t will be needed, because only the liner has to be replaced. This corresponds to 58 % of the amount of material that the current design has to replace every time.

  In reference to the cost comparison, some estimations had to be made as there is no detailed financial analysis for the new design. It is predicted that the liner and the support pieces will cost 0.5 and 1.5 times as much as the current design, respectively. Based on those two estimations, it can be assumed that the new design will initially cost twice as much as the current one. Even though the initial cost is higher, the new design will depreciate the cost for the support pieces before the first replacement of the liner, because by that moment the current design would have already been replaced twice.

  Thus, the proposed design manages successfully to achieve significant improvement in the three aspects of the thesis objective. It has the potential to at least triple the current operational period of the bend pipe, requiring less than 60 % initial amount of material and reducing the replacement cost by 50 %. To conclude, the fact that the total thickness reduction is 140 % of the original thickness, while for the current design is just 31%, shows careful management of the material, which leads to a more sustainable solution.

6.2. Recommendations

Throughout the study, a number of limitations were presented in different areas which influenced the final model. The limitations of this study are discussed in this section, as well as potential avenues for future research. The first part suggests practices that can be adopted by the company in order to improve the general situation of bow coupling, while the second part refers to recommendations for the scientific world.

6.2.1. Royal IHC

Initially, the unusual connection from the female to the male part, most probably affects the flow, as suddenly the diameter of the pipe increases, and the flow that does not go directly into the male part eventually hit the surrounding walls. This situation increases the wear of that area, the energy of the flow is reduced and lastly there must be intense turbulence which affects even the bend pipe, which is before the connection. Thus, it is recommended to investigate the possibility of having the male part connected to the bend pipe, and the female part at the end of the floating pipeline.

The next point is about the reason for the most important limitation, the connection procedure. The space around the female part and the bend pipe, which should be kept free for the hoisting cables constitutes the main obstacle to a good support system. The support of the female part from all directions could lead to a strong and stiff structure, that minimizes any loads on the bend pipe. Another hoisting procedure can be found or even a more radical modification of the way the floating pipeline connects to the vessel.

Moreover, the connection of the floating pipeline could be monitored by a camera placed at the bow coupling area. In that way, the operator of the winch could watch the procedure from a safer place than being on the platform. Due to the dynamic loading on the hoisting cables and the impact loads of the floating body on the platform and the female part, the cables might break. In general, as explained in the report, it is recommended that there are no people involved in any process or close to areas with moving parts.

The last recommendation has to do with the inspection of the bend pipe and in general any part that shows high wear rates. It would be good to make regular inspections and keep the data for future designs, and to validate numerical and experimental models. The measurements can be connected to the corresponding operational conditions, such as mixture velocity and particles' properties. This practice will significantly contribute to a better understanding of the wear process, especially nowadays when there is no available information about the wear in dredging bend pipes. An additional step would be to replace the inspection plugs with permanent measurement devices. However, despite the high cost, this solution requires a special connection of the device to the liner wall, which can keep it in place under vibrations.

6.2.2. Future research

Generally, the areas that require further research revolve around the wear and the loading on the bend pipe.

Bend pipe wear

Starting with the wear of the bend pipe, it is clear that there is a limited number of studies that used slurry transport in large-scale bend pipes or even straight pipes. The two ways to increase knowledge about that field are numerical, with computer software, and experimentally, in the lab or the field. There are various available software that can simulate the flow in the pipe and the subsequent wear of the wall. The difference between those analyses depends on the considered interactions and their interaction models, among others. Precisely, an engineer should consider all interactions simultaneously, fluid-particle and particle-particle influence the flow behaviour and the particles' path, while the particle-wall determines the wear in the pipe. Inter-particle collisions had been neglected in numerous calculations conducted in the past. Collisions of particles will significantly reduce wear in a pipe due to the loss of energy and the buffer effect. Consequently, ignoring inter-particle collisions will inevitably lead to a significant overestimation of wear rates and the worn areas, especially at relatively high concentrations.

Also, the considered wear mechanisms can play an important role in the final result. Although, erosion appears to have a dominant effect on the wear profile of the bend pipe, abrasion and even corrosion shall be included. Corrosive wear is the most challenging mechanism to be applied, as it is not a mechanical process.

Another suggestion about the numerical analyses is to include more than one size of particles in the same simulation instead of taking the average size. This will bring the model even closer to a real situation of slurry transport, a multi-phase mixture.

The last point that can be taken into account and affect the wear of the bend pipe is the geometry. As explained in the report, the wear process removes material from the pipe surface, and therefore the inner geometry of the pipe is slightly changed. The recommendation is to apply the thickness reduction on subsequent simulations to evaluate after what point the reduced thickness affects the flow and the wear. This can also be used at the connection between two pipes, in order to determine the maximum allowable thickness, at which the local wear rates increase dramatically. Finally, the scenario of having two different materials in the bend pipe should be investigated. As explained in the report, the option of having harder material at locations with higher wear rates was rejected to avoid unexpected wear patterns in the bend. The option of having two different wall materials in some areas of the pipe should be investigated.

These points will significantly contribute to a more realistic analysis, predicting more accurately the wear of the new model, or validating the results from experiments or inspection measurements. However, most studies tend to simplify their models in order to reduce computational effort and time, another important issue, that has to be improved.

In reference to experimental studies about slurry transport and the resulting slurry wear, it is indeed difficult and expensive to create a lab for large-scale pipes. Thus an alternative solution would be to take advantage of the dredging vessels, which already use this equipment. Some parts along the pipes could be used for detailed measurements, such as wall thickness, velocity, concentration, particle size distribution, and flow regime.

Another area that requires more research, especially for large-scale applications, is the wear reduction in the pipelines. One way to achieve that is by protecting the pipe from the abrasive mixture, like coatings and other hard materials, like chocky bars. Even though both of these solutions are notably expensive, chocky bars have been found application in the dredging industry. On the other hand, the promising coatings are far behind this step, because of their inability to provide adequate protection from the slurry transport, and the deposition methods. Further research should take place on both protection technologies, or even a new, innovative one.

Also, wear resistance materials are in general more brittle, depriving of their application in areas where the pipe is under high dynamic loading, like bends. Probably not only the chemical composition but also the production process can lead to components that can combine both aspects, by aiming also on recycled and low-cost solutions.

Moreover, the shape of the bend pipe proved to be a solution to reduce the wear in bend pipes. Most of these designs, however, require significant modifications to the pipelines, and they have only been applied to much smaller bend pipes. If research shows that any of them has good performance in large-scale pipeline systems, that would change the market of the dredging industry crucially.

The last suggestion about the wear in the bend pipe has to do with the analysis that should take place for every new design. First and foremost, several flow conditions shall be tested in the actual geometry of the new pipe. Specifically, for the bow coupling bend pipe, the female part and the connection with the male part should be involved in the analysis. It is recommended to consider the male part at 15^o angle from the vertical axis and in different directions on the horizontal plane, as applied in the FEM analysis of this project. Also, the previous pipes should have the same geometry as the vessel on which the system is planned to be installed. In that way, the results approach a more realistic situation and therefore the expected wear in the pipe has lower uncertainty. From a simulation like that, the reaction forces on the pipe walls can also be used as input for the FEM analysis of the involved support pieces.

Bend pipe loading

Moving on, the loads exerted on the system are another part that needs more investigation. Precisely, the complex behaviour of the floating pipeline and consequently the subsequent forces on the female part have not been studied yet. For that reason, companies use rules of thumb to determine the load that should be applied on the support point of the floating pipeline, the female part. However, a scientific analysis should be done to investigate the loads generated from the hose. The analysis should consider the pipelines' geometry, materials' properties, wave and current forces, total length, fixed support at the shore side and most importantly the ball connection with the vessel, which is affected by waves and current, as well.

Additionally, the accelerations and angles caused by the sea state must be taken into account for the analysis of any part of the vessel. In general, the vessel motions, the flow passing through the pipelines, and especially the floating pipeline create different dynamic loads acting on the bend pipe. Thus, it is recommended that not only strength but also fatigue analysis shall be considered for the system.

The last point about the loading is about the liner, which usually does not take part in the analysis. As it was explained in the report, most probably the pressure passes through the pipes' connections and equalizes the pressure on both sides of the liner wall. Even though this project analyzed the liner, it was under the assumption that there are no loads transferred from the support pieces to the liner. Thus, another model could be developed which includes both support pieces and liner, connected at the two ends of the bend pipe. In that case, the pressure and the flow force shall be applied only on the liner.

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Appendix

A

Research paper

Structural design of a bow-coupling bend pipe

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Abstract—This research work provides an understanding of the existing challenges for bow coupling bend pipes and explores where improvements can be made. Precisely, the effect of slurry wear in dredging pipes is explained, addressing the wear mechanisms and discussing the parameters that influence the wear rates in bend pipes. Furthermore, a literature review takes place on wear patterns in various bend pipe geometries and operational conditions. The wear parameters and some assumptions help to determine the expected wear pattern in the bow coupling bend pipe, considering five levels of wear rates. Additionally, different concepts are generated, aiming at the reduction of the pipe replaced material and the extension of its lifetime, keeping the overall cost into consideration. Several weighted factors are used to determine the final design, and several load cases are applied to the system to check its technical feasibility. Finally, this work provides the wear pattern that can be expected in large-scale bow coupling bend pipes and at the same time, the company can adopt the new design to save a significant amount of material and consequently money.

Index Terms—Slurry wear, Bend pipe wear pattern, TSHD, Bow coupling design, Bow coupling loading.

I INTRODUCTION

In recent decades, there have been many technological developments in the dredging industry. Specifically, larger "Trailing suction hopper dredger" vessels have been built in order to accelerate the whole process, by reducing the number of cycles. Thus, the equipment related to the process has become larger and heavier. One of the parts that require reconsideration is the bow coupling bend pipe, which has a double role. The primary role is as means of transport of the dredged material, while the second function is to provide structural support to other parts. At the same time, the very abrasive nature of the dredged mixture constitutes another challenging part for the dredging pipelines, with considerably more wear effect in curved pipes. It is therefore important to investigate whether the bow coupling installation can be constructed differently.

The main objective of this work is to explore the potential of developing an alternative bow-coupling design that minimises the replaced amount of material and extends its lifetime, in a cost-effective manner.

The research paper's outline is as follows. First, the required background of the dredging industry is provided and the problem of the current design is expressed. Then, the focus is on slurry wear, providing an introduction to slurry transport, the mechanisms and the parameters that affect them. A literature review on the wear profiles in bend pipes is the next step, which is used to determine the wear pattern of the bow coupling bend pipe. Moving on, the procedure of developing concepts is explained while the final concept is decided after comparing all of them according to weighted factors. The selected design is then analysed under various load combinations for its technical feasibility, whereas a comparison with the current design takes place to evaluate the proposed solution. Finally, in conclusion, some remarks about this work are made.

II PROBLEM FORMULATION

Starting with the current design, Fig. 1 shows the main parts of the bow coupling. At the bottom of the figure, the male connector part can be seen with the floating body. This is the end of the floating pipeline which connects the vessel to the onshore discharging area. To start discharging, the male connector part is lifted and led into the female connector part, which locks it there. As can be seen, the female part and the floating pipeline are supported by the bend pipe, because it is the only part that is connected to the vessel. Thus, there are additional loads exerted on this specific bend pipe compared to the regular ones. Firstly, all curved dredging pipes have to withstand the high inner pressure and the flow forces, because of the redirection of the flow. However, this bend pipe has to support the weight of the female part as well as the high loads from the floating pipeline. Also, environmental loads, are applied to the system, although their influence is much lower compared to the floating pipeline load. All those loads are more closely reviewed in the analysis of the system. At the same time, the slurry transport, in combination with the small bend curvature ratio, leads to extensive wear in the pipe, which has to be relatively frequently replaced. Thus, the bow coupling bend pipe has two functions, the flow function like all other bend pipes, and the support function, because of the additional loads.

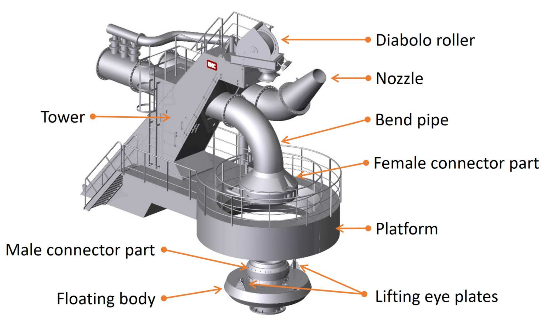

Fig. 1: Bow coupling main parts.

Currently, the inner diameter of the bend pipe is 1000 mm and the bend radius is 1700 mm. Moreover, for its double contribution to the system the bend pipe has a wall thickness of 35 mm, resulting in a total mass of 3001 kg. The minimum thickness before replacement is 24 mm, meaning that the maximum thickness reduction due to wear is 11 mm. The last 24 mm are necessary to avoid failure because of the loads, under which the bend pipe should operate. As a result, more than half of the initial amount of material has to be replaced with a new cast pipe of about 3 t. A relatively frequent replacement of such parts has economic consequences due to the large cast model and the required subsequent fabrication.

III SLURRY WEAR

A Slurry transport

One of the most important elements of dredging is the transport of slurry, a multiphase mixture that consists of one or more different types of particles and a fluid, in this case sea water. The size of the solid particles can vary from the scale of micrometres to centimetres, and it is one of the most significant parameters that affect the slurry's rheology. Additional parameters that influence the physical behaviour of the slurry are the density and viscosity in reference to the carrier liquid, and with regards to the particles, the density, size, and shape [1, 2, 3].

According to Berg [4], the flow of solids-liquid mixtures through a pipeline may be broken down into four different flow regimes. "Homogeneous" flow regime is when all particles are fully suspended and there is a uniform distribution of them across the flow section. The second category is the "heterogeneous" flow regime, where although all particles are in suspension, there is a significant concentration difference between the top and bottom of the pipe. Next, with the "sliding bed" regime, a portion of the solids particles are carried as a suspended load and the remainder is moved as a bed load. The last regime is the "fixed bed", where even though the slurry flow continues, some particles settle down and remain as a stationary deposit on the bottom. The transitional velocity, from the sliding bed to the heterogeneous regime, is the critical velocity because at this speed the hydraulic gradient appears to be at a minimum, which corresponds to minimum resistance offered by the pipeline [5].

MTI Holland [4] has developed, by using regression analysis, a simple equation that is based on the results of experiments in the laboratory and measurements during dredging activities. The equation is valid for sand with a density of $2650\,\mathrm{kg/m^3}$ and fluid with sea water density, $1025\,\mathrm{kg/m^3}$.

$$v_c = 1.7 \cdot \left(5 - \frac{1}{\sqrt{d_m}}\right) \cdot \sqrt{D} \cdot \left(\frac{c_v}{c_v - 1}\right)^{\frac{1}{6}} \tag{1}$$

Where v_c in the critical velocity in m/s, d_m is the mean grain diameter in mm, D is the inner diameter of the pipe in m, and c_v is the volumetric concentration on solids.

For the current project, medium sand is considered and therefore the mean grain diameter is taken as 0.4 mm. The inner diameter is 1 m, as it is specified in the study requirements. The volumetric concentration is selected to have the maximum possible value, in order to minimize the risk of shifting to a lower flow regime, [4]. TSHD vessels can reach a concentration of 0.48 during discharging, which corresponds to a mixture density of $1800\,\mathrm{kg/m^3}$, with carrier liquid and particles' density, at $1025\,\mathrm{kg/m^3}$ and $2650\,\mathrm{kg/m^3}$, respectively.
The critical velocity according to Equation 1, under those conditions, is $5.63\,\mathrm{m/s}$. As the operational line speed is about $6\,\mathrm{m/s}$, that value comes to prove that the system falls into the desired heterogeneous flow regime, by keeping also the hydraulic gradient at the lowest level.

B Wear mechanisms in dredging pipelines

Wear is the most significant factor that reduces the bend pipe's lifespan, and it can be defined as the progressive volume loss from a surface, [6]. This section presents the three wear mechanisms that are considered during slurry transport in dredging pipes, erosion, abrasion and finally corrosion.

According to Bitter [7], erosion is defined as "material damage caused by the attack of particles entrained in a fluid system impacting the surface at high speed" while Hutchings and Winter [8] define it as "an abrasive wear process in which the repeated impact of small particles entrained in moving fluid against a surface results in the removal of material from that surface", [9]. In the 1960s, Finnie [10] and Bitter [7, 11] systematically studied slurry erosion for the first time, proposing also erosion models like Neilson and Gilchrist [12], and Hutchings [13], [14]. Since then, various evaluation techniques and test methods have been used to assess erosion, [15].

An investigation of previous erosion models by Meng and Ludema [16] revealed that 28 erosion models were linked to solid particle impingement, as well as 33 key parameters affecting erosion rate. However, robust models have not yet been developed for slurry erosion, which is a complex and understudied area. Certainly, this is the case for pipeline erosion, which is compounded by the fact that most literature on the wear of pipelines focuses on pneumatic conveying systems, [17].

Stachiowak and Batchelor [18] showed that the erosion mechanisms mainly depend on the impact velocity and angle as well as the ductility of the target surface. Brittle materials have much higher wear resistance, and the maximum erosion rate occurs near normal impact. Ductile materials show maximum erosion at intermediate impact angles, with the exact number varying between 20^o and 50^o. For example, Al-Bukhaiti et al. [19] observed that the maximum erosion for steel AISI 1017 occurred between 40^o and 50^o. Still, Berg [4] and Patel et al. [6], stated that 20^o - 30^o is the range for the maximum erosion while Barkoula and Karger-Kocsis [20], Oka et al. [21] and Hufnagel et al. [22], concluded at 30^o, 30^o and 20^o, respectively.

Abrasion, or sliding wear, can be defined as the loss of material during relative motion between two solid surfaces in contact under load, as stated by Hutchings and Shipway [23]. According to Stachiowak and Batchelor [18], the four abrasion mechanisms are cutting, fracture, grain pull-out and fatigue by repeated ploughing. The particle sharpness and the relative hardness between the particle and the surface determine which mechanism will occur.

Corrosive wear is the process of chemical or electrochemical metal degradation, [6]. However, the importance of corrosion in the dredging industry arises when it occurs with other wear mechanisms, erosion and abrasion in that case. In literature, the cumulative effect of both wear, chemical and mechanical, is perceived as erosion-corrosion [24, 25, 26] where erosion includes both impact and sliding wear in pipes. It is proved that the total weight loss of materials during the erosion-corrosion process is generally higher than the sum of pure electrochemical corrosion and pure mechanical erosion due to the synergistic effect of erosion and corrosion [27, 28, 29, 30, 31]. Nonetheless, as [32, 33, 27] stated, the contribution of pure corrosion at the bend pipe's total wear is much smaller than erosion, a situation that indicates that pure corrosion is not a dominant factor.

C Wear parameters for bend pipes

In this section, the parameters that affect the wear, in the pipe are to be explained. These parameters can control the level of wear in a pipe and consequently, proper management of them can lead to reduced wear. Nonetheless, by wear, erosion is mainly meant, and most of the parameters are related to that mechanism. Generally, the parameters can be divided into four groups, concentrating properties and conditions of different components involved in the process. The four categories are the particles, the slurry, the wall surface and lastly the particle-wall contact. A detailed explanation of those parameters is given by Javaheri et al. [15].

From all of them, it is clear that any parameter that has to do with the particles and the carrier liquid, cannot be changed. On the other side, the operational conditions like velocity, and concentration, can be changed but the only way is by lowering both of them, in order to keep the heterogeneous flow regime. However, this solution would result in slower discharging operations, which is not economically beneficial for the company. In reference to the pipe geometry, the pipe diameter and bend radius cannot be increased because of space limitations. Also, an enlarged pipe in the same space would

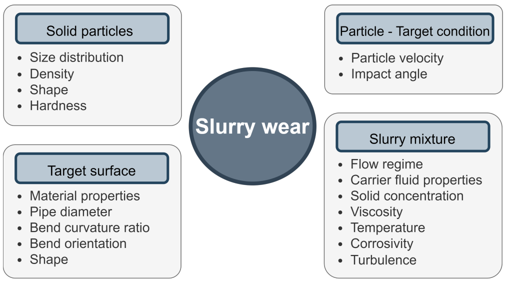

Fig. 2: Slurry wear parameters.

result in a lower bend radius, which increases wear, and secondly, the required power would be much higher. Considering all of them, the only one that can be changed is the material of the bend pipe.
As mentioned before, the ductility of the target surface plays a major role for the wear. In general, brittle materials show much greater wear performance. The prevailing opinion is that hardness is one of the most significant factors, influencing the erosion behaviour of a wide variety of materials, [34, 35, 36, 37, 38, 39, 40, 41]. However, it should not be considered as the only representative parameter since there are cases for which material hardness does not constitute a reliable indicator to predict erosion rate, [36, 42, 43, 39, 34, 15].

IV WEAR PATTERN

When reviewing the available literature, one may notice that the majority of studies were carried out using gas as a carrier fluid. There has been no available research on large-scale bend pipes and especially utilizing dredging operational conditions. However, this section gathers numerous projects that investigated the wear in bend pipes under various conditions. The outcome is then used to estimate the wear pattern of the bow coupling bend pipe.

A Background

Peng and Cao [14] performed simulations for 34 scenarios, using different pipe sizes, bend orientations, bending angles, bending ratios, mixture velocities, particle sizes and flow rates. First of all, they compared predicted and experimental penetration rates for elbows for five erosion models with different particle-wall rebound models. Even though there were significant differences regarding the magnitude of penetration rates for the models, all of them showed clearly the gradual increase of erosion along the bend pipe maximizing their values at the outlet region. A representative example used 0.2 mm particles and 0.2 kg/s mass flow, with 10 m/s mixture velocity in a 40 mm pipe diameter. Initially, there was a slight increase until 35^o, after which the wear rate remained almost stable until 60^o, where the significant increase started and despite the fluctuations, there was a peak at about 75^o and another one even higher just before 85^o.

The aim of Zeng et al. [27] was to investigate both erosion and corrosion in a 90^o elbow with a 50 mm inner diameter. The focus of this study was on the experiment, but the authors proceeded to CFD analysis to understand better the behaviour of the mixture in the bend and then relate that to their experimental results. The mixture had 4 m/s velocity and was composed of water and sand particles of 450 µm diameter, 1.2 % mass concertation and 0.235 kg/s mass flow rate. In the upstream straight pipe, the values at the innermost side are greater than those at the outermost side, whereas, in a downstream straight pipe, the opposite occurs. The erosion at the elbow is significantly higher at the outer wall and escalates along the flow direction reaching the maximum value at the outlet area. In that area, the sand concentration is higher because of the inertia that led them to the outer wall and the second reason is the secondary flow effect which is more intense near the outlet and drives the sand particles towards the outer wall. Both reasons result in a frequent impact of sand particles on the wall and consequently cause severe erosion there. The inertia is mainly responsible for the erosion along the axis of symmetry at the outer wall, increasing gradually towards the outlet. The secondary flow also increases the erosion at the centre of the outer wall but that happens after about 45^o, where the secondary flow strengthens until the outlet, where it is fully developed. As a result, the erosion does not only increase towards the outlet but also spreads to the sides of the bend, with a decreasing magnitude, though. Overall, the erosion appears to be higher at the bottom of the bend inlet, which decreases significantly by entering the bend. About the outer wall, there is a tiny increase until about 20^o, then a more important rise until about 60^o, where the erosion rate increases more dramatically until about 80^o where the maximum rate was measured on the elbow. Also after about 60^o, the erosion rate has a significant magnitude towards the sides of the outer wall. Finally, the measurement points at the downstream pipe showed significant damage on the outer wall and the sides, indicating that the wear pattern continues after the bend exit. These results seemed to have a great agreement with the study of Khan et al. [26], who created similar operating conditions in order to validate their model.

Additionally to those, there are more studies that found the maximum erosion at the outlet of the outer wall, with the wear gradually expanding to the sides of the pipe along the direction of the flow, [44, 45, 46]. Chen et al. [47] used discrete element model (DEM) and CFD techniques to evaluate the wear rate of different elbow configurations with 0.15 mm sand particles and found that the outlet was the most eroded one for all elbows configurations. Moreover, Zhang et al. [48], using a bend with 3.5 curvature, found the maximum impact force at the bend outlet and the next pipe, while the first 27^o of the outer wall was intact. Mouketou et al. [49] and Blanchard et al. [50] using a standard elbow pipe found similar wear profiles, with the maximum values at 87^o and 85^o of the outer wall, respectively. Khan [51] stated that the first about 25^o did not show any wear sign, but between 25 and 65^o, the simulation predicted low wear while the area after 65^o had medium wear rates, to find the high wear at about 90^o. Kannojiya and Kumar [52] showed that the erosion rate starts at around 30^o at the centerline of the outer wall and continues on the outer wall, until about 90^o. Lastly, Blanchard et al. [50] claimed that the maximum wear location varies from 75^o to 105^o, and suggested that it should be expected at $85^o \pm 15^o$ around the bend on the outside surface on the line of symmetry, independent of particle size or bend geometry.

In reference to studies with gas as a carrier fluid, their lower density and viscosity influences the location of the maximum wear in the bend. Thus, the sand particles can more easily deviate from the fluid streamline and impact the outer wall. Specifically, simulations and experiments showed that for various flow conditions in bend pipes with mainly a 1.5 curvature ratio, the maximum wear is located between 45^o and 60^o, [53, 54, 55, 56, 57, 58, 59, 60, 61].

Based on everything discussed in this chapter and mainly the previous section, the wear pattern of a bend pipe is illustrated in Fig. 3. Precisely, this figure shows the outcome of the literature review if it is directly applied on a bend pipe with a small diameter and 1.5 and curvature ratio, like most projects used. It can be seen that there is a total of four wear zones, according to the expected level of wear.

The critical zones are separated based on the wear rate differences, without meaning that the wear rate in a zone is constant. The literature clearly showed that there is very low wear at the inlet of the pipe, the first about 25^o, and the entire inner wall. Then, medium level of wear was found after 25^o at the outer wall, and finally, after 60^o, the erosion increased significantly, with the maximum rates being in the last 10^o, at the outlet area.

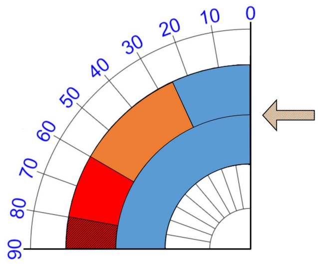

Fig. 3: Bend pipe wear zones: Literature outcome.

B Wear zones

Since there is no slurry wear research for the specific bow coupling bend pipe, five adaptation steps are considered in order to bring the literature result as close as possible to the real bend pipe.

Step 1: Geometry: The first step was to apply this configuration on the bow coupling bend pipe, 1 m diameter and 1.7 m bend radius. The influence of geometry on the overall wear is described in the fourth step.

Step 2: Inner wall: Even though the literature did not show any sign of wear there, in reality, there is wear, at least at the inlet, because of the heterogeneous regime in the upstream pipe. Consequently, the inner wall is considered separately, with green colour, whose wear rate is between the low and the medium wear zones.

Step 3: Inner wall - Inlet: Considering a mixture with lower velocity, higher concentration and even larger particle size, the flow regime and consequently the wear profile of the straight pipe would be different. A flow with saltation or even a sliding bed would increase significantly the wear rate at the bottom of the pipe. As a result, it is assumed that the maximum wear of the upstream pipe can reach the rates of the medium wear zone, and this is expected to continue for the first 30^o of the bend inner wall.

Step 4: Outer wall The larger the diameter is the lower and more distributed the overall wear is going to be. Also, a larger worn area with lower wear rates is the result of an increased curvature ratio, which can also shift the maximum wear slightly away from the bend outlet. Another part that is related to the geometry is the short length of the upstream pipe, which can lead the particles to unexpected locations. At the same time, generally, a high concentration of particles expands significantly the worn area, especially with smaller particles which are driven by the carrier liquid and can hit any location of the pipe. Small particles can also increase the wear of the outer wall and the sides close to the outlet and probably the downstream pipe, because of the secondary flow. All those parameters show that the maximum wear can be found a bit before the bend outlet, while the area with noticeable wear can be larger. Thus, the medium and high erosion zones are shifted by 5^o towards the inlet.

Step 5: Downstream Pipe The connection from the female to the male part is expected to generate very intense turbulence in that area, and even slightly affect the flow in the bend pipe, depending on the orientation of the floating pipeline. It was decided to add 5^o to the maximum wear zone at the outer wall. In reference to the inner wall, the last 5^o are expected to show medium wear.

To sum up, the first 20^o of the outer wall is expected to have the lowest wear of the pipe, after which the wear rates take an upward trend until 55^o. After that point, there is an even more significant increase in the wear rates, which reach a maximum in the last 20^o of the bend pipe. The inner wall is not expected to show significant wear rates, with the exception of the two bend ends, first 30^o and last 5^o, which are affected by the flow situation in the corresponding connected pipes. The lack of specialized wear analysis for a project increases considerably the level of uncertainty, and therefore all steps tended to be more conservative.

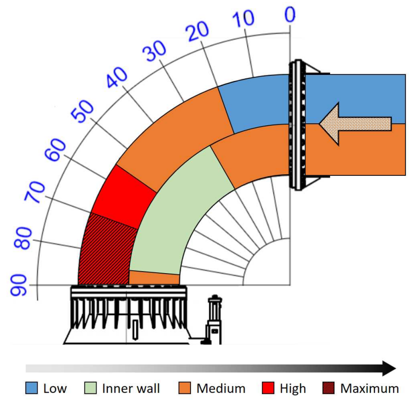

Fig. 4: Bend pipe wear zones.

The following assumptions were considered for the final wear pattern of the bow coupling bend pipe. The last two assumptions are influenced by Peng and Cao [14], Zeng et al.[27], and Khan et al. [26].

1. Corrosion is uniformly distributed in the entire bend pipe, thus, it does not constitute an indicator of the maximum wear in the bend.
2. The literature outcome can be considered as a reference model, despite the significant size difference between the research bend pipes and the bow coupling bend pipe.
3. The previous straight pipe has a fully developed heterogeneous flow regime, despite its very short length.
4. The upstream pipe can reach the same wear rate as the “Medium zone”.
5. The wear rate from the upstream pipe will continue for the first 30^o of the inner wall.
6. The maximum wear in the pipe shifts from the inner to the outer wall at 25^o, between the overlap of the two medium zones.
7. The wear rate on the outer wall increases non-linearly along the flow direction, from the inlet to the outlet. The wear slope in every zone becomes steeper, from the “Low zone” to the maximum wear in the “Maximum zone”.
8. The maximum wear rate of the “Maximum zone” is five times the maximum wear rate of the “Low zone”.

V CONCEPT DEVELOPMENT

This section describes the process of generating concepts and then compare them in order to end up with the optimum solution.

A Limitations

Limitation 1: The position and orientation of the inlet and outlet of the bend pipe are standardised by the company. The maximum bend radius is 1700 mm, for the highest and lowest connection to the tower and the female part, respectively, see Fig. 5.

Limitation 2: The area shown in figure 6 with pattern lines shall be remained free for the hoisting cable and chains. At the platform’s level, the free space around the female part should be 200^o.

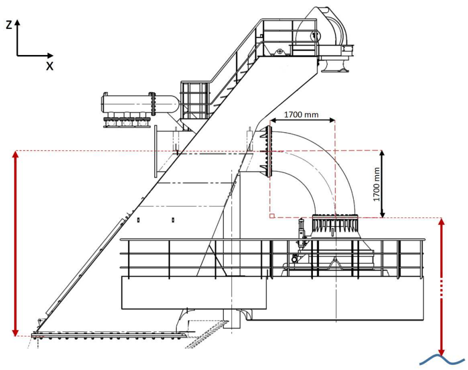

Fig. 5: Limitation 1: Connection points.

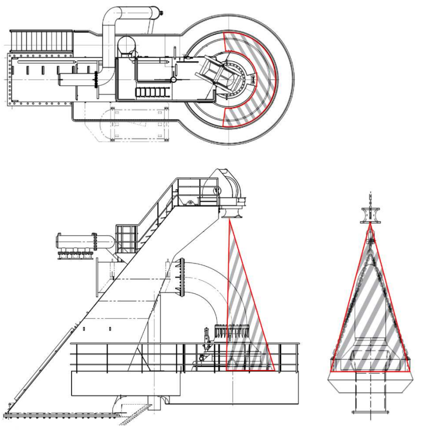

Fig. 6: Limitation 2: Hoisting cables space.

B Requirements:

- *Safety:* All concepts should have as a priority the safety of the people.
- *Reliability:* The system should be able to function under loading, without failure for the expected lifetime.
- *Flow redirection of 90-degrees:* The bend pipe should redirect the flow from the horizontal axis coming from the tower, to the vertical axis leading to the female part.
- *Inner diameter:* The inner diameter of the bend pipe should be 1 m.
- *Same hoisting and connection procedure:* The connecting procedure with the floating pipeline remains unchanged so the existing floating pipelines can be attached.
- *Access for maintenance and pipe inspection:* People should be able to go close to all pipe parts for inspection and maintenance wherever is needed.
- *Separated flow and structural function components:* The pipe components that are responsible for the flow of the dredged material should not be considered on the support function or affected by the support demands of the system. Vice versa, the structural parts should not be considered for the flow function and they should not be affected by that.

C System functions

The separation of the two functions of the system comes with crucial advantages for the new concept. Firstly, the last requirement aims at the minimisation of the loads on the bend pipe. In that way, a harder material can be used, which can lead to lower wear rates and consequently longer lifetime of the pipe. Also, fewer loads on the bend pipe correspond to lower required strength and therefore the minimum operational thickness can be less, before being replaced, extending the pipe's lifespan. Furthermore, the amount of material that has to be replaced is reduced because only the flow function parts will have to deal with the wear of the pipe. The support function parts will be permanent as the wear does not affect them. Bellow, the functions and their sub-functions are defined, and the proposed solutions are concentrated in the morphological chart, see Fig. 7.

Support Function (Permanent): *"Provide sufficient strength and stiffness to the construction in the operational and out-of-service situation."*
1.1. Discharge switching
1.2. Strength – Stiffness

Flow Function (Replaceable): *"Transport of the dredged material which comes from the horizontal pipe in the tower and goes to the female part, vertically downwards"*
2.1. Redirection of the flow
2.2. Wear of the pipe

Flow function comparison

Before developing the concepts, it was decided to approach the two functions separately. What is meant is that the solutions for the two sub-functions of the flow function are compared and their best combination of flow redirection and pipe wear solutions, is applied to all concepts. The reason behind that decision was to focus on the support function where much more solutions can be unfolded, considering that some of them can even be combined. On the other side, the flow function has limited proposed solutions, among which some have a clear advantage over others, that may even be infeasible under certain circumstances.

Flow redirection: The two options are the regular cast bend pipe and the sections bend pipe, see Fig. 7. Firstly, the sections' bend pipe affects the flow and specifically worsens the wear rate in the pipe, at the connection points. Thus, the regular cast bend pipe is preferred over the sections' bend pipe.

Pipe wear: There are three solutions about the wear in the pipe, the increased thickness, the bend flip and finally the chocky bars, see Fig. 7. To begin with, the last solution is a straightforward method for extending the lifetime of the bend, as the bars are worn down before reaching the pipe wall surface. The top part of a chocky bar is made of a very hard material which provides high wear resistance, while the bottom part is mild steel to allow the welding process. Chocky bars are made of very hard and expensive material which also needs to be welded in the pipe. Welding of straight bars in a bend pipe requires numerous working hours and it does not result in a smooth surface, which is expected to affect the flow in the pipe and probably the wear profile. Also, the replacement requires a lot of machining of the inner surface of the pipe to make it appropriate for the welding of the new ones.

The other two solutions are significantly cheaper compared to chocky bars and they can always be recycled, as well. Both of them are developed based on the expected wear pattern of the bend pipe, that the inlet and the outlet of the outer wall are the two extreme wear locations, minimum and maximum respectively. The first solution increases the thickness of the outer wall along the bend pipe while the second option flips a regular bend pipe at a certain minimum thickness, so the inlet side goes to the outlet and vice versa. However, the casting of those pieces requires a minimum wall thickness of 25 mm. Also, one of the assumptions for the pipe wear pattern was that the outlet has five times the thickness reduction of the inlet. The last two points make the first solution practically infeasible because the maximum thickness at the outlet of the outer wall should be exceptionally high,

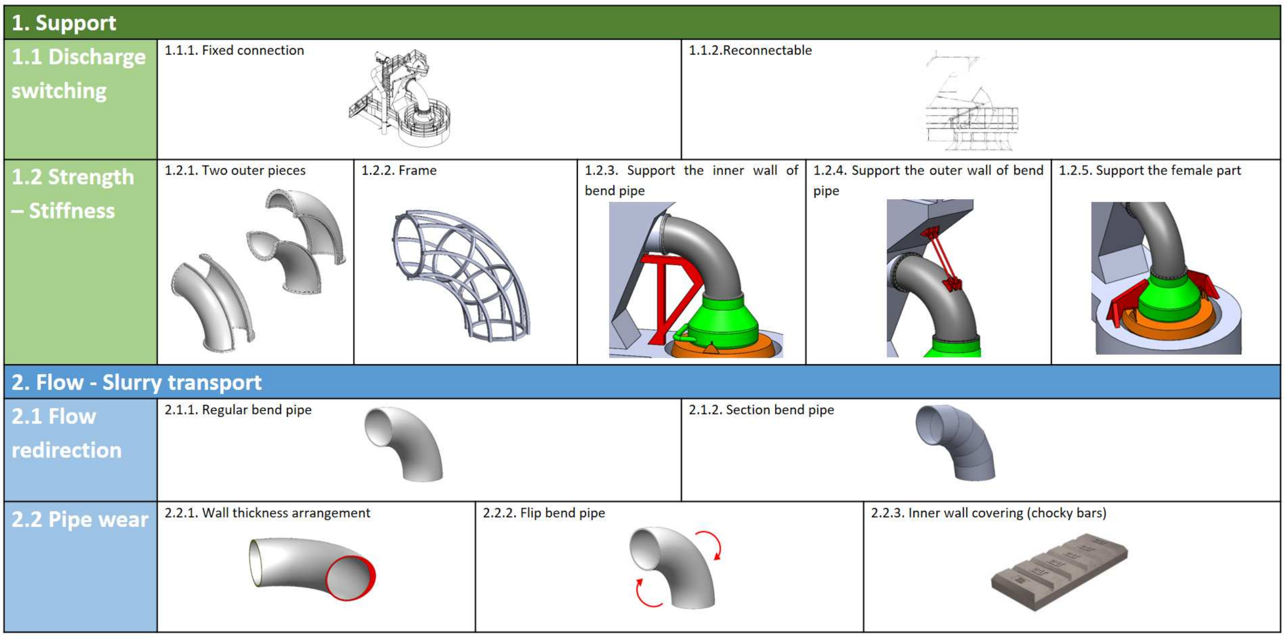

Fig. 7: Morphological Chart.

to achieve a uniform thickness reduction for the outer wall.

The "Flip bend pipe" takes advantage of the fact that the two extreme wear rates of the bend are in the first and last 20^o of the outer wall. The flip of the bend pipe will bring the inlet with the minimum thickness reduction to the outlet, where the wear is maximum, and conversely, the outlet with the maximum thickness reduction will be placed at the inlet where the wear is the lowest of the pipe.

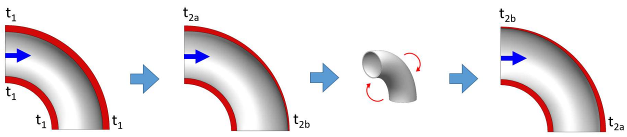

Fig. 8: Flip bend pipe.

Fig. 8 depicts the steps that the bend pipe should follow, showing at the same time the thicknesses at the ends of the bend pipe, where $t_1 > t_{2a} > t_{2b}$. Starting from the left side, this is the initial pipe with uniform thickness, while the next step shows the reduction of the wall thickness in the pipe. The thickness t_{2b} constitutes the "flipping thickness", and it is expected in the last 20^o. It can be seen that after flipping the pipe there are a lot of similarities with the first solution, in reference to the increased thickness along the outer wall. Finally, compared to the other two solutions, the second one is the most cost-efficient, with a longer lifetime and less replaced material.

Overall, the optimum solution for the flow function is the regular cast bend pipe, which has to be flipped after a certain point. This combination is considered for all concepts.

D Concept selection

Table 1 shows all the combinations for concepts, while Fig. 9 presents the eight developed concepts.

Variable requirements

For the comparison of the generated concepts, eight variable requirements are considered, with different weights representing the project's priorities.

TABLE 1: CONCEPTS SOLUTIONS COMBINATIONS.

	1. Support							2. Flow	
	1.1.1	1.1.2	1.2.1	1.2.2	1.2.3	1.2.4	1.2.5	2.1.1	2.2.2
Concept 1	■		■					■	■
Concept 2	■					■		■	■
Concept 3	■		■	■		■		■	■
Concept 4	■			■				■	■
Concept 5	■		■		■			■	■
Concept 6	■						■	■	■
Concept 7		■					■	■	■
Concept 8		■	■					■	■

1. *Cost of flow/replaceable parts (weight: 5):* The "flow function" parts are to be replaced on a much more frequent basis than the permanent parts because they are exposed to the slurry wear. Specifically, the cost is determined by the amount of material, manufacturing process and additional fabrication to make the parts appropriate for installation.
2. *Ease of replacement (4):* The main replacement parameters are the number of steps, the required installation equipment, the fabrication that has to take place and the ease of connection.
3. *Cost of structural/permanent parts (4):* The cost of the "support function" parts is determined by the manufacturing process, type of support, structural complexity and lastly if it is a known or a new concept.
4. *Loading on bend pipe/replaceable part (4):* The strength and stiffness of the support should be as high as possible in order to minimize the loads exerted on the "flow function" parts.
5. *Bend pipe inspection accessibility (3):* This requirement refers to the ease of inspecting the replaceable parts from their outer wall surface.
6. *Required support on the existing structure (3):* Even though the new design can be supported by surrounding existing components, some of

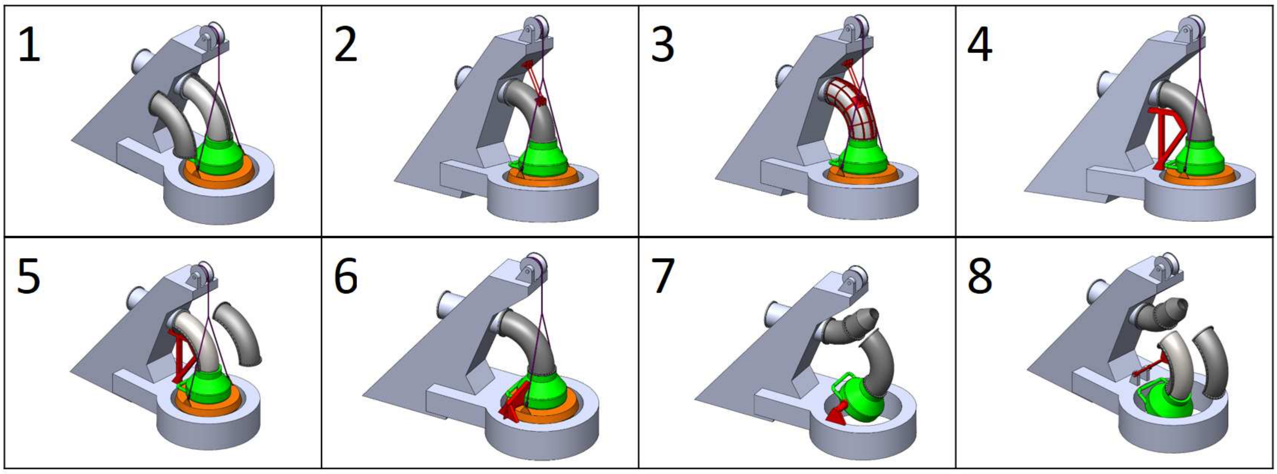

Fig. 9: Concepts.

them might not have the required strength and as a result, additional support will be necessary.

7. *Safety - Protection (2):* This requirement covers four areas: leakage in case the “flow function” parts fail, safe walking space for people, moving parts close to crew areas, and protection of the bend pipe from the outer environment.
8. *Ease of applying on existing vessels (1):* The required modifications of the vessel to install the new design.

Concepts rating

At this stage, the concepts are evaluated for each of the defined factors, on a scale of one to five. The author and four experts from the company Royal IHC, evaluated all concepts and the final averaged rates are presented in Table 2.

TABLE 2: CONCEPTS’ AVERAGE RATES.

		Concepts							
Variable Requirements	Weight	1	2	3	4	5	6	7	8
Cost of replaceable parts	5	5.0	3.2	3.2	2.6	5.0	3.2	3.2	5.0
Ease of replacement	4	3.4	3.6	2.4	2.8	3.4	4.6	4.4	2.8
Cost of permanent parts	4	3.8	4.8	3.0	4.2	3.2	3.8	2.8	2.2
Loading on replaceable parts	4	3.8	2.4	3.6	3.2	4.8	4.2	3.8	4.2
Bend pipe inspection accessibility	3	1.6	4.2	4.2	4.2	1.6	5.0	5.0	1.8
Required support on existing structure	3	5.0	3.8	3.8	3.4	3.4	2.8	2.6	3.2
Safety - Protection	2	4.8	2.8	4.0	2.8	4.2	3.0	2.4	3.4
Ease of applying on existing vessels	1	5.0	4.0	3.4	3.2	3.0	2.0	1.8	2.2
Summation:		**103.4**	**92.8**	**87.4**	**85.4**	**97.0**	**97.8**	**89.4**	**85.8**

As can be seen, Concept 1 had the highest score, followed by Concept 6 and Concept 5, respectively. A brief sensitivity analysis was performed in order to define how the final decision is influenced by the change of weight for some factors. Overall, by changing up to two weights at a time, the only scenario that finds Concept 6 as the best one, is when the weight of "Bend pipe inspection" increases to 5.

Final concept

Concept 1 uses two connected cast pieces for support, which are also bolted to the previous straight pipe, at the inlet, and the female part, at the outlet. For the flow function, there is a cast bend that has the role of a liner, and it has to be flipped once, before being replaced. Figure 10 shows the final concept with one support piece disconnected, the liner with pink color, and lastly the female part and the floating body with green and orange color, respectively.

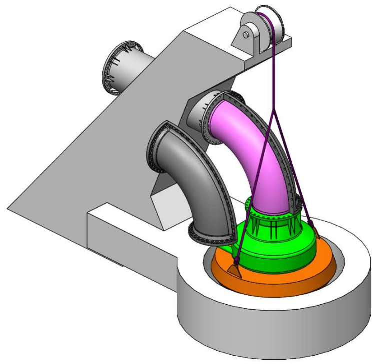

Fig. 10: Final concept.

Inspection plugs

As shown from the rating table, the inspection of the bend pipe constitutes the main weakness of the selected concept. It was decided to add eight inspection plugs at critical locations according to the expected wear pattern of the bend pipe. Fig. 11 shows the locations of the inspection plugs before flipping and their configuration in the two support pieces.

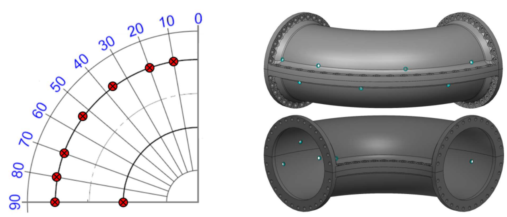

Fig. 11: Inspection plugs.

VI Proposed design

This section starts with the analysis of the new design, which determines the final geometry and materials. Having all the parameters of the proposed design, it is then compared with the current one.

A Design analysis

For the analysis of the system, specific loads shall be applied to the new design to evaluate its behaviour under real circumstances. From this analysis, the final geometry and materials are going to be determined.
Figures 12 and 13 show the two models for the analysis. The first model is the inner bend pipe, the liner, and the second one is the support of the bend pipe. The aim of Model 1 is to find the minimum operational thickness before the bend pipe is replaced. On the other hand, the second model, as the support of the system, should be able to withstand various loading scenarios. Model 2 consists of the straight pipe, coming from the tower, the two support pieces and finally the female part.

Fig. 12: Model 1.

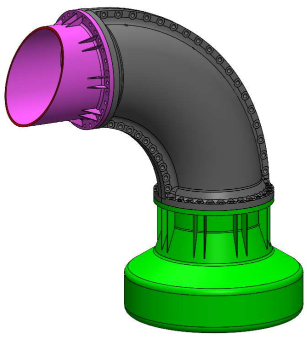

Fig. 13: Model 2.

Loads on the system:

Pressure: The operational internal pressure is 25 bar and it shall be applied on both models, the inner surface of the liner and the support pieces.

Floating pipeline: The connection of the floating pipeline on the female part constitutes the most significant load of the bend pipe. The load from the hose depends on the mechanical properties, weight, buoyancy, waves, pulling force, and lastly its relative motion with the vessel. Due to the complex behavior of that part, the company Royal IHC determined a load as a rule of thumb, according to the internal diameter of the bow connection. The F_{Hose} for 1 m diameter is 1 MN, and this load should be applied in various directions, as the ball connection allows for a maximum angle of 15^o from the vertical axis. Fig 14 shows the load in the cross-section side view of the male part, while from the top view the four considered directions can be seen.

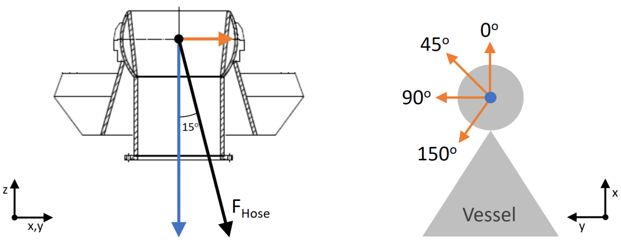

Fig. 14: Floating pipeline force.

Flow force: This load is generated from the redirection of the flow, resulting in a momentum force exerting on the bend. Based on the conservation of momentum, the rate of change of momentum of a body is equal to the net force acting on the body.

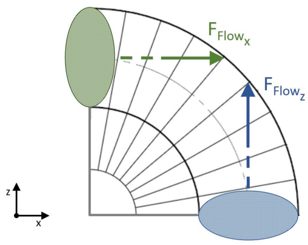

Fig. 15: Flow force.

Horizontal flow force:

$$F_{flow_x} = \rho \cdot A \cdot (v \cdot cos(\theta_1)\breve{}\, v \cdot cos(\theta_2))^2 = +5103541N \tag{3}$$

Vertical flow force:

$$F_{flow_z} = \rho \cdot A \cdot (v \cdot sin(\theta_1)\breve{}\, v \cdot sin(\theta_2))^2 = +5103541N \tag{4}$$

Where ρ = fluid density = $1800kg/m^3$; A = Cross section area of the pipe = $0.79m^2$; v = Flow velocity = $6m/s$; θ_1 = Inlet angle = 0^o; θ_2 = Outlet angle = -90^o.

Both loads are distributed on a circular area of 1 m diameter, like the inner diameter, and they should be applied on the outer wall. Even though the flow forces act directly on the liner, these loads are considered for both models as the only support of the liner is the outer shell.

Winch force: This is an accidental load when the operator lifts the floating pipeline more than is needed during connection, and consequently, the floating body pushes the female part upwards. As the maximum pulling force from the winch is 450 kN and the floating line mass is about 10 t, the resulting force is 351.9 kN.

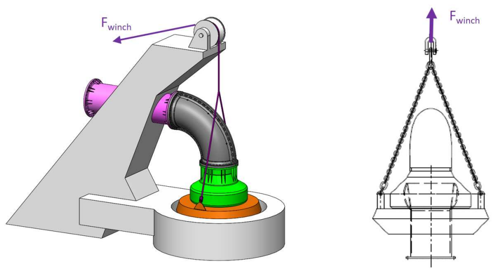

Fig. 16: Winch force.

Wind: The wind pressures and the corresponding velocities are calculated in accordance with "DNVGL-ST-0378".

- Occasional wind (24 m/s; 360 Pa): 1728 N
- Extreme wind (44 m/s; 1200 Pa): 5760 N

Vessel motions: The standard "DNV GL rules for classification of ships RU SHIP Pt.3 Ch.4 Sec.3", was used for the calculations and therefore those are extreme values for ship accelerations. Table 3 shows the three combinations of the vessel accelerations, which are computed for the relative position of the bend pipe on the vessel. These accelerations are the result of all ship motions, heave, sway, surge, yaw, pitch, and roll. The gravity is not included.

TABLE 3: LOAD COMBINATIONS FOR VESSEL ACCELERATIONS.

Load combination	α_x	α_y	α_z
Beam sea	0	$\alpha_{y_{Beam}}$ = 6.09 m/s²	$\alpha_{z_{Beam}}$ = - 4.59 m/s²
Oblique sea	$\alpha_{x_{Oblique}}$ = 0.80 m/s²	$\alpha_{y_{Oblique}}$ = 3.65 m/s²	$\alpha_{z_{Oblique}}$ = - 4.92 m/s²
Head sea	$\alpha_{x_{Head}}$ = 1.33 m/s²	0	$\alpha_{z_{Head}}$ = - 4.92 m/s²

Load cases:
According to DNVGL-ST-0378, a total of three load cases should be considered, for which the required safety margins differ, in order to make the nominal safety based on the probability of the loading.

- Case I: Regular operational conditions
- Case II: Occasional operational conditions
- Case III: Exceptional conditions

Load combinations
Table 4 concentrates all load combinations (LC). It can be seen that there is a total of eleven scenarios that are analysed, using finite element methods (FEM).

B Results

Figures 17 and 18 depict the simulation results of the two models. Specifically, 17 is the liner with the lowest allowable thickness, which was found to be 4 mm. 18 depicts LC5 which caused the highest stresses on the second model and therefore played a critical role in the final geometry.

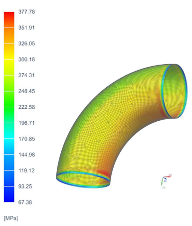

Fig. 17: LC 1.

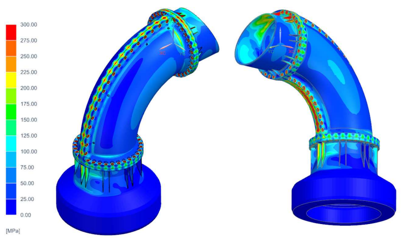

Fig. 18: LC 5.

The liner is a regular cast bend pipe with an original thickness of 25 mm, the minimum thickness for casting, which corresponds to 1.75 t. There is no need for a thicker liner because this would lead to larger and heavier structures, not only for the bend pipe but also for the connected pipes. In reference to the material used for the liner, there was a series of five different cast materials suitable for abrasive and erosive environments, from which *Wearmet S2* was considered appropriate. The inevitable loads on a bend liner do not allow for the selection of materials with very low impact value. Although it is a slightly more expensive material compared to *Wearmet S1* that the current design uses, it provides almost twice material hardness, one of the most important wear parameters.

For the structural parts, cheaper materials could be selected as they do not require high wear resistance. For the straight pipe and the female part, the structural steel *S355J2* was used, while for the two cast pieces *G28Mn6 (+QT1)* was selected. The total mass of the support pieces is about 3.62 t, with a wall thickness of 25 mm and flanges of 80 mm.

To determine the flipping and replacing thicknesses, the last two assumptions from Section IV were considered. The maximum wear rate of the "Maximum zone" is five times the maximum wear rate of the "Low zone", and the wear of the outer wall increases exponentially from the inlet towards the outlet. This means that after flipping the inlet and outlet areas of the outer wall are expected to show the maximum loss of material. To maximize the total wear thickness reduction it was decided that the inlet and the outlet should reach the minimum operational thickness at about the same time. The total wall thickness reduction is defined as the summation of material loss before flipping and before replacing the pipe at the "Maximum zone". Eventually, the liner with the support pieces have to be flipped when the wall thickness in the "Maximum zone" reaches 8 mm, see t_{2b} in Fig. 8. The final expected thickness is 4 mm and 4.48 mm for the outlet and inlet of the outer wall, respectively. Finally, the total wall thickness reduction of the outlet outer wall is 34.6 mm.

C Comparison with the current design

This section compares the new with the current design in three aspects according to the main objective of this study, the replaced amount of material, pipe lifetime and cost.

Lifetime: The current design has an initial thickness of 35 mm and considering the minimum operational thickness of 24 mm, the maximum thickness reduction is 11 mm. On the other side, the new design is expected to reach 34.6 mm as a total thickness reduction. That case corresponds to about a three times longer operational lifetime. Furthermore, the fact that the new design can be made of a more wear-resistant material prolongs, further, its lifetime.

Replaced amount of material: The mass of the current design is about 3 t, while the new design weighs about 5.37 t, 3.62 t for the support pieces and 1.75 t for the liner. Although the new design initially requires two more tons, only 1.75 t will be needed for its first replacement, because only the liner has to be replaced. This corresponds to 58 % of the amount of material that the current design has to replace every time.

Cost: As there is no detailed analysis of the cost, the comparison was made based on estimations, for the support pieces and the liner. Starting with the support pieces, they need slightly more amount of material, they require more machining because of the flanges and the number of bolts, and lastly, the fact that they are two pieces increases the casting cost. However, considering also the cheaper material, it is estimated that the support pieces will cost 1.5 times as much as the current design. Regarding the liner, the new material is slightly more expensive than the current one but its production requires 1.25 t less amount of material. Also, it is a simple cast design and there is no need for afterwards fabrication. It is predicted that the liner will cost half of the current design. Based on those two estimations, it can be assumed that the new design will initially cost twice as much as the current one. Although the initial cost is higher, the new design will depreciate the cost for the support pieces before the first replacement of the liner, because by that moment the current design would have already been replaced twice.

Thus, the new design has the potential to at least triple the current operational period of the bend pipe, requiring less than 60 % initial amount of material and reducing the replacement cost by 50 %.

VII CONCLUSION

This work provides information about the slurry wear in bend pipes, defining also the wear pattern that can be expected in large-scale bow-coupling bend pipes. Furthermore, a new design is proposed which achieves a reduction of the replaced amount of material and extension of its lifespan, in an economical way. Precisely, the new design has at least three times the lifetime of the current design, while for every replacement the amount of material and production costs decreased to about 60 % and 50 %, respectively. In addition, the fact that the total thickness reduction is 140 % of the original thickness shows a careful management of the material, which leads to a more sustainable solution.

In the future, research on slurry wear in large-scale bend pipes should be initiated, as well as a more intensive campaign regarding the protection of dredging pipes needs to be carried out. Finally, a specific wear analysis of the bow coupling bend pipe, including the geometry of the previous and the next pipes, is necessary, but also a detailed loading analysis that considers the complex behaviour of the floating pipeline.

TABLE 4: LOAD COMBINATIONS FOR ALL LOAD CASES.

LC	Case	Model	SF	Gravity	Pressure	F_{Flow}	F_{Hose}	Wind		Vessel accelerations			F_{winch}
								Occasional	Extreme	α_x	α_y	α_z	
1	I	1	1.5	-z	1	1	0	0	0	0	0	0	0
2	I	2	1.5	-z< 2°< +x	1	1	0°	0	0	0	0	0	0
3	I	2	1.5	-z< 5°< +y	1	1	45°	0	0	0	0	0	0
4	I	2	1.5	-z< 5°< +y	1	1	90°	0	0	0	0	0	0
5	I	2	1.5	-z< 2°< -x	1	1	150°	0	0	0	0	0	0
6	II	2	1.33	-z	1	1	0°	+y	0	$0.3\ \alpha_{x_{Head}}$	0	$-0.3\ \alpha_{z_{Head}}$	0
7	II	2	1.33	-z	1	1	45°	+y	0	0	$0.3\ \alpha_{y_{Beam}}$	$-0.3\ \alpha_{z_{Beam}}$	0
8	II	2	1.33	-z	1	1	90°	+y	0	0	$0.3\ \alpha_{y_{Beam}}$	$-0.3\ \alpha_{z_{Beam}}$	0
9	II	2	1.33	-z	1	1	150°	+y	0	$-0.3\ \alpha_{x_{Head}}$	0	$-0.3\ \alpha_{z_{Head}}$	0
10	III	2	1.1	-z	0	0	0	0	+y	0	$\alpha_{y_{Beam}}$	$-\alpha_{z_{Beam}}$	0
11	III	2	1.1	-z	0	0	0	0	0	0	0	0	1

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B

Trailing Suction Hopper Dredger (TSHD)

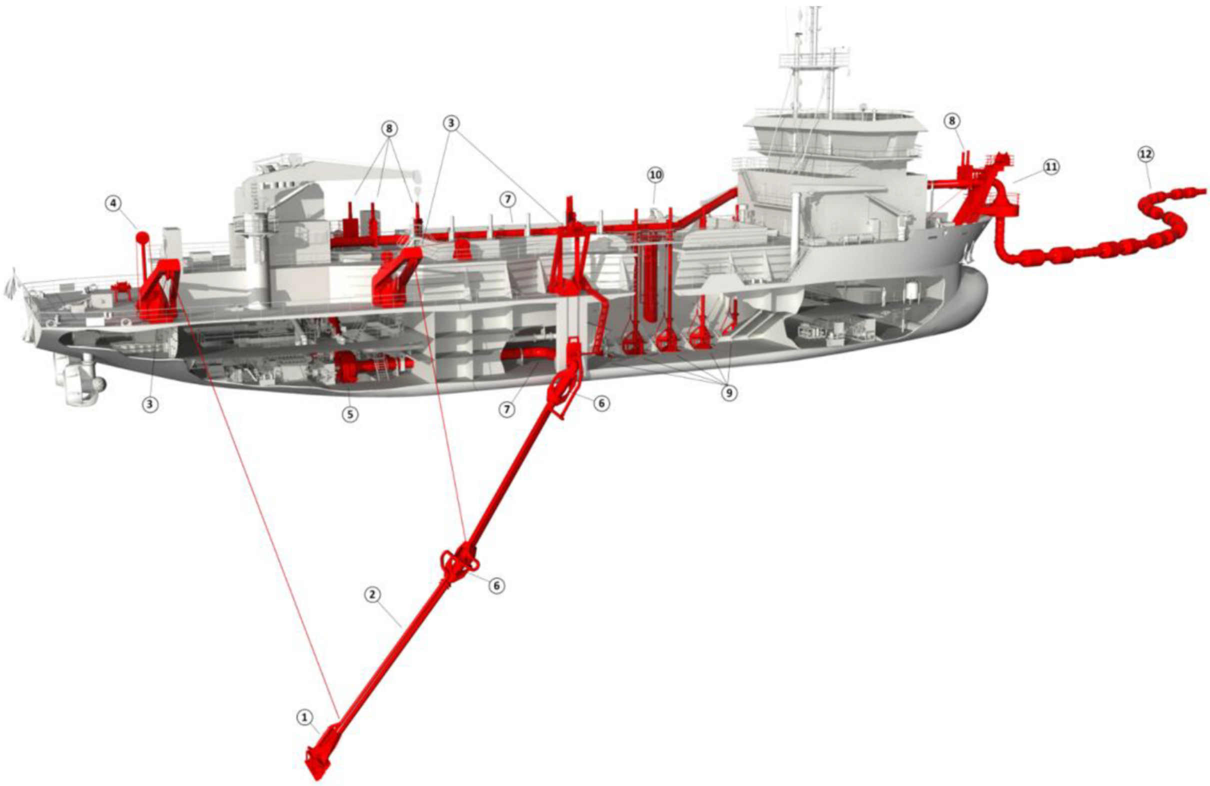


Figure B.1: TSHD: Main dredging equipment.

Table B.1: Vessel details

N.	Equipment	Description
1	Draghead	Draghead is at the bottom when dredging. It loosens the soil, before it is transported through the suction pipe
2	Suction pipe	The total installation of the pipe consisting of the lower pipe, upper pipe.
3	Gantry	Lifting equipment to be able to lower or hoist the suction pipe alongside the vessel.
4	Swell compensator	Ensures the constant tension of the hoisting wire during sailing of the vessel. In that way, it keeps the draghead close to the seabed.
5	Onboard pump	Pump installation to create pressure difference in the pipes to transport the mixture.
6	Onboard rubber hose	Rubber hose in the pipeline ensures that the pipeline has multiple points to be able to bend.
7	Onboard pipeline	Guides the dredge mixture at the inside of the TSHD.
8	Dredge valves	Dredge valves regulate the route of the dredging material follows through the system. The specific ones, switch the discharging between the bow coupling and the nozzle.
9	Bottom doors	One of the discharging ways. Opening these doors, results in dumping via the bottom of the vessel.
10	Overflow	The overflow is a way to get rid of the excess water that remains on top in the hopper, while the heavier sand particles settle at the bottom.
11	Bow coupling	The point that the floating pipeline is connected for onshore discharging. The same area usually hosts a nozzle to spray the mixture.
12	Floating pipelines	Pipelines that are able to float and transfer the dredged material onshore. The end of the pipeline connects to the female part of the bow coupling.

C
Slurry wear

C.1. Types of soil

Main type of soil		Particle size	
		Identification	size in [mm]
Boulders	Granular Non-cohesive	-	> 200
Cobbles		-	200 – 60
Gravel		Coarse	60 – 20
		Medium	20 – 6
		Fine	6 – 2
Sand		Coarse	2 – 0.6
		Medium	0.6 – 0.2
		Fine	0.2 – 0.06
Silt	Cohesive	Coarse	0.06 – 0.02
		Medium	0.02 – 0.006
		Fine	0.006 – 0.002
Clay		-	< 0.002

Figure C.1: Soil types and particle sizes, [4].

C.2. Slurry Transport: Velocity and concentration profiles

Velocity and concentration distribution according to [4]:

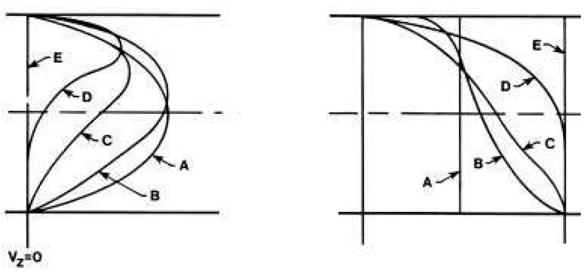


Figure C.2: Velocity and concentration profiles, [4].

- Curve A is a symmetric suspension flow rate, the so-called homogeneous suspension for fine materials or coarse materials at high flows. In this region, the slurry and water curves are essentially parallel, indicating fully suspended homogeneous turbulent flow.
- Reducing the flow rate results in an increase of the in situ concentration at the bottom of the pipe and a decrease in the top of the pipe with a corresponding skewing of the velocity profile towards higher velocities in the top and lower velocities in the bottom. This results in curve B, an asymmetric, fully suspended flow rate.

- Curve C corresponds to an actual bed of solids deposited on the pipe's bottom. . This bed is loosely packed at a somewhat lower concentration than the maximum for solids packing with water in the voids. The fluid drag on its upper surface exceeds the frictional drag of the pipe wall on its lower surface. Thus, the fluid and "slides" along the pipe bottom drag along the bed. This phenomenon is called "sliding bed" flow. It is particularly detrimental to the integrity of the pipe wall in terms of abrasion.
- Curve D corresponds to the situation in which the frictional drag of the fluid on the top of the bed is less than that of the pipe wall on the bottom of the bed. Here the velocity of the lower portions of the bed falls to zero and the majority of the deposited solids become stationary. This is referred to as the "stationary bed" flow rate. Some portion of the upper layers of the bed will be partially re-suspended by the drag and lift forces generated by the rapidly flowing fluid. These re-suspended particles will be carried along for a short distance and redeposited further down the pipe. This process is repeated so that the surface particles in the bed undergo a periodic hopping or skipping motion down the surface of the bed. This type of motion of the particles is called the "saltation" flow rate. It gives rise to ripples and dunes on the surface of the bed.
- As the velocity is decreased too far, even saltation cannot occur and all the solids entering the pipe deposit and none are removed, with the result that the ultimate in hold-up occurs and the pipe plugs. Curve E corresponds to this most unhappy case.

C.3. Final wear pattern of the bend pipe:

A deeper view of each wear zone is given, explaining the expected trends of the wear at four angles along the bend. Thus, the limits of the four quadrants are explained along the bend pipe, at $\phi = 0^o, 90^o/270^o, 180^o$ and $\theta = 0^o - 90^o$.

C.3.1. Outer wall:

Blue zone:
$\theta = 0^o$: minimum erosion at $\phi = 0^o$ and maximum at $\phi = 90^o$ and $\phi = 270^o$, small difference between max and min.
$\theta = 20^o$: max at $\phi = 0^o$, min at $\phi = 90^o, 270^o$, expect small difference between max and min.
Positive wear slope at $\phi = 0^o$, from $\theta = 0^o$ to $\theta = 20^o$.

Orange zone:
Everywhere maximum at $\theta = 0^o$.
Wear expansion towards the sides is more at $\theta = 55^o$ compared to $\theta = 20^o$.
Higher than Blue zone wear slope at $= 0^o$, from $\theta = 20^o$ to $\theta = 55^o$.

Red zone:
Maximum at $\phi = 0^o$.
Wider wear area than the Orange zone
Higher than Orange zone wear slope at $\phi = 0^o$, from $\theta = 55^o$ to $\theta = 70^o$.

Maximum zone:
Maximum at $\phi = 0^o$.
Wider wear area than the Red zone, assume to reach $\phi = \pm 90^o$.
Maximum and Red zones can be considered as one zone, which continues the trends of the Red zone in the Maximum zone, where the peak of the increasing wear is expected to be found.

C.3.2. Inner wall:

Orange zone:
At $\theta = 0^o$: maximum erosion at $\phi = 180^o$ and minimum at $\phi = 90^o, 270^o$, a large difference between max and min.
$\theta = 30^o$: max at $\phi = 180^o$, min at $\phi = 90^o, 270^o$, expect medium difference between max and min.
Negative wear slope at $\phi = 180^o$, from $\theta = 0^o$ to $\theta = 30^o$.
Negative wear slope at $\phi = 90^o, 270^o$, from $\theta = 0^o$ to $\theta = 30^o$.

Green zone:
$\theta = 30^o$: maximum erosion at $\phi = 180^o$ and minimum at $\phi = 90^o, 270^o$, medium difference between max and min.
$\theta = 85^o$: max at $\phi = 90^o, 270^o$, min at $\phi = 180^o$, expect medium difference between max and min.
Negative wear slope at $\phi = 180^o$, from $\theta = 30^o$ to $\theta = 85^o$.
Positive wear slope at $\phi = 90^o, 270^o$, from $\theta = 30^o$ to $\theta = 85^o$.

Orange zone:
$\theta = 85^o$: max at $\phi = 90^o, 270^o$, min at $\phi = 180^o$, expect medium difference between max and min.
$\theta = 90^o$s: max at $\phi = 90^o, 270^o$, min at $\phi = 180^o$, expect medium difference between max and min.
Positive wear slope at $\phi = 90^o, 270^o$ and $\phi = 180^o$, from $\theta = 85^o$ to $\theta = 90^o$.

C.4. Pipe geometry as wear parameter

Over the past decades, several methods have been proposed to reduce elbow wear. To achieve that, a number of studies have been conducted on changing the shape of pipe walls and adding extra members to reduce wear. Specifically, this section gathers some of those research cases, despite the fact that most of them utilized gas as a carrier fluid and their pipe sizes are much smaller than this project studies.

A method investigated by Duarte et al. [118] for mitigating erosion, was to add a vortex chamber to the arches of a regular bend pipe, see figure C.3a. The downstream of the chamber entrance, positioned at about the centre of the bend outer wall, deflects the flow and consequently results in erosion mitigation. Also, the results showed that as the concentration of particles increases the inter-particle collisions in the chamber are potentialized, resulting in a more efficient cushioning effect. Similar results were found by Ribeiro Duarte et al. [6], who compared a plugged tee (figure C.3b) and a vortex-chamber elbow with a standard elbow, highlighting that the cushioning effect became more active with increased mass loading. Furthermore, San et al. [128] studied various chamber sizes, concluding that the minimum erosion rate was when the chamber diameter was 1.5 times the pipe diameter and they expected that the larger chambers can lead to even lower rates. At the same time, Hassan et al. [129] investigated plugged tees and standard elbows and observed that by increasing the diameter of the pipe, the erosion rate in the plugged tees exceeds the rate of the standard elbows.

An innovative pipe wall design has recently been proposed by Duarte and Souza [7] to minimise bend erosion by twisting the straight upstream pipe wall and keeping a regular bend pipe, see figure C.3c. It is also possible to create a swirling flow in such a configuration, preventing particles from concentrating on specific areas at the bend wall. Based on the same principle, Santos et al. [8] found that inserting twisted tape upstream of a bend pipe reduced direct collisions against the bend wall due to the swirl imparted by the tape, see figure C.3d.

Apart from those approaches, some studies propose more simple shape modifications that seem to have a positive influence on wear reduction. Bahmani and Nazif [9] considered oval-shaped pipes with different pipe aspect ratios and found that the particle trajectories are greatly affected by the cross-section geometry, see figure C.3e. Consequently, this case has a direct impact on the erosion rate of the elbow channel and finally the results showed that the minimum wear was in the pipe with a 0.4 aspect ratio, meaning that the distance between the outer and the inner wall is 40% of the distance between the elbow sides. Additionally, Zolfagharnasab et al. [53] concluded that square ducts have the potential to lead to wear reduction compared to regular pipes and they also show less wear dependency on particle size and velocity.

However, as an alternative to the structure modifications, research has also been carried out for adding ribs on the elbow wall as an anti-wear technique. Song et al. [130] carried out experimental and computational analyses to demonstrate the effects of setting ribs on the walls of squared-shaped bends. Yao et al. [131] observed experimentally that a rib-settled wall could enhance the ability to protect a bend from erosion, as shown in figure C.3f. Fan et al. [10, 132, 133] investigated the influence that the shape of the rib has on particle erosion, in square-section elbows. Among the three shapes considered, it was found that the isosceles triangle ribs reduced wall erosion the most. Li et al. [88] used only one bump on the outer wall of a squared-shaped bend and through their numerical simulation and experimental research, they found that the installation of a bump in the area with the maximum wear can effectively reduce the erosion rate. The same technique was adopted by Zhu and Li [11], who installed a trapezoidal rib in a regular bend pipe and the numerical study conducted proved the erosion reduction, see figure C.3g.

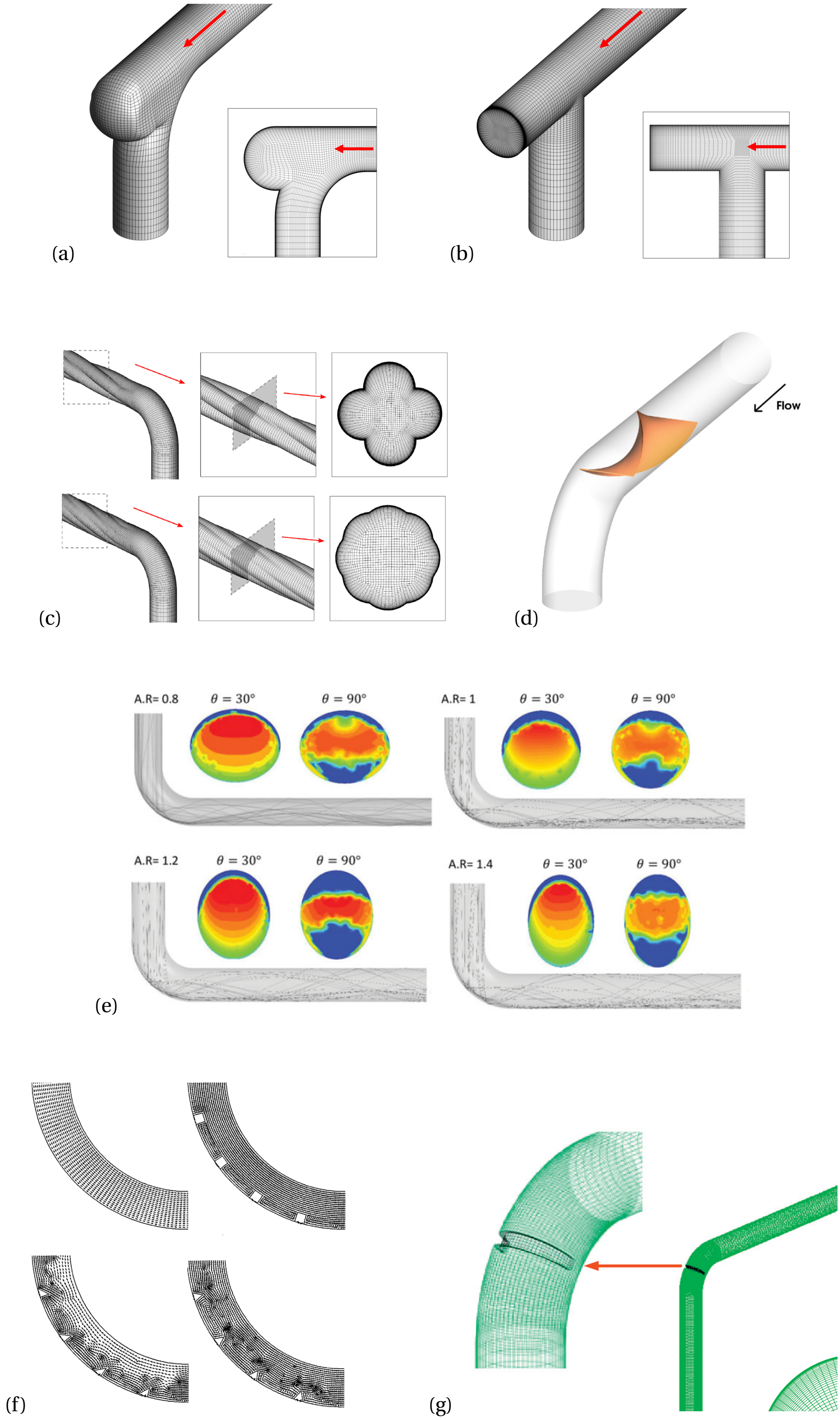


Figure C.3: Elbow geometries: (a) Vortex chamber, [6]; (b) Plugged tee, [6]; (c) Twisted wall , [7]; (d) Twisted tape, [8]; (e) Oval-shaped bend, [9]; (f) Squared-shaped bend: Multiple ribs, [10] ; (g) Bend pipe: Single rib, [11]

D

Conceptual design

D.1. Comparison of bow coupling designs

The main advantage of the concepts with one discharge pipeline was supposed to be the sustainability of replacing only one pipeline instead of two like the classic design. However, the usage of only one pipeline for both discharging ways needs to be replaced in a shorter time period. That means that the same amount of material is going to be replaced at the end, but the only difference is that the single pipeline will be replaced more times, increasing the overall replacement cost. To take it one step further, the concept with the nozzle at the female part causes the same issue not only to the straight pipeline but also to the bend pipe and the female part. Thus the last two significantly expensive parts are used always, regardless of the discharging way. Another important disadvantage of that concept is the complex, time-consuming and indeed costly replacement procedure. In regards to the reconnectable concepts, a specific drawback is the pipe wall thickness difference between the tower pipeline and the bend pipe and nozzle, respectively. The continuing use of the straight pipe is expected to reduce its wall thickness more than its subsequent pipes resulting in a non-smooth transition to the next pipe, and consequently erosion rate at the inlet of the next pipe.

All in all, the concept with the nozzle at the female part comes with a lot of disadvantages and therefore it is not considered in the conceptual design. That leaves two options for Function 1.1, the fixed bend pipe with a fixed nozzle and the reconnectable bend pipe. The former is already widely used for 1 m pipe diameter pipes, it does not have moving parts, and finally, it has faster and non-human involved switching, but costly valves. The latter has additional issues with the mechanism's cost and maintenance, moving parts and humans involved in switching.

In any case, the location and operation of the nozzles are not affected by the conceptual design. To be more specific, for both concepts the focus is only on the bend pipe and only the modification of that is allowed. In general, the first sub-suction is divided into the fixed bend pipe and the reconnectable bend pipe and these options have a place in the morphological chart.

D.2. Comparison of flow function solutions

D.2.1. Flow redirection

Starting with redirection of the flow, the two options are the regular cast bend pipe and the sections bend pipe. Firstly, The sections' bend pipe affects the flow and specifically worsens the erosion rate in the pipe. Also, with the sections bend the maximum wear in the pipe cannot be predicted as the outcome of the literature considered only regular bend pipes. However, it is a cheaper solution than the cast one, without undermining that there is a great cost difference between the two options since the fabrication cost, including working hours and means, of the sections pipe is quite high as well. Large thick plates should be cut precisely, then curved and finally weld all pieces to form the bend pipe. One last point is that the cast bend pipe can be made of harder material for better wear performance, something which is difficult for a sections' bend. Brittle materials would have a high risk of failure under all those fabrication steps, and therefore special treatments had to be adopted to achieve the desired shape.

For all these reasons, the company has already started reducing the utilization of such bends for the flow function. Overall, the cast bend pipe constitutes a better option for accuracy, flow conditions and expected eroded area in the pipe.

D.2.2. Pipe wear

Moving to the solutions about the wear in the pipe, there are three options, the increased thickness, the bend flip and finally the chocky bars. To begin with, the last solution is a straightforward method for extending the lifetime of the bend, as the bars are worn down before reaching the pipe wall surface. Chocky bars are expensive material which also needs a lot of working hours to weld them in the pipe. Also, the second half of the choky bars is not hard enough to be exposed in so erosive environment. Thus the erosion rate will increase when the hard part of the chocky bars is worn out. Then there are two options, either the bend pipe material is capable of erosion and the operation continues until reaching the minimum thickness of that pipe wall, or the bend pipe material is not capable of wear and the operation should stop. Then the next step is the machining of the inner side of the bend pipe to remove the partially-done chocky bars to make the surface smooth again to weld the new ones. Both scenarios lead to expensive replacements, and at the same time, the first option deprives the bend of being recycled as there are chocky bars welded at the non-worn areas. With regard to the other two solutions for the wear in the pipe, both of them are cheaper solutions compared to chocky bars and they can always be recycled. Additionally, they can have relatively hard material as one of the requirements for the support function is to minimise the loads on the bend pipe. Both of them aim for the least amount of replaceable material, taking advantage of the literature outcome, that the inlet and the outlet of the outer wall are the two extreme wear locations, minimum and maximum respectively. On the one side, the first solution increases the thickness of the outer wall along the bend pipe while the second option flips a regular bend pipe after a certain minimum thickness, so the inlet side goes to the outlet and vice versa. If one takes a closer look, the second option uses the first option twice with the same pipe. This can be distinguished by picturing the shape of the bend pipe at the flipping step, a bend pipe with increasing thickness from the inlet towards the outlet. The only difference is the fact that the outer diameter is constant now, while for the first option the inner diameter was uniform. That proves that the flipping bend pipe can provide a longer service life than the increased thickness concept. However the latter option requires less amount of material than the former one and consequently, the production cost is lower, even though the increased thickness can slightly increase the casting cost. As both solutions show comparable benefits the answer to the dilemma is given by the casting process of the bend.

One crucial point is that the minimum thickness for a cast piece is 25 mm, which means that the increased thickness option should have at least 25 mm thickness for the inner wall and 25 mm at the inlet of the outer wall. At the same time, one of the main parameters for the concept selection, see 4.5.1, is to minimize the loading on the flow function component to reduce the minimum required thickness, which indicates the wear in the pipe as the main factor for the wall thickness. Hence, considering that the outlet erosion is about five times as the inlet at the outer wall, the maximum thickness had to be much more than 35 mm, to have well-distributed wall thickness reduction at least for the outer wall. A design like that is practically infeasible because of firstly the enormous resulted mass and secondly the thickness of the next pipe which should increase accordingly. As the maximum wear is expected close to the outlet, the next pipe should also consider the same allowable thickness reduction. The next part is the female connector which currently weighs more than 5 t with an allowable thickness reduction of 20 mm, and therefore such extremely increased thicknesses are not considered. Thus, the increased thickness option should either increase dramatically this size and the overall mass of the already heavy parts or remain with a low inlet-outlet thickness difference and has no well-distributed thickness reduction. None of those is more beneficial than the flipping concept, which constitutes the most cost-efficient and sustainable solution with a longer lifespan and less replaced material.

It achieves more uniform wall thickness reduction which corresponds to the optimum solution compared to the other two. Better use of the inner wall because it applies both sides after the straight pipe where the bottom side has medium erosion. The fraction of the final bend mass over the original mass determines the efficiency of the wear management of the pipe. The lower this fraction is the more uniform the distribution is and the longer the service lifetime is, for a certain original amount of material. Ideally, the wall thickness arrangement of the bend pipe would be proportional to the wear rate of every point. This is a scenario with no loads exerted on the bend, or at least they are uniformly distributed because the loads on specific areas might require different wall thickness.

D.3. Inspection plugs: geometry and locations

For the thickness inspection of the pipes, ultrasonic measurement devices are used. The probe is placed perpendicular to the pipe surface and the thickness is presented on the screen of the device which is connected to the probe by a long cable.

In reference to the inspection holes, firstly the shape of the hole on the pipe should be defined. The high pressure that the plug should withstand in case of liner failure and the better distribution of the loads around the plug led to a circle shape. The concept can be described as large plugs that can be screwed and cover the hole. The diameter of the inspection plugs depends on the probe size and the required distance between measurements for every spot. An average ultrasonic measurement device has a probe of about 15 mm in diameter. In relation to the distance between measurements, the two indicators are the presence of cracks in the wall and the ripple wavelength on the worn wall. As stated in Chapter 2, the impact of the particles on the wall starts forming a wavy surface, whose amplitude and wavelength primarily depend on the slurry properties, wall material, impact angle and duration, at least for the initial stage [134]. In any case, the wavelength can be some microns for brittle materials, while ductile can even enter the millimetres' scale, after a long operational time, [46, 135].

Thus, considering also the space needed to access the bend pipe properly, it was decided that the diameter of the plug should be 45 mm. That fits three probes in line, which means the inspector can check the centre of the hole but also around it. The option of several measurements for the same location leads to a valid representative thickness, decreasing also dramatically the error probability.

The position of the inspection plugs is made aiming for the minimum number of them which can give adequate information for a safe operation, without unexpected liner failure. For that reason, the outcome and the assumptions from Chapter 3 are used to determine the most critical points to inspect before and after flipping the bend pipe. More inspection plugs increase the cost and weaken the strength of the support, especially at points close to the inlet, where the bend pipe is supported. Also, it was decided that the support pieces will remain connected during the flipping process, disconnecting and switching only the inlet and outlet flanges. The rotation of the support-liner system, without disconnecting the support pieces saves a lot of time, considering the connection/disconnection process and the alignment of the bend pipe. Firstly, the critical points before and after flipping are to be stated and then their actual location on the support pieces.

- Thickness reduction before flipping:
  At first, the maximum wear of a new bend pipe is expected to be in the last 20^o of the outer wall. The importance of that area requires a close look as it determines the flipping timing and probably the replacement. For that reason, three locations were decided as appropriate, the two extremes at 70 and 90 and lastly one more in the middle, at 80^o. However, as stated at the beginning of the section, the inspection of the bend pipe outlet can be done from the inside, having access from the female part. As a consequence, only two inspection plugs are necessary, at 70^o and 80^o.

  Thus only two plugs are necessary, a case which leaves a range of 10^o for every hole, which corresponds to 5^o maximum distance of unknown thickness, along the outer.

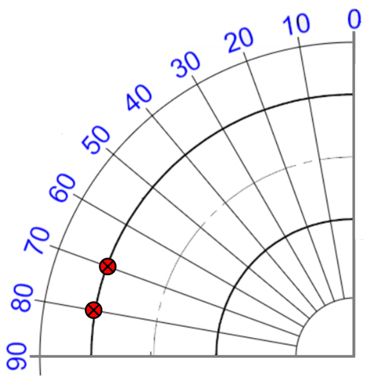


Figure D.1: Inspection plugs: Locations before flipping.

- Thickness reduction after flipping:
  After rotation the outlet is still of major importance, necessitating the two inspection holes as discussed in the previous paragraph. Also, it should also be kept in mind that the conclusion of Chapter 3 indicates that the wear rates increase more rapidly towards the outlet, for the outer wall. In that case, the summation of thickness reduction after flipping will be concentrated at both ends of the outer wall. Even though the bend outlet is assumed to have about five times the wear rate of the inlet, the new inlet, which initially was the outlet, apparently requires a close look. However, these three inspection holes are needed because there is no access from the previous pipe. Thus, the after-flipping arrangement should have five plugs so far, two at the outlet and three at the inlet, 0^o, 10^o and 20^o. Moreover, the scenario of the worn previous pipe should be taken into account, which means that the new outlet can have less thickness than it is expected, at least for the inner wall and maybe the sides. Then, the inspection from the female part gives access and therefore no more plugs are needed in that area.

  Nevertheless, if the bend pipe is flipped without the replacement of the previous pipe, the inspection of the inner wall is necessary, especially if the liner of the previous pipe is more efficient with wear. One could argue that the bend inlet cannot reach less thickness than the previous pipe, but this is true only when they have the same material. Otherwise, the bend pipe inlet will initially lose the thickness difference relatively fast, and then the wear will continue at a higher pace than the previous pipe. this example concerns only the inner wall of the inlet, where the wear rate can be high. Lastly, despite the highest thickness reduction at the two extremes of the outer wall, two more inspection plugs are added between 20^o and 70^o for extra safety and a better picture of the bend pipe thickness status.

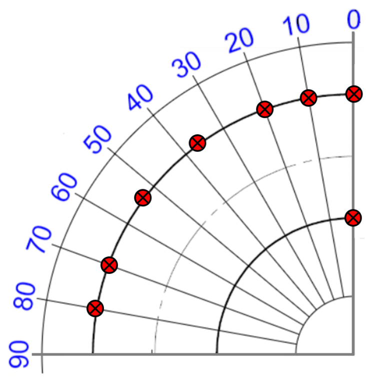


Figure D.2: Inspection plugs: Locations after flipping.

Overall, after flipping the critical points are much more than the initial orientation. A total of eight inspection plugs was decided, with only one of them being at the inner wall, and precisely at 0^o after flipping. The outer wall requires at 70^o and 80^o before flipping and after that at $0^o, 10^o, 20^o, 36.67^o, 53.33^o, 70^o, 80^o$. As can be understood, the two locations before flipping (70^o and 80^o) are already considered in the after-flipping locations (10^o and 20^o). Thus, D.2 shows the final arrangement of the inspection plugs. The only non-symmetrical spots are at the inlet after flipping. Therefore, in order to avoid disconnecting the support pieces during rotation, the side with the two extra holes will be bolted to the female part first.

E

Concepts' rating

E.1. Scoring reasoning

E.1.1. Cost of replaceable parts: the amount of replaceable material and required connection/disconnection fabrication

1: Casted bend pipe, which works as a liner. It is the least amount of material without any other fabrication than the machining of the edges.
2: Casted bend pipe with flanges, fabrication for bolt holes, and additional precise welded part for connection to the support structure.
3: Casted bend pipe with flanges and fabrication for bolt holes. The weight of the flange is small compared to the total mass but the fabrication of flanges takes a lot of time. It should be machined on both sides to be perfect for the bolt connections. Also, 28 holes of about 35 mm diameter on a 50 mm thickness pipe require more working time and therefore money.
4: Same as 2
5: Same as 1
6: Same as 3
7: Same as 3
8: Same as 1

E.1.2. Ease of replacement: refers mainly to the replacement procedure, required steps

1: Steps: disconnect the tower, lift it with the female part, disconnect the female part, and disconnect the two pieces. More complicated replacement compared to the current procedure because the additional step of disconnecting and connecting the two outer pieces takes place.
2: Steps: disconnect the tower, disconnect the support, lift it with the female part, disconnect the female part. The support should be stiff so no pretension is needed because this would make the replacement very challenging, like cables.
3: same as 2, additional step to disconnect the frame, like concept 1.
4: same as 2
5: disconnect the top piece, weld a lifting pad at COG and lift the bend pipe. The problem with this idea is that the new bend will be lifted from a pad eye, which should be removed and then machine the area, so the top piece can be connected. Also, the alignment of the liner should take place on board, a very challenging process, which requires time and space. Apart from that, only the half flange is connected, and there is a moment because of the female part's weight. For safety, another special part can be added between the platform and the female part in order to support it. If this is not possible, then the entire system of "bend pipe-support pieces-female part" should be removed together, and then disconnected at the shipyard.
6: disconnect the bend pipe from the tower and the female part and lift the bend pipe. The only problem that can arise with this concept is when the casted bend pipe does not fit perfectly on the female part and the tower pipe. As both connection points are fixed, any imperfection from the casting can cause significant issues, and then the flanges should be machined accordingly.
7: Disconnect the female part from the support, lift the bend pipe with the female part, disconnect the female

part. Otherwise, add an extra part to support the female part and remove only the bend pipe.
8: same as 3

E.1.3. Cost of structural/permanent parts

1: Two cast pieces with flanges at every side.
2: Support from the top side (steel wires or rods or plates structure).
3: Same as 2, with the addition of the two pieces frame.
4: Relatively large structure made of welded plates, not very complex manufacturing process.
5: Similar to 4, plus the two outer pieces (like concept 1). The combination reduces the strength requirement of both supports -> The structure is cheaper than concept 4 and the two pieces are cheaper than concept 1. However, the overall cost is expected to be higher.
6: Probably two relatively large structures are needed. High moment loads because of the distance between the platform and the female part.
7: Similar to 6. Also, the rotational connection is more expensive and requires more maintenance. Additionally, very high cost for the rotational mechanism (like hydraulic cylinders), and its maintenance. Extra cost for sea-fastening support.
8: Two casted pieces (like 1) and one rotational frame. Additional cost for the winch power and lifting cable because the bend is a much heavier structure than the floating pipeline that this winch normally lifts. Cost for sea-fastening support.

E.1.4. Loading on bend pipe/replaceable part

1: Can reach high strength and stiffness providing uniform support to the bend pipe. It has to carry its weight. Heavier structure than the current case.
2: Probably it has the highest loading on the bend pipe. Small support area on the top of the bend pipe, higher stress. It cannot be a very large connection on the bend because of the hoisting cables limitation. The force from the connecting part (cables/beam/rod) tends to rotate the pad anticlockwise. Also, the outer wall is under high compression between the pad and the tower, and under high tension between the pad and the female part. The entire inner wall is under compression, especially close to the inlet.
3: Much more uniform support of the bend pipe compared to 2. However, the connection to the tower and the female part is through the flanges of the bend pipe. That means the bend pipe receives a large amount of loading, most probably more than concept 1.
4: High compression and bending load at the connection point because of the bend pipe-female part weight. The internal pressure works in favour of the bend pipe, in that case, to avoid buckling there. Probably it has fewer loads than concept 2 because the support of concept 4 can be much larger, as there are no space limitations for the cables and the covered area has a low risk of wearing and therefore no frequent inspection is needed.
5: The high strength and stiffness of that concept minimizes significantly any load on the bend.
6: Another concept that provides high strength and stiffness to the bend pipe. This concept increases the stiffness in a more smart way by just supporting the female part, from which the highest loads are generated because the floating pipeline is connected there and it is far away from the tower where the bend pipe finds support.
7: Similar to 6. a drawback of that concept is the rotational degree of freedom which leaves bending loads for the bend pipe.
8: Similar to 1. However with lower strength and stiffness since the flanges are at the sides and do not contribute to the system.

E.1.5. Required support on existing structure

1: probably small support of the tower flange area because the new total weight is higher. Similar loading conditions as the current case. The mass of the inner bend pipe and the two outer shells is expected to be higher than the current design with one but thicker pipe.
2: Probably just additional support of the tower, at the point where the bend support is attached. 3: Similar to 2, higher load because of the extra frame weight.
4: It is close to the tower but probably the mass of the structure and the load that the structure support result

in additional support at the beginning of the platform.
5: Similar to 4, but with a higher load because of the two pieces' support bend.
6: Most probably the platform requires additional support, at least the area between the tower and the support point. Long distance from the tower, which is the closest part that can provide support.
7: Similar to 6. Additional support for the horizontal load that will be applied from the female part support when the bend pipe is at the sea-fastening position. Also, support of the tower at the location where the mechanism is attached.
8: Probably at the base of the rotational frame when the bend pipe is at the sea-fastening position. Support also for the tower because the cable from the diabolo roller is responsible to lift and rotate the bend pipe.

E.1.6. Accessibility – Inspection: ease of inspecting the wall thickness of the bend pipe

1: Impossible to inspect the inner pipe thickness from the outside. The only realistic alternative solutions for inspection are either having access from the bottom of the female part when the system is out of service but this option arises safety issues or removing the outer support pieces to inspect. To do that the bend pipe should be disconnected entirely. Otherwise, they can inspect from the inside, without disconnecting the support pieces, if they remove the bend pipe from the vessel and place it at the shipyard.
2: Easy inspection, not possible only for the small connection area.
3: Similar to 2, but with a lower available inspection area. However, this is not a problem since the frame bars are not expected to be very wide to cover a lot of bend pipe's surface. Adequate inspection can still take place considering also that the erosion in the bend pipe is going to be more uniformly distributed.
4: Similar to 2, just with probably a larger connection area, in a less wear risky part of the bend.
5: Same as 1.
6: The easiest inspection, no covered surface at all.
7: Same as 6.
8: Similar to 1. It has the advantage of visual inspection when the bend pipe is at the sea-fastening position.

E.1.7. Safety - Protection of bend pipe

1: One of the safest concepts. It prevents leakage, it has proper walking spaces, there are no moving parts, and the pipe is protected from hoisting chains.
2: no leakage safety, safe walking areas, no moving parts, no chains protection.
3: Similar to 2, but with chain protection.
4: No leakage safety. The walking space might not be enough, at the position of the structure on the platform. No moving parts. No chain protection.
5: Similar to 4, but with leakage prevention and chain protection.
6: It has chain protection and there are no moving parts. However, this is no safety for leakage and the walking spaces are very narrow.
7: Similar to 6. The extra disadvantage is the presence of moving parts.
8: Similar to 1. The extra disadvantage is the presence of moving parts.

E.1.8. Ease of applying on existing vessels

1: Probably the easiest to apply. Only the connection flanges need to be replaced because of the wider support diameter.
2: Relatively easy application. Only minor modifications of the connection point with the tower.
3: Same as 2.
4: Fabrication at the connection points. Probably more than concepts 2 and 3.
5: Same as 4.
6: A lot of fabrication on the female part and the platform for strength and walking space.
7: Similar to 6. Additional difficulties are the installation of the rotational mechanism and the sea-fastening position.
8: Significant consideration for the tower strength for the lifting of the bend pipe and the female part. Also, the winch power should be checked.

E.2. Scoring tables

		Concepts							
Variable Requirements	Weight	1	2	3	4	5	6	7	8
Cost of replaceable parts	5	5	2	3	2	5	3	3	5
Ease of replacement	4	3	3	2	3	3	5	4	2
Cost of structural/permanent parts	4	4	5	3	4	3	4	1	2
Loading on bend pipe/replacebale part	4	4	1	3	2	5	4	3	4
Bend pipe inspection, accessibility	3	1	4	4	4	1	5	5	1
Required support on existing structure	3	5	3	3	4	4	3	2	2
Safety - Protection of bend pipe	2	5	3	4	2	4	3	2	3
Ease of applying on existing vessels	1	5	4	3	4	3	2	1	2
Summation:		102	77	79	78	95	99	73	74

Figure E.1: Concepts' rating: Author.

1	2	3	4	5	6	7	8
5	3	4	3	5	4	4	5
4	3	3	3	5	5	4	3
4	5	3	5	4	3	2	2
3	3	3	4	4	5	5	3
2	4	4	4	2	5	5	2
5	4	4	4	4	3	3	4
5	3	3	3	5	3	2	2
5	4	4	4	4	2	2	2
105	93	90	97	109	104	94	81

(a) Mechanical Engineer, Dredging Equipment

1	2	3	4	5	6	7	8
5	3	2	2	5	3	3	5
3	5	1	3	2	4	4	2
3	5	3	4	2	4	3	1
5	2	3	2	5	4	3	5
1	4	4	4	1	5	5	2
5	4	4	3	3	2	2	4
5	3	4	3	4	3	2	4
5	4	3	2	2	1	1	2
102	97	73	75	83	91	81	85

(b) Product specialist, Dredging Equipment

1	2	3	4	5	6	7	8
5	4	4	4	5	4	4	5
3	4	2	2	3	4	5	3
3	5	3	5	4	5	5	3
5	3	5	3	5	4	4	5
1	5	5	5	1	5	5	1
5	4	4	3	3	3	3	3
5	2	4	3	5	3	3	4
5	4	3	3	3	2	2	2
102	103	98	93	98	104	108	91

(c) Product Manager, Product Management

1	2	3	4	5	6	7	8
5	4	3	2	5	2	2	5
4	3	4	3	4	5	5	4
5	4	3	3	3	3	3	3
2	3	4	5	5	4	4	4
3	4	4	4	3	5	5	3
5	4	4	3	3	3	3	3
4	3	5	3	3	3	3	4
5	4	4	3	3	3	3	3
106	94	97	84	100	91	91	98

(d) Engineer, Dredge Line Components

Figure E.2: Concepts' rating: Experts

F

Analysis

F.1. Vessel details

Table F.1: Vessel details

Symbol	Parameter	Value
L	Length, [m]	150
B	Breadth, [m]	25
D	Moulded depth, [m]	14
CB	Block coefficient	0.8
Kr	Roll radius or gyration, [m]	9.75
GM	Metacentric height, [m]	1.75
f_{BK}	Coefficient for no bilge keel	1.2
f_T	Draught ratio	1
R	vertical coordinate of the ship rotation centre, [m]	7
z	X, Y and Z coordinates of the considered point with respect to the coordinate system, [m]	10
y		0
x		80

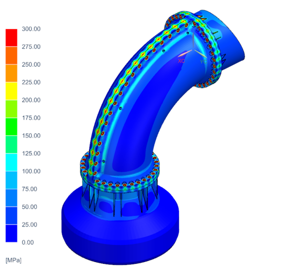
F.2. Load Combinations

Table F.2 shows the magnitude of every load as well as the axis on which they should be applied. It can also be seen that there is an additional column, named "Extra mass". This indicates the mass of the mixture, which shall be applied as an additional mass in the corresponding pipes, depending on their inner volume.

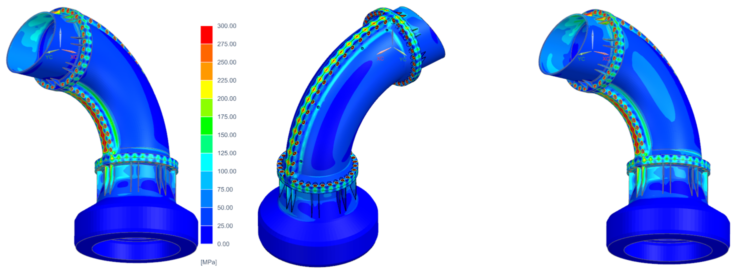
Table F.2: Load combinations with application values

LC	Model	SF	Gravity [m/s²]			Extra mass	Pressure (bar)	F_{Flow} [51.04 kN]		Wind Force [kN] +yy'		F_{hose} [kN]			Vessel accelerations [m/s²]			F_{winch} [kN]
			x	y	z			x	z	Regular	Extreme	x	y	z	x	y	z	z
1	1	1.5	0	0	-9,81	No	25	+1	+1	0	0	0	0	0	0	0	0	0
2	2	1.5	+0.342	0	-9.804	Yes	25	+1	+1	0	0	+258.82	0	- 3256.15	0	0	0	0
3	2	1.5	0	+0.855	-9.773	Yes	25	+1	+1	0	0	+183.01	+183.01	- 3256.15	0	0	0	0
4	2	1.5	0	+0.855	-9.773	Yes	25	+1	+1	0	0	0	+258.82	- 3256.15	0	0	0	0
5	2	1.5	-0.342	0	-9.804	Yes	25	+1	+1	0	0	-224.14	+129.41	- 3256.15	0	0	0	0
6	2	1.33	0	0	-9,81	Yes	25	+1	+1	+1.73	0	+258.82	0	- 3256.15	+0.4	0	-1.477	0
7	2	1.33	0	0	-9,81	Yes	25	+1	+1	+1.73	0	+183.01	+183.01	- 3256.15	0	+1.827	-1.376	0
8	2	1.33	0	0	-9,81	Yes	25	+1	+1	+1.73	0	0	+258.82	- 3256.15	0	+1.827	-1.376	0
9	2	1.33	0	0	-9,81	Yes	25	+1	+1	+1.73	0	-224.14	+129.41	- 3256.15	-0.4	0	-1.477	0
10	2	1.1	0	0	-9,81	No	0	0	0	0	+5.76	0	0	0	0	+6.088	-4.586	0
11	2	1.1	0	0	-9,81	No	0	0	0	0	0	0	0	0	0	0	0	+ 351.9

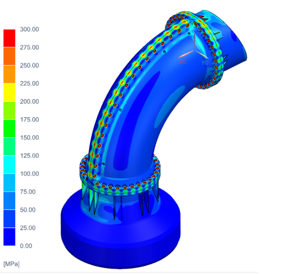
F.3. Simulation results



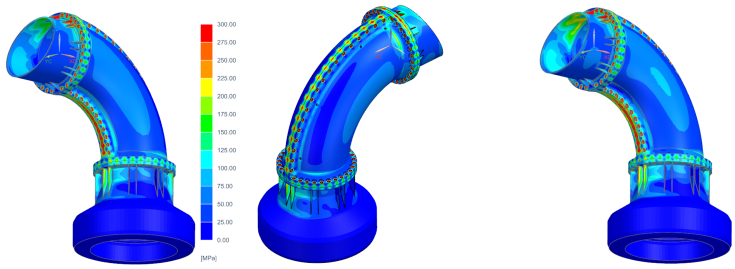
(a) LC2



(b) LC3

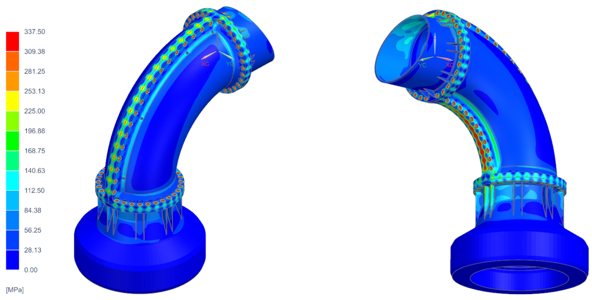


(c) LC4

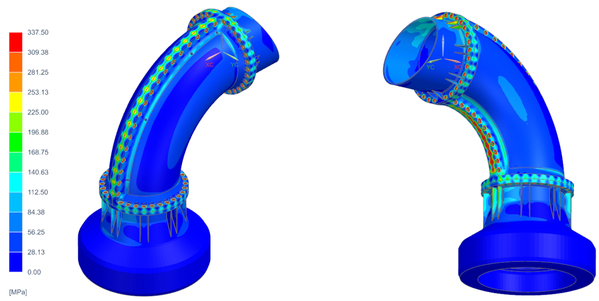


(d) LC4

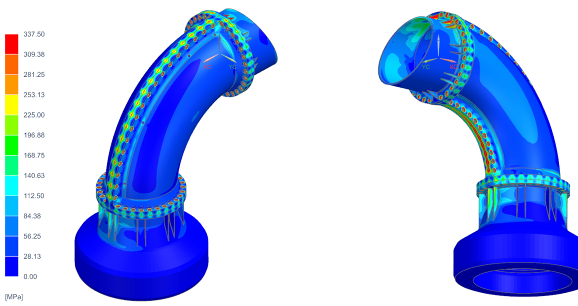
Figure F.1: Simulation results: Model 2: Case I



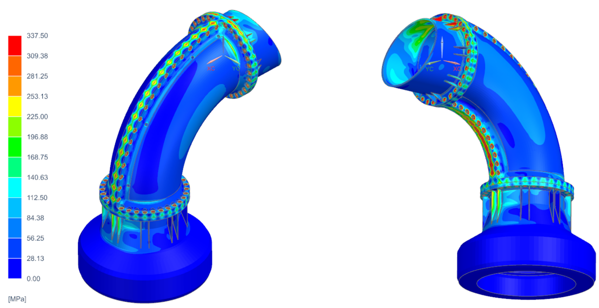
(a) LC6



(b) LC7

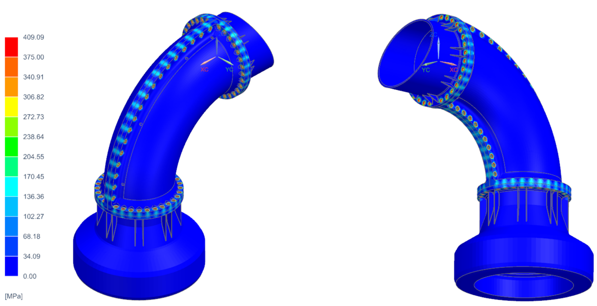


(c) LC8

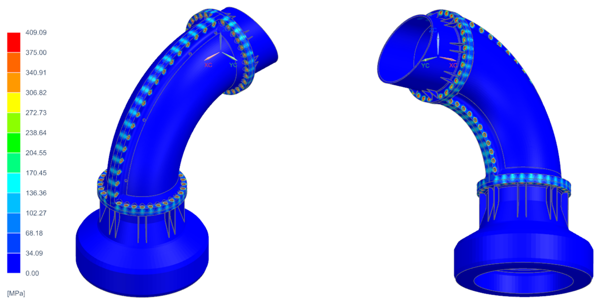


(d) LC9

Figure F.2: Simulation results: Model 2: Case II



(a) LC10



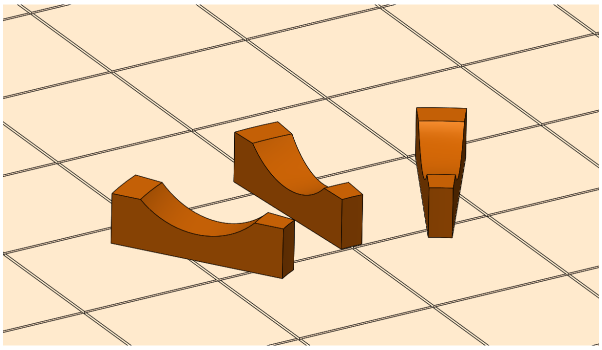
(b) LC11

Figure F.3: Simulation results: Model 2: Case III

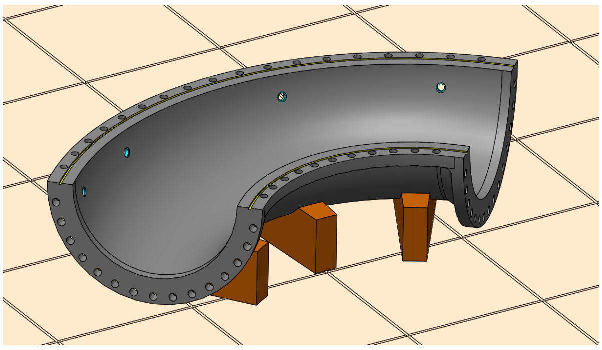
F.4. Liner connection procedure

This section describes the procedure that has to be followed for the installation of the liner in the support structure. First and foremost, the most important and simultaneously challenging task, during assembling the parts, is the alignment of the liner. The inner surface of the liner should be perfectly aligned with the liners of the connected pipes. To achieve that, the bolt holes are used as a reference point because the flanges are machined to fit with the corresponding pipes. Additionally, a gap of 15 mm between the outer surface of the liner and the inner surface of the support was decided as appropriate to deal with the tolerance of the three large cast models.

- **Step 1:**
  - A large wooden base is created according to the outer diameter of the support piece.
  - The support piece with the o-ring grooves is lifted and placed on the base, with the inner wall facing upwards.



(a) Wooden base



(b) Bottom part

Figure F.4: Connection step 1

- **Step 2:**
  - At least two small plates are placed at the two ends of the pipe, on the inner wall. Wedge-shaped plates, which will be used for the alignment at a later stage.
  - Also, a larger plate shall be welded in the centre of the pipe, to maintain the gap between the support piece and the liner. It is important to use the minimum number of plates so the inlet and outlet of the liner will be able to be adjusted.

    *It should be clarified that the number of plates, shape, size and locations depend on each application individually.

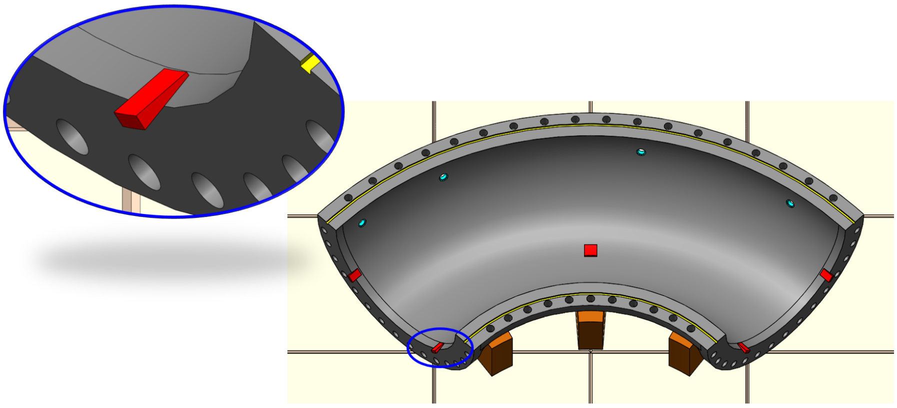


Figure F.5: Connection step 2: Alignment plates

- **Step 3:**
  - The liner is lifted and placed on the support piece, specifically on the five plates.

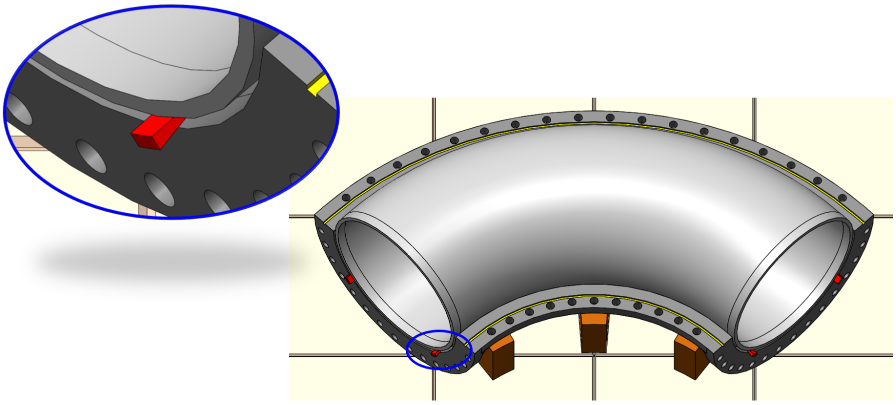


Figure F.6: Connection step 3: Liner

- **Step 4:**
  - The two o-rings are positioned in the corresponding grooves.
  - A large plate is welded at the centre of the top support piece.

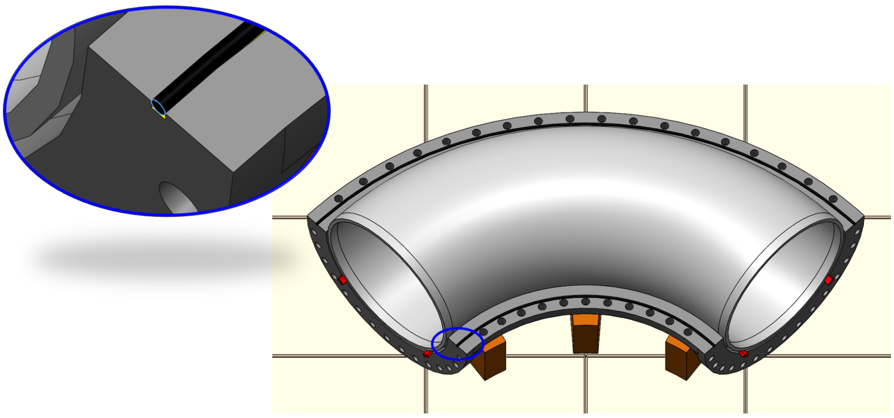


Figure F.7: Connection step 4: O-ring

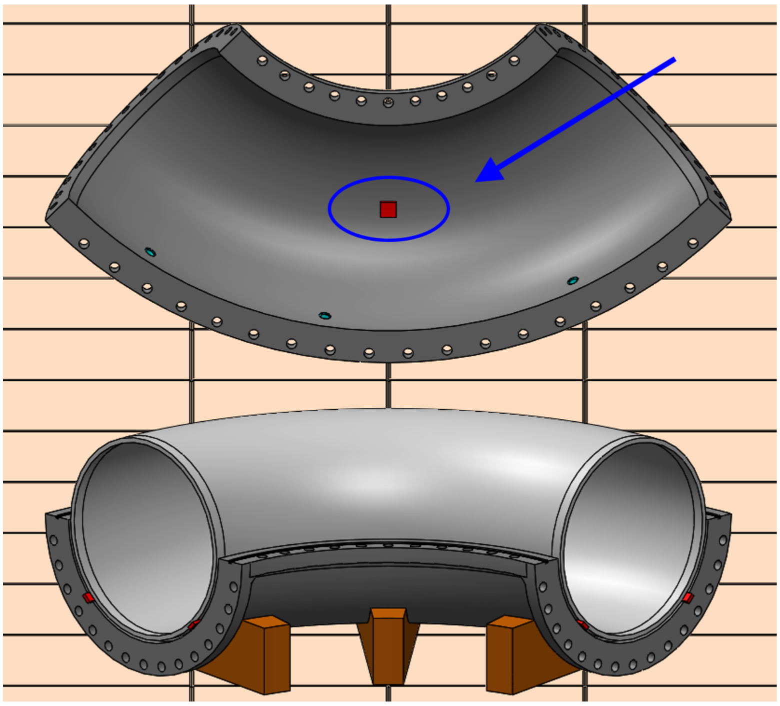
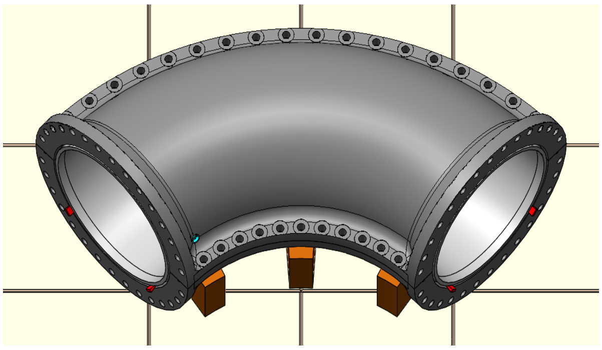
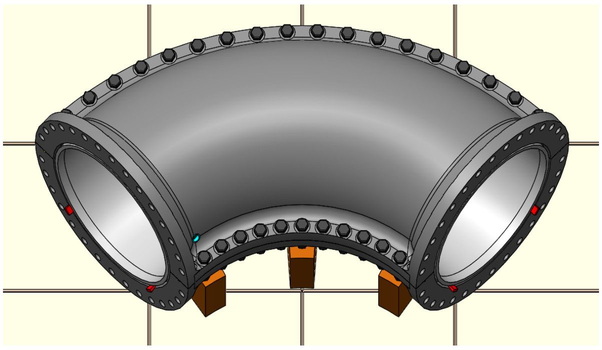


Figure F.8: Connection step 4: Plate on top part

- **Step 5:**
  - The top part is lifted and placed on top of the first part.
  - All bolts are screwed and the support has the final shape.



(a) Top part



(b) Bolts

Figure F.9: Connection step 5

- **Step 6:**
  - Another four wedge-shaped plates are used between the liner and the top support piece.
  - The eight small plates are carefully hammered to adjust the direction of the liner on both sides.
  - The inlet and the outer of the liner are aligned with respect to the circular flanges' bolt holes.

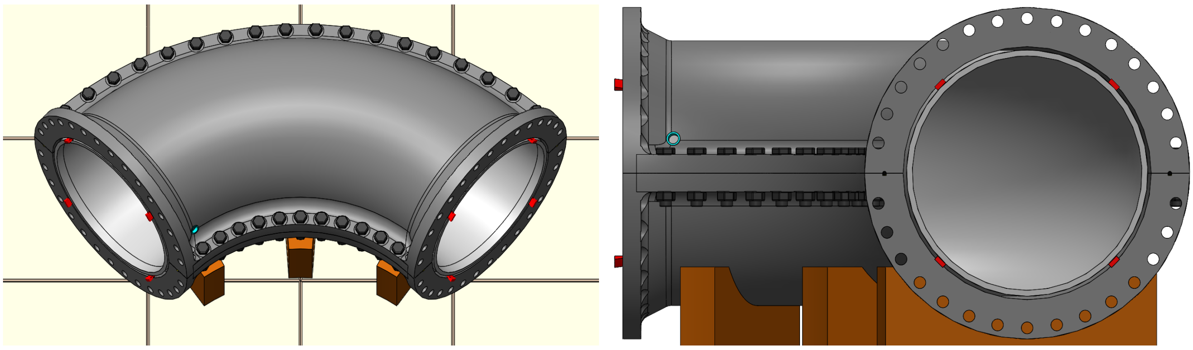


Figure F.10: Connection step 6

- **Step 7:**
  - When the alignment is completed, the wedged plates' part that protrudes from the flange surface is cut off.
  - Then, the plates are welded with the support pieces as well as the liner. Figure F.12 shows the welded points of a liner in a straight pipe, with a smaller diameter than the study pipe.
  - The last step is to grind all surfaces so that the flange can fit with the other pipes.

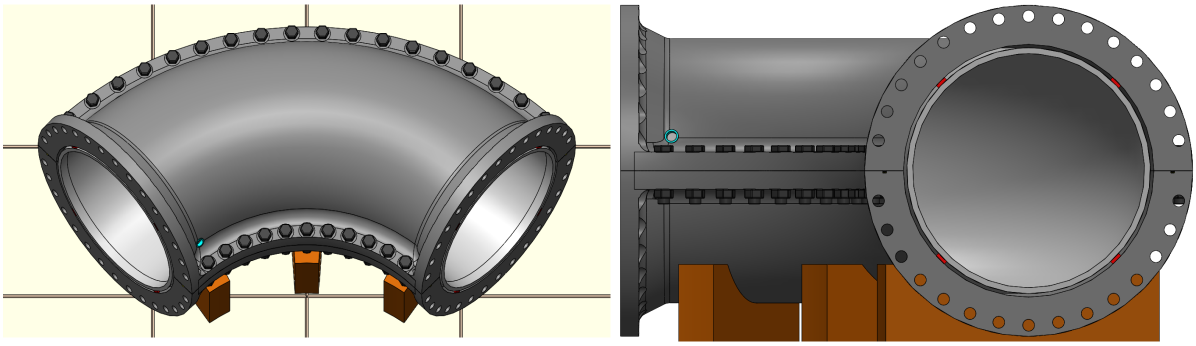


Figure F.11: Connection step 6

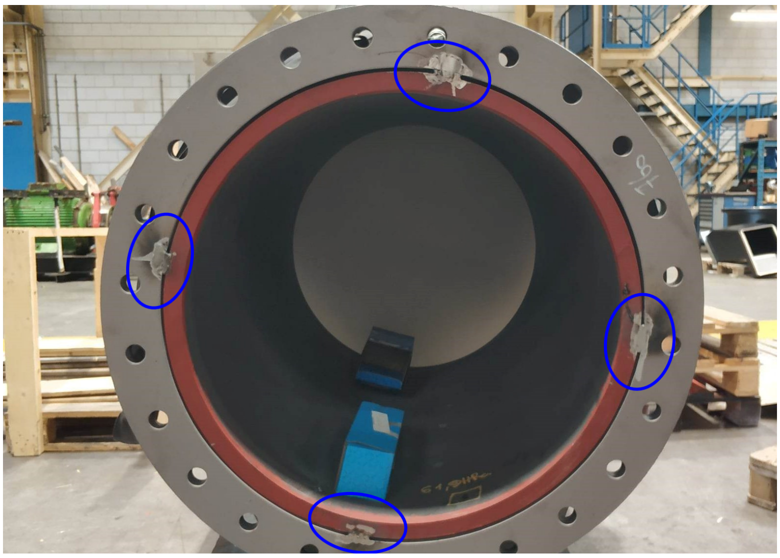
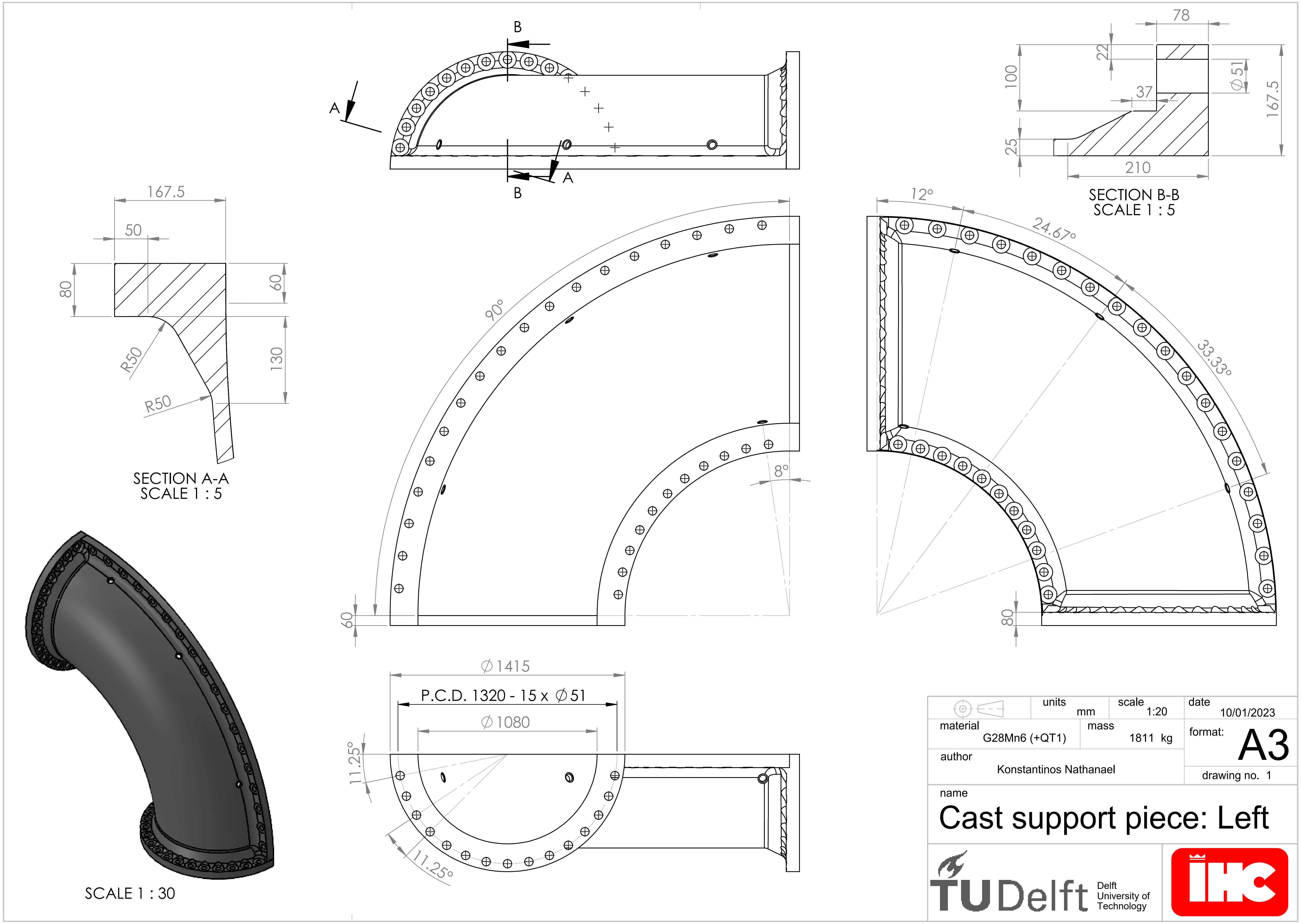
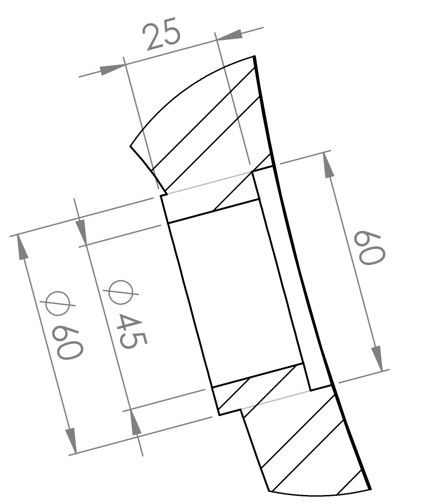


Figure F.12: Connection step 6: Welded points

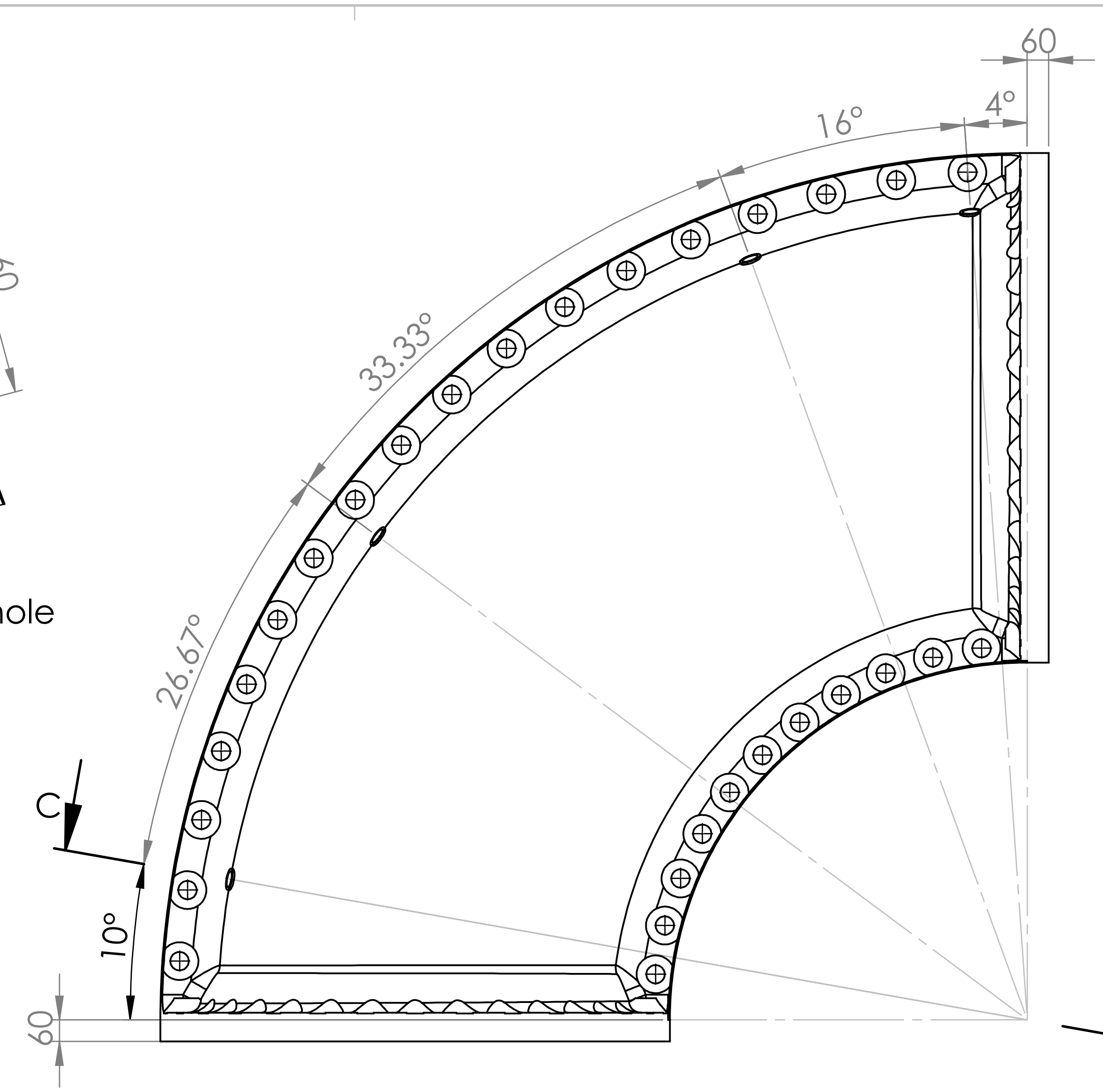
G

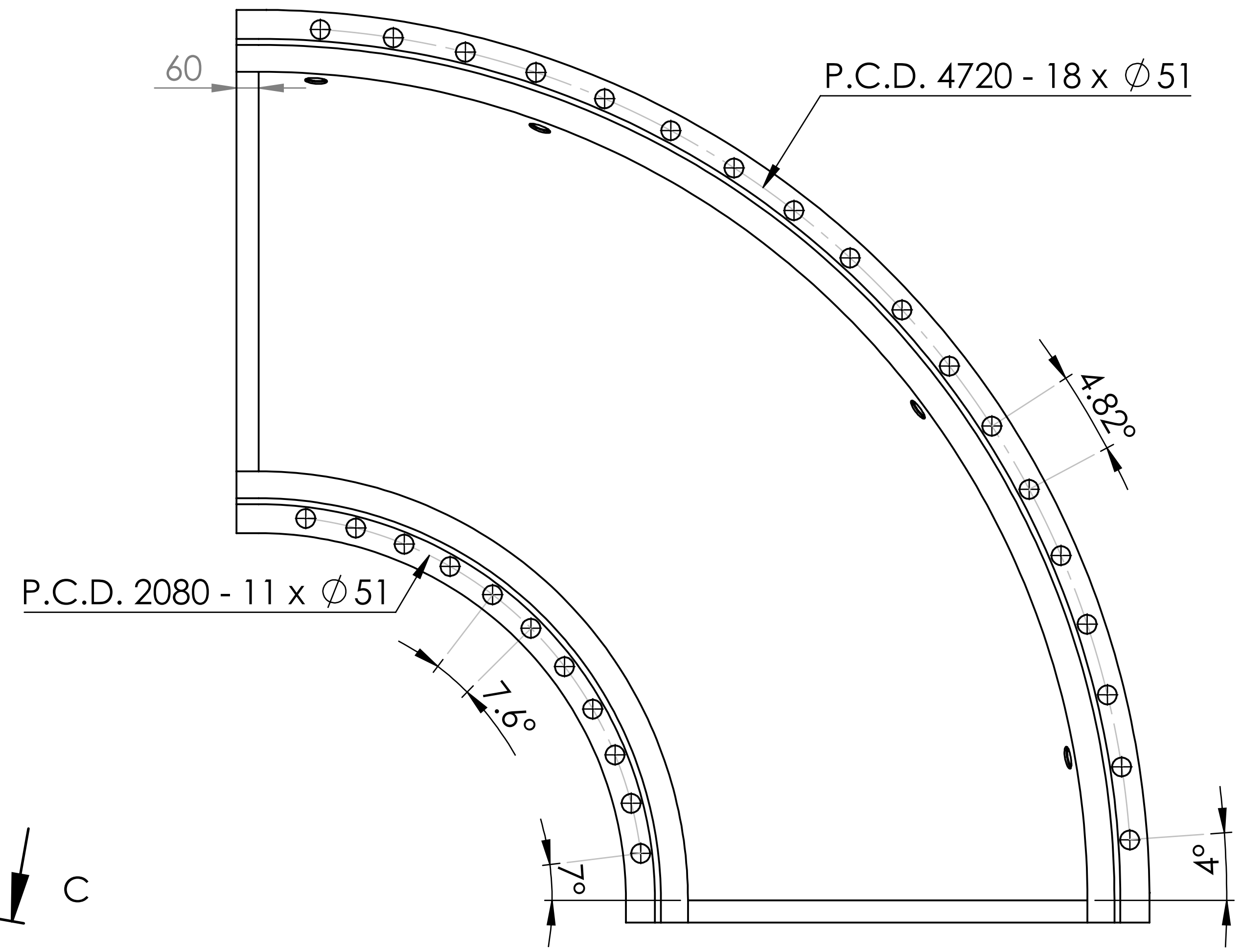
Mechanical drawings

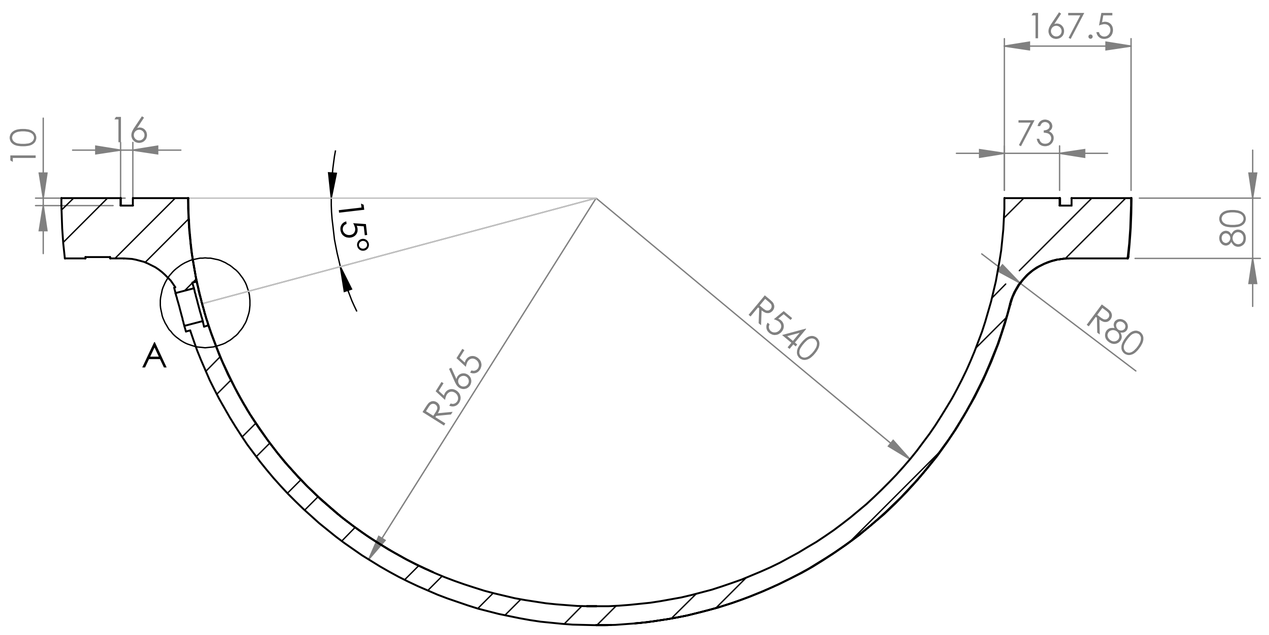

B
A
B
A
78
22
100
37
∅51
167.5
25
210
SECTION B-B
SCALE 1 : 5
167.5
50
80
60
R50
130
R50
SECTION A-A
SCALE 1 : 5
90°
8°
60
12°
24.67°
33.33°
80
∅1415
P.C.D. 1320 - 15 x ∅51
∅1080
11.25°
11.25°
SCALE 1 : 30
units mm
scale 1:20
date 10/01/2023
material G28Mn6 (+QT1)
mass 1811 kg
format: A3
author Konstantinos Nathanael
drawing no. 1
name
Cast support piece: Left
TU Delft
Delft University of Technology
IHC



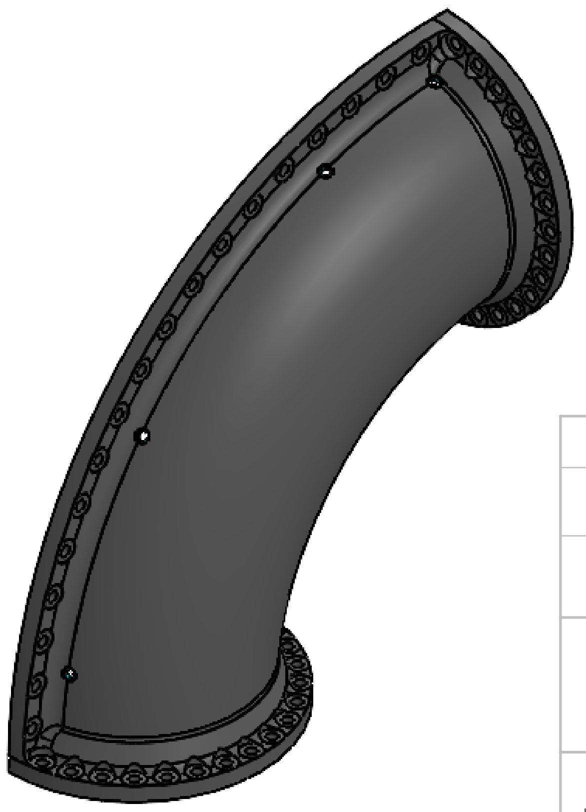
DETAIL A
Inspection plug hole
SCALE 1 : 2







SECTION C-C
SCALE 1 : 10



SCALE 1 : 30

	units	scale	date
	mm	1:20	10/01/2023
material: G28Mn6 (+QT1)	mass: 1811 kg		format: A3
author: Konstantinos Nathanael			drawing no. 2
name: Cast support piece: Right			

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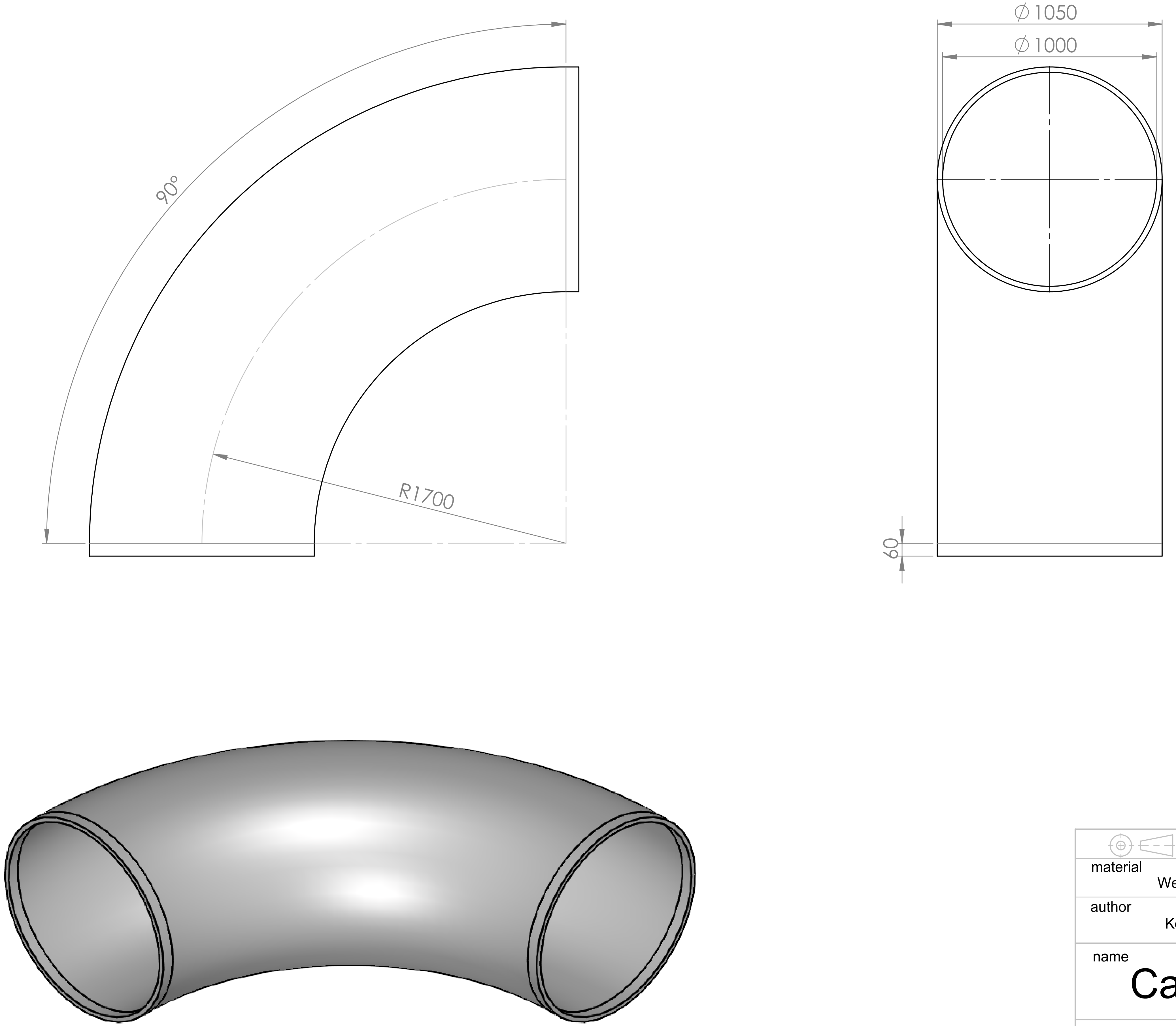

units mm	scale 1:20	date 10/01/2023
material Wearmet S2	mass 1752 kg	format: A3
author Konstantinos Nathanael		drawing no. 3
name **Cast bend pipe - Liner**		

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