Re-designing the Vibro-drill for post-installation retrieval

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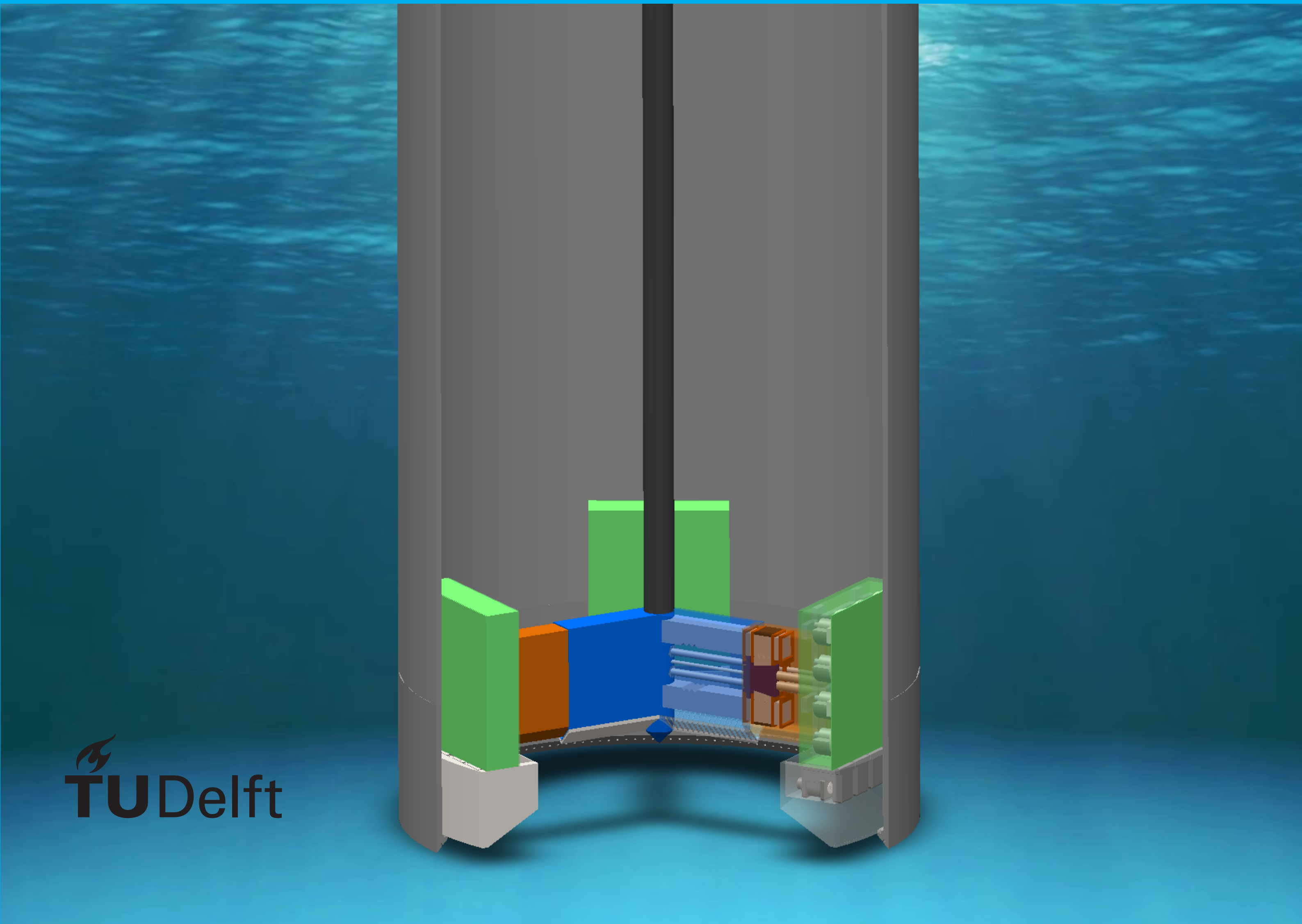

Re-designing the Vibro-drill for post-installation retrieval

Redesigning the Vibro-drill to include functionalities for retrieval. Report on the design process, structural analysis, the detailed design and recommendations to GBM Works.

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Project duration: May 1, 2019 – January 29, 2020
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Preface

This Master thesis is the final step of graduating for the Master Offshore and Dredging Technology at the Delft University of Technology. In collaboration with GBM Works, the topic of monopile installation has been transformed into a project which is of interest for all parties. The graduation committee consists of Ir. J. Hoving and Prof. Dr. A. Metrikine from the TU Delft and Ir. B. Arntz from GBM Works.

During this project the committee has provided me information, inspiration but most of all critical comments which forced me to improve my understanding of many subjects. Even though some of my meetings have been hindered or unprepared, the committee invested time and effort in guiding me towards a successful project. In special I would like to thank Jeroen Hoving for making time for additional meetings whenever I requested your advice or second opinion. These meetings have been very useful in separating essentials and side issues. I would also like to thank Ben Arntz for giving me the opportunity to graduate at GBM Works and all of the many brainstorm sessions of which some happened before the project had even started.

The rest of my colleagues at GBM Works have been a huge help to me as well. By providing me with comments on my presentations and report, but also by providing a great working atmosphere, they have helped me fly through the past nine months. For this I'm grateful to them.

Finally I want to thank my friends and family which I have seen slightly less frequent during the past half year. Whenever I needed a distraction you were there to help see some things are more important than this project.

Thomas Taks
The Hague, January 2020

Abstract

With increasing diameters of offshore monopile foundations, impact hammers require more energy per blow in order to drive them into the seabed. This does not only increase the fatigue loading on the monopile, but it also increases underwater noise that affects marine life. The increase in installation costs due to additional material on the monopile, noise mitigation and longer installation times drives contractors to look for alternative installation methods.
GBM's Vibro-drill strives to install monopile foundations through a combination of jetting, liquefaction and fluidization to reduce the soil resistance to such low levels that the monopile will penetrate the soil under its own weight and thereby significantly reducing underwater noise. Although the working principles are still being tested, the Vibro-drill design needs to be improved for commercial application. The design must include the three operational functionalities stated above, but to make the Vibro-drill re-usable for consequential monopile installations, additionally retrieval and structural rigidity are required.
For this thesis, the Vibro-drill is re-designed to include these functionalities. Starting from a process description, all load cases and additional design requirements were found. Based on a first global load case, a first selection was made for the connection mechanism. Subsequently, a Multi-Criteria Analysis was performed yielding two suitable concepts: the Vibro-Boxer and the Vibro-Polyp.

The Vibro-Boxer relies on three extending arms that connect with a specially designed pile shoe using twelve hydraulic grippers. To feed the system with hydraulic fluid, water and air, an umbilical connects the installation vessel with the central hub of the Vibro-drill. At the end of each arm, eccentric weights are used to vibrate the pile tip.
The Vibro-Polyp has a similar lay-out except for the arms and connection mechanism. Nine hinged arms form three sets of folding parallelograms to expand or contract the Vibro-drill. The connection is made through a set of pins on each arm that slide into holes in the pile shoe.

Due to unknown soil properties and effectiveness of jetting, liquefaction and fluidization, the loads on the structure cannot be determined deterministically. Therefore, a Monte Carlo simulation is used to perform the structural analysis based on a best guesstimate for the distribution of these unknown parameters. Based on this approach, a tool is created that aims at performing the structural analysis while iteratively improving the design. As test results will become available after finalizing this thesis, input for the unknown parameters can be updated to improve the results of the structural analysis.
Subsequently, based on this progressive insight, the design of the Vibro-drill can be optimized further. Note here that for the structural analysis, the system is assumed to be quasi-static which is a gross simplification of reality. Although this imposes limitations to the reliability of the results, this assumption is allowable for the preliminary re-design phase considered in this thesis. The structural analysis described here has been applied to both suitable concepts. For each concept, the external loads were identified and the internal loads were calculated. A series of structural checks in the Monte Carlo simulation provided a probability of failure. For both concepts the probability of failure was initially too high and therefore improvements to the structural components were made.

Based on the design criteria, the two optimized design concepts were compared. Due to the smaller frontal area of the Vibro-Polyp its soil resistance on the tip of the monopile and the Vibro-drill is the smallest and therefore requires the least eccentric force. Nevertheless, the structural analysis shows that its hinged arms are fatigue sensitive yielding a higher probability of failure. Therefore, despite its larger frontal area, the Vibro-Boxer has a higher reliability and is the most suitable concept for the Vibro-drill.
Finally, a more detailed design of the Vibro-Boxer concept is delivered that complies with all design requirements. For this design, recommendations are given on how to update the design based on the dynamic analysis and future test results.

Contents

List of Figures

List of Tables

Nomenclature

ALARP	As Low As Reasonably Possible
CPT	Cone Penetration Test
DAF	Dynamic Amplification Factor
ISO	International Organization for Standardization
MCA	Multi-Criteria Analysis

α	Pile-clay friction factor
β	Angle
β	Shaft friction factor
$\Delta\epsilon$	Strain range
η	Efficiency
γ	Effective unit weight of soil
γ_{DAF}	Safety factor for Dynamic Amplification
μ	Friction coefficient
ω	Angular frequency
ρ	Density
σ	Stress
σ_{yield}	Yield strength
b	Fatigue strength exponent
c	Outer fiber distance
e	Eccentricity
f	Frequency
f	Unit skin friction capacity
H	Height
L	Length
M	Moment
m	Mass
N	Number of cycles
N_q	End bearing capacity factor
p	Pressure

p_0	Effective lateral stress
P_{crit}	Critical buckling load
q	Distributed load
q	Unit end bearing capacity
Q_{skin}	Total skin resistance
Q_{tip}	Total tip resistance
Q_{tot}	Total soil resistance
r	Radius
S	Section modulus
s_u	Undrained shear strength
T	Torque
t	Time
V	Volume
W	Width
z	Depth
A	Area
E	Young's modulus
F	Force
g	Gravitational acceleration
I	Moment of inertia
SF	Security Factor

1

Introduction

During the past two decades the offshore wind industry has been constantly growing and developing. The wind turbines in modern wind farms are larger and more powerful than ever before and it is expected that this trend will continue as can be seen in Figure 1.1. To compensate for the greater loads on the turbines, the foundations are increasing in size as well. This has led to larger installation vessels which install monopiles using ever bigger impact hammers. An impact hammer installs a monopile by delivering a large impact on the top hammering it down with every single blow. With an increase in size of the monopile, the size of the impact hammer and the amount of energy released per blow increased as well. This causes more noise which disturbs the marine life by deafening or even killing animals. Even though regulations and noise mitigation methods are making an attempt to keep up, the impact of these hammers on the marine life increases every the year. Despite the fact that impact hammers are still the most used installation method, more quiet alternatives are gaining market share because of this problem. On a path towards the most effective method to install a monopile, many different concepts have been rejected. There seems to be a conversion towards using vibratory hammers instead of impact hammers as this method spreads out the energy over a longer period of time. This reduces the peak loads and the harmful noise. Besides, it is easier to scale up a vibration hammer than an impact hammer making it more suitable to cope with the increasing size of the monopiles.

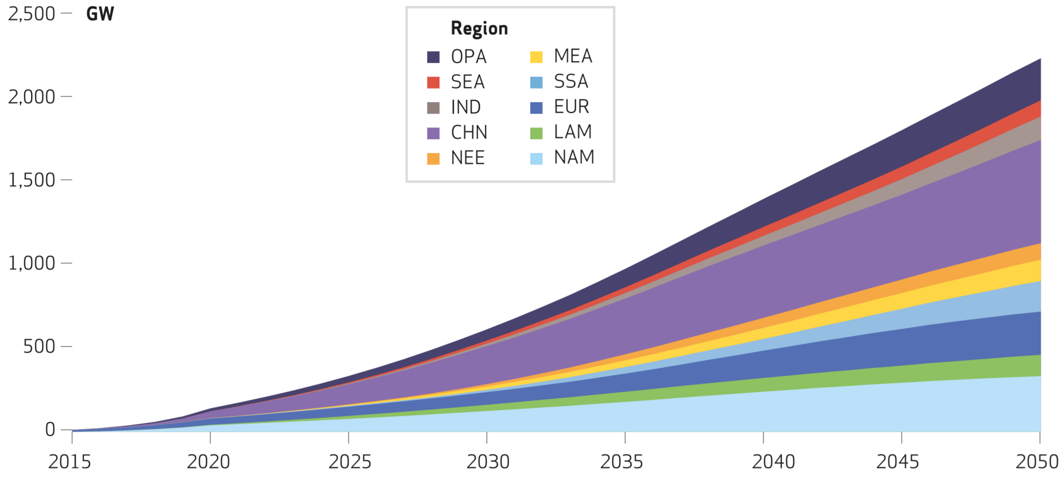


Figure 1.1: Worldwide forecast of offshore wind capacity by region (DNV-GL, 2017)

Besides the ecological motivation to find a better way of installing monopiles, there are also economic incentives for innovation. With low oil prices and reductions in subsidies the margins are decreasing. Every possible cost-saving avenue is explored and therefore also the monopile installation process needs improvement. A large cut in the costs are the rent of the installation vessel and bubble curtain. By decreasing the installation time and removing the need for noise mitigation costs can be saved.

GBM Works is developing a machine which is capable of silently installing a monopile at a lower cost than currently available alternatives. After several tests with prototypes a commercial version of the GBM Vibro-drill is now under development. The GBM Vibro-drill relies on three basic methods to reduce the soil resistance up to a point where the gravitational force of the the monopile and Vibro-drill is greater than the counteracting force from the soil resistance. At that point the monopile will sink into the soil without the need for an additional impact. The soil resistance can be split up in three sections namely; the tip resistance, the shaft resistance on the inside of the monopile and the shaft resistance on the outside of the monopile. For each section a separate method is used to reduce the resistance.

By jetting water down from the tip of the monopile the soil tip resistance is significantly reduced. A low pressure jet is used for sandy soils and clay is cut into small pieces by high pressure jets. The soil will be flushed away by the flow of water reducing its ability to resist the movement the monopile.

There is also water flowing from the top of the Vibro-drill towards the top of the monopile. Here the function is different. The large volume water fluidizes the soil on the inside of the monopile, creating a water-soil mixture which flows freely along the inside of the monopile. The water-soil mixture then reduces the soil resistance.

The shaft resistance on the outside of the monopile is reduced by vibrations which are generated by eccentric weights on the bottom of the monopile. The vibrations in the monopile make the soil particle move and this results in the soil temporarily acting as a liquid; liquefaction. Depending on the soil type the frequency and amplitude can be adjusted to improve the effect of liquefaction and with that the driving speed.

The combination of these three methods results in a silent and quick way of installing offshore monopiles. Thanks to the scalability of the Vibro-drill, GBM Works will be able to install larger monopiles cheaper and in agreement with strict noise regulations in the near future.

1.1. Problem Description

After installation the Vibro-drill is located under the tip of the monopile thirty to fifty meters below the seabed. The current prototypes for the Vibro-drill are not designed to be retrieved causing them to stay in place after use. Leaving the Vibro-drill behind after installation means that for every single monopile one Vibro-drill will be needed. As the cost of a Vibro-drill is greater than the cost of a bubble screen, this installation method would not be economically viable. Retrieval and re-use of the Vibro-drill resolves this problem. Finally, the economic benefits of re-using the module saves a lot of material, fabrication costs and time making it a sustainable alternative to the impact hammer. Retrieval is therefore a crucial functionality which needs to be added to the current design.

Before the Vibro-drill can be retrieved it needs to detach from the monopile. This requires that the connection which keeps the Vibro-drill in place has to be temporary. Besides this property, it is also important that the connection between the Vibro-drill and the monopile is able to resist all forces acting upon it during installation such that it maintains all of its current functionalities. During the whole process of mounting the Vibro-drill on the monopile, installing the monopile and retrieving the Vibro-drill it cannot be damaged beyond repair as this will eliminate the possibility of re-use.

During an early exploration only the connection between the monopile and the Vibro-drill was researched as this component is crucial for retrieval. It was concluded however that the design of this connection is dependent on the design of the Vibro-drill and vice versa. From this conclusion it follows that the current design can not be adjusted to comply with the desired functionalities in the most optimal way. Therefore the Vibro-drill needs to be redesigned for all of the functionalities described above. This leads to the following research question:

"How can the Vibro-drill be redesigned such that it can be retrieved after installation without losing its functionalities during installation?"

The challenge in this design exercise is to cope with uncertainties. To design the Vibro-drill such that it is still functional during installation it needs to be able to resist all loads acting on it. Because soil is unpredictable the soil resistance does not have a deterministic value. Besides this, the methods used to reduce the soil resistance have not been tested on this scale before and are therefore also uncertain.

The sum of these uncertainties gives a broad range on the demand of the structural components making the design process difficult. A compromise, based on uncertain parameters, must be made between a low probability of failure and a lean and lightweight structure.

1.2. Aim of the report

This report provides insight into possible solutions for designing the Vibro-drill. It describes a systematic path towards a suitable design which meets the design requirements. The reader is informed about critical design choices and arguments for the rejection of other solutions. A final design is presented and explained which is meant for GBM to use for further development of the Vibro-drill. This design should meet all requirements for normal operations, retrieval and reuse. Besides the advantages, disadvantages, possible improvement and risks of the design are discussed. Together with test results, this report provides the basis for a commercial version of the Vibro-drill.

1.3. Scope of the project

For the design of the Vibro-drill currently used technologies, as well as new unproven technologies are reviewed. From this broad range of solutions a selection is made based on basic calculations. Due to time restrictions only the concepts that have a high potential are analysed in more depth, in terms of design and structural capacity. A 3D model is made of the most suitable concept, including all main functionalities and some secondary functions. This final design will be optimized to increase its functionality and decrease its cost. The required magnitude of force from the eccentric weights and dimensions of the monopile are given by GBM Works. Also the frequency range of the vibrations, engine type and power and water supplies are fixed parameters. The transportation of the modules from the tip of the monopile to the installation vessel is not a part of the scope. It is assumed that when the module is released from the monopile a set of cables will pull it through the fluidized soil before it is retrieved from the top of the monopile. Finally the type of loads in combination with the structural behaviour and unpredictable soil causes a complex system of components with inertia and unknown amplitudes, varying frequencies, stick-slip behaviour and damping which require complex dynamic models if one were to solve this. As this is a graduation project on its own, it is decided that a simplification of this system will suffice for the design process of the Master thesis.

1.4. Approach

To form the basis of the project a problem analysis is performed in which a sharp research question is composed. From a process analysis the required functionalities of the Vibro-drill are determined and based on these the design is split into four separate components which together form one concept. Both innovative concepts and current methods from the offshore, pile driving and subsea engineering sectors are used as an input for a broad range of solutions for each of these components.

The load analysis follows from the process description and is used as an input for the requirement check. These values are based on estimates and assumptions concerning the load on the structure. A more detailed load analysis which accounts for the uncertainties of the soil is used for the structural analysis at a later stage in the project. A literature study helps to quantify the soil parameters which can be encountered during the installation of the monopile using the Vibro-drill. The process description and load analysis will help identify the design requirements to which the possible solutions are tested. Basic calculations help identify show-stoppers in a quick manner and produce a selection of potential solutions.

All the potential solutions are then qualitatively reviewed and graded based on design criteria in a Multi-Criteria Analysis (MCA). Based on those results a few solutions are combined into two concepts with a high potential.

These concepts are then worked out into more detail and will undergo a structural analysis to secure a desired probability of failure. The dimensions of the structural components are determined using a Monte Carlo simulation in which the uncertainties of the demand of the structure is incorporated. After a final comparison the most suitable concept is chosen for the detailed design.

In the final, stage water lines, hydraulic hoses, hooks and other details are added to complete the design. Also minor components can be adjusted to improve the design in terms of cost and/or weight. A conclusion will list all of the important findings and design choices which led to the final design. Finally a discussion will elaborate on possible improvements for GBM and highlight uncertainties in the design process which should be tested.

The approach for the project is displayed in Figure 1.2 and clearly shows a design path converging towards the final design.

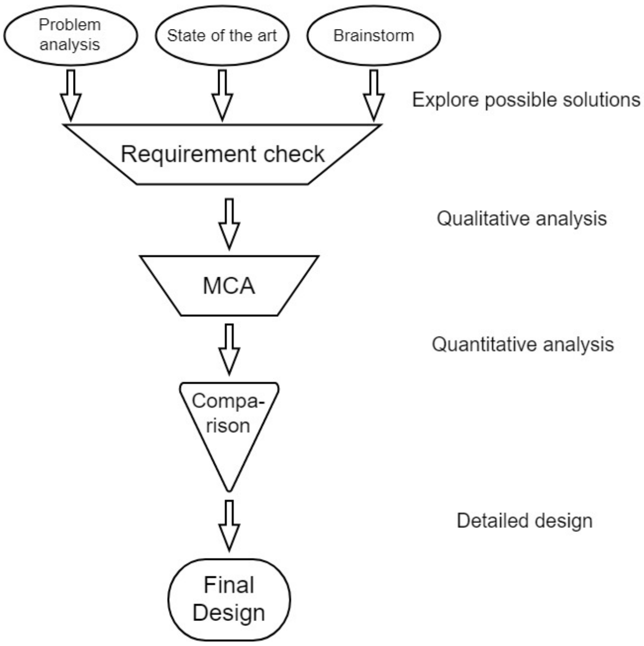


Figure 1.2: Schematic drawing of the design path

1.5. Report structure

The report starts with an investigation on the problem and a brief overview of the state of the art in Chapter 2. Afterwards three stages of the design path lead to the final design followed by a conclusion and a discussion. Part I is the Initial design phase. The third chapter contains the design requirements including the process and load analysis. Chapter 4 describes and tests numerous potential solutions that follow from the brainstorm and state of art overview. Finally the qualitative analysis and MCA are presented in the same chapter and two concepts follow from this.

The concepts that are determined to be of high potential are elaborated on in the first chapter of Part II. Then a structural analysis is performed to dimension the structural members and determine the probability of failure. A comparison between the concepts points out the most suitable solution for the design. Arguments for this choice are mentioned in Chapter 7.

Secondary functions and more detailed components are added to the design in Chapter 8 in Part III. Characteristics and properties of the final design are explained here as well. The global overview of the design will be accompanied by renders of the 3D model for a better understanding. Finally Part IV consists of the conclusion of this report. In the same chapter some statements from the conclusion are discussed. Furthermore weaknesses of the design, risks and ways to mitigate these are explained. The report will end with recommendations to GBM.

Throughout the report references will be made to the appendices, Part V, for additional information on certain subjects. These are to be found after the bibliography.

2

State of the art

The offshore wind industry has many years of experience when it comes to installing monopiles. Besides the offshore wind industry, the oil and gas industry has gained knowledge on the installation and handling of equipment offshore. The research that has been conducted in these sectors will be used for an exploration of the current state of art. Additionally land-based pile driving technologies will be reviewed. The result of this initial exploration is an overview of current products and technologies that might be implemented in the design of the Vibro-drill.

As mentioned in the approach, Section 1.4, the design of the Vibro-drill will be split up into four components. The main points of interest for an engineering point of view are the connection mechanism and the extending structure. For this reason they are both separate components. The other two major design choices to be made are the surface to which the connection mechanism is attached and the number of arms/units the Vibro-drill has. The connection can be placed directly onto the monopile, but it is also possible to add some material to the monopile which creates a surface to attach to with more grip, margin for misalignment or force. The number of arms/units influence not only the force on each of the arms, but also their size. During this exploration of the state of the art the first two components, the connection mechanism and the extending structure, can be compared to currently used machinery in similar sectors. This subsea equipment has a lot of basic principles to cope with the harsh underwater environment. The chapter is therefore divided in three main topics, namely: The connection mechanism, the extending structure and the subsea equipment.

2.1. Connection mechanism

To temporarily connect something to a (mono)pile several operations have to be investigated. Upending, pile driving, pile dredging, (internal) pile cutting and lifting are some of them. Besides temporary connections, also permanent connections might be altered to be removable. Below, relevant options, which are available on the market right now, are discussed.

Hydraulic grippers are being used in the onshore and offshore pile driving. Especially in combination with vibratory hammers, hydraulic grippers are commonly used. The fact these machines are already being used for this purpose under these vibrations and in similar conditions proves their potential for the Vibro-drill.

For the handling of monopiles quite often hooks and latches are used, but while this is a good option for lifting for connecting the Vibro-drill to the monopile it is not. Loads and vibrations must be passed on to the monopile. Other types of pile handling tools, mostly for smaller diameter piles, use internal presses that move outside. For the Vibro-drill this is not an option as these machine plug the whole inside creating a lot of soil resistance. A similar system which takes up a smaller frontal area might be looked into.

Bolting is used on transition pieces and automatic bolting machines are developed for underwater use. The use of these machines for on the Vibro-drill may be interesting to investigate.

2.2. Extending structures

For extending structures, mobile cranes can provide solutions. As these machines extend in a relatively small area their scissored structure might be useful for the Vibro-drill. Using a well calculated system of arms and hinges allows for movement in a single direction. Sideways loads in combination with vibrations may cause bending and/or fatigue in the system. This risk must be checked in the structural analysis.

Another solution may be a drawer like arm which is powered by a hydraulic piston. When an arm is guided along a slider it can take loads in certain directions as long as the correct sliding mechanism is used. The overlap between the different part of the slider is critical when a large force is applied on the end of the extending arm. Once again this solution has potential when the structural rigidity can be proven in the structural analysis.

2.3. Subsea equipment

Subsea equipment is developing rapidly as technology improves. At the same time underwater robotics are more often used in the offshore industry. From this sector details for the design can be used. As certain methods technologies don't work under water or need special housings, looking at the subsea equipment can help. The reliability of the Vibro-drill must be guarantied even though the machine is worked $40m$ below the sea bed surrounded with sea water and soil particles.

For lubricated equipment the housing should be water and air tight, but also the chamber should be filled with a combination of lubrication and air to keep the flow moving.

All other moving parts should be shielded from the surrounding seawater and soil as much as possible. Scratches, wear and tear in combination with vibrations can cause growing hair cracks. During the detailed design these remarks need to be applied.

I

Preliminary design

3

Design requirements

After the exploration of the problem and state of art this chapter will provide the first steps towards a solution. Design requirements follow from a process analysis and a load analysis. In Section 3.1 one operational cycle is described with the goal of finding main and secondary functionalities, determining geometry requirements and identifying load cases. These load cases are then quantified in Section 3.2. A summary of the design requirements but also a list of design criteria will be given in the last section of this chapter. In Chapter 4 these requirements will be used to check possible solution and make a first selection of potential solutions and in Chapter 7 the design criteria will be used to determine which is the most suitable concept.

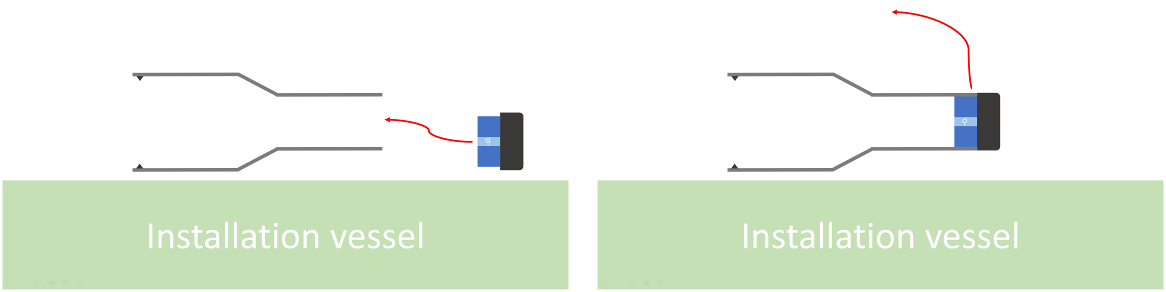
3.1. Process analysis

In order to identify all design criteria for the Vibro-drill and the monopile, the full process is described in detail using schematic drawings. Because the Vibro-drill has never been used to install a monopile assumptions need to be made and solutions must be found when problems arise. The result of this is the process as described below in three parts namely; before the installation of the monopile, during the installation of the monopile and after the installation of the monopile. Because the Vibro-drill requires different functionalities during these parts, the process and corresponding requirements are discussed separately. The conclusion of this section will form the basis for the load analysis and design requirement in the remainder of this chapter.

3.1.1. Before installation

Before the installation of the monopile, the Vibro-drill needs to be mounted on either the monopile itself or an attachment on the monopile. If adjustments to the monopile are required, these must be placed right after fabrication of the monopile such that it is completed before the monopile is loaded on the installation vessel. The Vibro-drill (depicted in blue) needs to be picked up and placed inside the monopile as depicted in Figure 3.1a. This may require specialized equipment, but using already present equipment is preferred. The Vibro-drill will be accompanied by a crane hook, suppressor and winch system (depicted in black). These additional systems are not part of this design exercise and therefore no further specifications are provided.

Mounting the Vibro-drill on the monopile will be done after upending the monopiles on the installation vessel, on a barge or in the water. The Vibro-drill is lowered from the top of the monopile to the bottom as can be seen in Figure 3.2 and 3.3. Because the difference in diameter, the Vibro-drill may need to extend itself before it can attach itself to the monopile. At this point the Vibro-drill is hanging inside the monopile without support from below and therefore this is a crucial process.



(a) Step 1/12 (b) Step 2/12

Figure 3.1: Installation procedure of a monopile (step 1+2)

Mounting the Vibro-drill in the monopile on land is safer and most likely cheaper, as there are fewer restriction in work area and time. The downside is that for every monopile that is in transport, waiting on installation or being installed, a Vibro-drill is needed. It will therefore require more Vibro-drills to install a wind farm than when they can be placed on the monopiles while sailing to the next location. Because of this the Vibro-drill will be mounted on the monopile on the installation vessel and is thus used for all of the monopiles in the wind farm. This also has major consequences for the life cycle and thus load analysis in Section 3.2.
Finally the monopile with the Vibro-drill in place is lifting of the deck off the installation vessel (depicted with the red arrow) and lowered onto the sea bed for installation. At this point the hoses which power the eccentric weights and feed the jetting and fluidization nozzles are inside the umbilical cord (depicted in black) and are guided from the installation vessel through the top of the monopile.

(a) Step 3/12 (b) Step 4/12

Figure 3.2: Installation procedure of a monopile (step 3+4)

(a) Step 5/12 (b) Step 6/12

Figure 3.3: Installation procedure of a monopile (step 5+6)

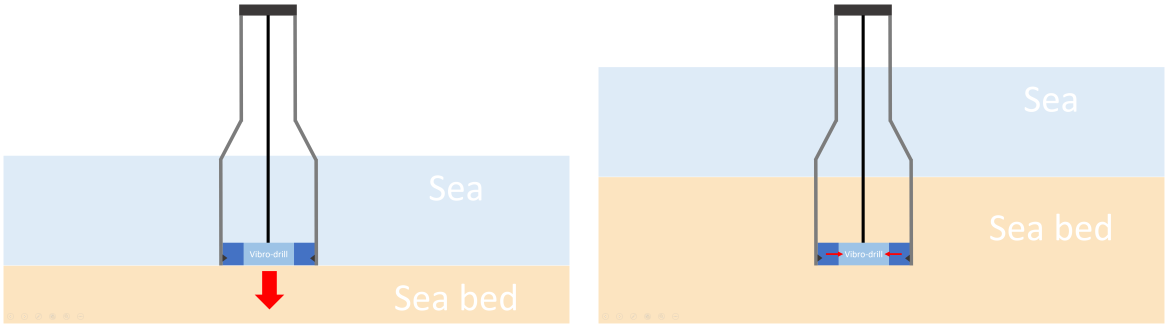
3.1.2. During installation

As the monopile is lowered on the sea bed, the Vibro-drill starts operating to reduce the soil resistance, allowing the monopile to sink into the soil. From this point until the moment the monopile is in place, the Vibro-drill is jetting, fluidizing and vibrating. Depending on the soil type the jetting should flush and/or cut the soil at the tip of the monopile and Vibro-drill. Sandy soils will mix with the water and are flushed away by the jetting. Clay is cut by the high pressured jets first and then flushed away. Nozzles should cover all downwards facing planes of the Vibro-drill and monopile to maximize the reduction in tip resistance. This means the bottom of the monopile should therefore be covered by either the Vibro-drill or attachments.

The sides and top of the Vibro-drill need nozzles for fluidizing the soil column inside the monopile. By mixing the soil with water a mud-like substance will be created inside the monopile. When the soil column is completely fluidized it will behave more like a fluid resulting in a drop in shear strength. This effect causes the shaft resistance on the inside of the monopile to drop drastically. Just like for jetting, for fluidization, it is also important that the water flow is adjusted to the soil type for the most effective use of water and pump capacity.

Liquefaction is a phenomenon which occurs when the pore pressure in the undrained saturated soil equals the effective confining stress. This can be caused by vibrations in the soil which move the particles such that they are suspended in the voids and the confining stress is reduced. During each cycle the contact between particles is lost and the shear strength. The result is cyclic mobility of the soil (Thomas and Vaid, 1995). As the vibrations are passed on to the monopile this effect takes place all along the shaft of the monopile reduces the shear strength and with that the soil resistance. For the liquefaction of the soil, it is important that all the eccentric weights are in phase and thus mechanically connected to each other. The monopile itself will be vibrating as well and thus liquefying the soil surrounding it. This requires a connection that will not only resist the vibrations, but also pass them on to the monopile without too much damping.

When failures occur to the Vibro-drill or connection, redundancy should prevent the monopile from becoming stuck without a chance of retrieval. The loss of a monopile is extremely costly as not only the monopile but also time and an installation location are lost. The risk of such an event should be As Low As Reasonably Possible (ALARP).



(a) Step 7/12 (b) Step 8/12

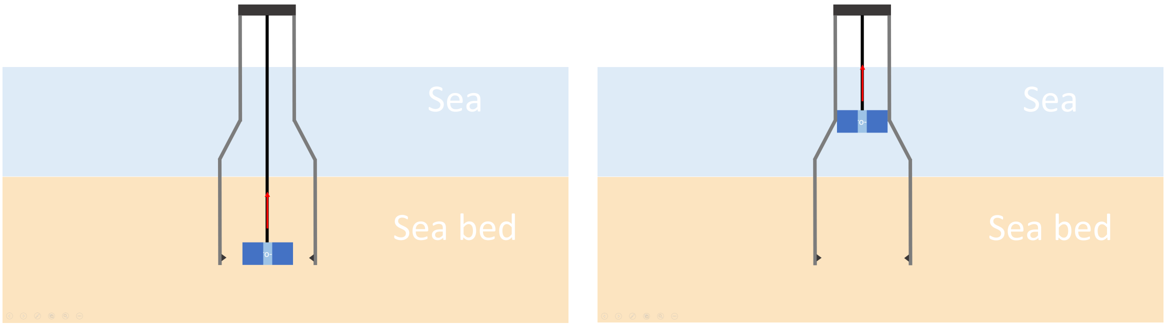
Figure 3.4: Installation procedure of a monopile (step 7+8)

3.1.3. After installation

As soon as the monopile is in place the jetting will stop as cavities may form under or outside the monopile which reduce the bearing capacity of the monopile. The vibrations are no longer required and the connection with the monopile will be broken. As the Vibro-drill will be retrieved from the inside, the soil there needs to remain fluidized. It therefore is important to keep the flow of water going on the inside of the monopile.

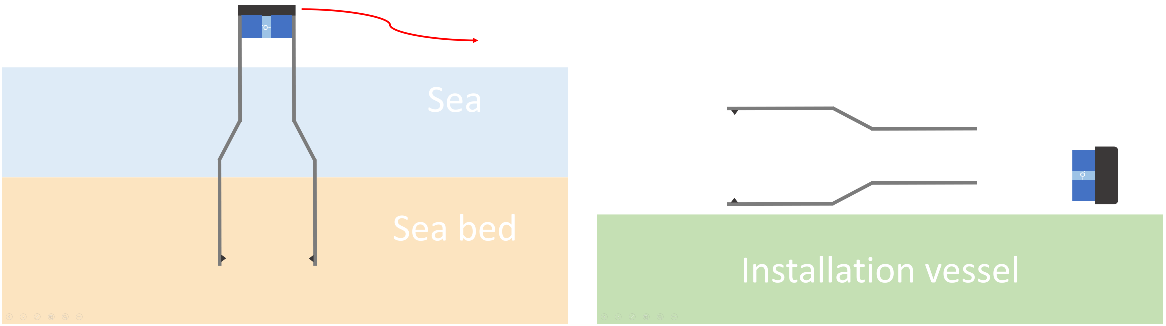
Before the Vibro-drill can be retrieved, the connection with the monopile needs to be released remotely. Large monopile are tapered along the length as a larger diameter in soil provides a greater bearing capacity and a greater resistance against the bending moment, but at the same time a smaller diameter above the seabed reduces the hydrodynamic loads and the weight of the monopile (Cathie et al., 2019). Because the top of the monopile has a smaller in diameter than the bottom, the Vibro-drill needs to decrease in size in order to fit through the top. A set of cables and winches will pull the Vibro-drill in one or multiple pieces through the top of the monopile. The Vibro-drill will then be placed on the deck where it can be mounted on another monopile while sailing to the next installation location.

For retrieval, the Vibro-drill needs to be pulled up through the soil in the monopile. In the worst case the fluidization doesn't work and the Vibro-drill will act as an anchor. The force needed to pull up the Vibro-drill will never be higher than the soil resistance it had to overcome to go down. This load is therefore covered by the worst case operational load as described above. Depending on the connection and release mechanisms and other moving or fatigue sensitive components the Vibro-drill might need repair, maintenance or checks in between installations.



(a) Step 9/12 (b) Step 10/12

Figure 3.5: Installation procedure of a monopile (step 9+10)



(a) Step 11/12 (b) Step 12/12

Figure 3.6: Installation procedure of a monopile (step 11+12)

3.1.4. Conclusion process analysis

Before installation the Vibro-drill is to be mounted on the bottom of the monopile and the connection must resist the forces acting upon it during the complete installation process. The cables have to be pulled through the monopile and connected to the pumps and/or power supply. The Vibro-drill further needs to extend itself in order to grab onto the monopile or attachments to the monopile.

During installation the Vibro-drill has to vibrate, jet water downwards and fluidize the soil above it. When it is operational the connection has to keep the Vibro-drill in place and pass the vibrations on to the monopile. Also the connection between the eccentric weights should keep all of them in phase to maximize the vibration. The structure of the Vibro-drill should be able to resist the vibration loads, the soil resistance and gravitational forces acting upon it.

For the retrieval and demounting of the Vibro-drill it is important that the connection can be broken from the installation vessel and that the Vibro-drill can be folded or broken down into smaller bits in order to fit it through the top of the monopile which has a smaller diameter. When the Vibro-drill is retrieved it has to be re-usable, preferably with the least amount of repair and/or maintenance to keep the costs low.

3.2. Load analysis

As stated in the process description above the Vibro-drill has to survive the installation of a complete wind farm. In order to do so, the structure and the mechanical components must remain intact and functional during approximately a hundred monopile installations. Because not all of the forces acting on the Vibro-drill can be simulated, a selection of three load cases has been made. The first being a very simple load case which

is only used to do basic estimations in Chapter 4. Here possible solutions for the connection mechanism are checked on their potential to hold the Vibro-drill in place. For the Structural Analysis in Chapter 6, two other load cases will be used. These load cases are more detailed and also suited to the geometry of the concept. In the second subsection the approach for the Structural Analysis is described such that the required input can be determined. After this, these parameters are explained and quantified.
Before the load cases are described first six assumptions are made concerning the loads and load cases.

- Throughout this project a monopile with a length of $80m$, a bottom diameter of $8.5m$ and a top diameter of $5.5m$ will be used. These dimensions are expected to be representative for monopiles used in the wind farms being build in the next couple of years. The wall thickness is constant at $100mm$ and with a density of $7800kg/m^3$ the mass of this monopile is approximately $1400mT$.

- During operations the Vibro-drill and monopile are excited in frequencies up to $1400RPM$ or $23.3Hz$. With the first (longitudinal) eigenfrequency being approximately below this frequency for a free modelled monopile, this means that the inertia plays a role in the internal loads (of both the Vibro-drill and the monopile) can't be neglected (Hermans and Peeringa, 2016), (Cook et al., 2002). Also the damping along the monopile and the stick-slip behaviour between the monopile and the soil make it very complex to perform a proper dynamic analysis. Considering the duration and goal of this project, it was decided that a simplification to a quasi-static system will suffice for this design exercise. For the fatigue damage the sinusoidal force of the eccentric weights is used to determine the strain range and to compensate for the effect of dynamic amplification on the internal loads a safety factor γ_{DAF} is used. The results will have their limitations in terms of reliability, but the consequences of this assumption are discussed in Chapter 9.

- It is assumed that the Vibro-drill inside the monopile does not contribute to the structural strength of the monopile in any direction. In the structural analysis the monopile will be modelled as a rigid body. Therefore the contact point on the monopile are considered fixed in place in the static system.

- As rocks can be present in the soil, a possible load case would be a rock hitting the Vibro-drill. During geotechnical surveys on the location medium and large sized rocks are most likely to be located and avoided, but small rocks could apply point loads on the structure. Because the rock is supported by soil, the total load of the rock will never surpass the normal soil resistance of the it covers. This load will result in higher local stresses and possibly in higher global stresses. As the risk of a rock hitting an arm is evident a separate rock impact analysis will be performed based on the geometry of the design. A rock of $1.0m$ will perform a load on the worst case location on the arm and the structural analysis will be repeated. If necessary improvements are made to ensure structural integrity.

- In between the installations of monopiles minor repairs and/or checks can be done to the Vibro-drill. For crucial components, for example connections and welds, recommendations will be made to prevent the service limit state to be surpassed on high cycle components.

- The resultant forces from the jetting and fluidization are negligible as these are several orders of magnitude lower than the other forces acting on the system.

3.2.1. Load cases

As mentioned above the first load case is a simplified version to get an estimate of the force the connection needs to be able to withstand. The second load case is apparent during the mounting of the Vibro-drill on the monopile. As can be seen in Figure 3.2b, the Vibro-drill needs to extend to fit the larger diameter of the monopile. A large moment is expected here as the Vibro-drill is hanging from one central point. Finally the third load case will occur during operational loads in tough soil conditions. The load cases are summed up briefly below and elaborated on afterwards.

1. In the first load case it is assumed that in a worst case scenario the soil resistance is too high and keeps the monopile fixed in place. The weight of the monopile and the downwards force of the eccentric weight try to move it. At this point the sum of these two forces act on the connection. Therefore the force acting on the connection is equal to $F_{gravity} + F_{eccentric}$. The values of these forces is determined below.

2. The load acting on the structure of the Vibro-drill during extending is mostly dependent on the geometry of the structure and design of the housing for the eccentric weights, as they are most likely to cause the biggest moment on the structure. As the concepts are to be made in a later stage of the project, the value of this load will be determined in the structural analysis.
3. The third load case is similar to the first, but a lot more detailed as the soil resistance is split up in multiple parts and is acting on different areas of the Vibro-drill and monopile. Also the geometry of the concept is taken into account. More importantly, this load case will be used to determine the structural demand of critical components and the probability of failure of the Vibro-drill.

Load case 1

The load on the connection equals the sum of the static load resulting from the monopile and the peak of the dynamic force from the eccentric weights. The static load equals the weight of the monopile and the Vibro-drill minus the friction force on the shaft due to soil resistance and minus the load carried by the crane which is required to keep the monopile straight up. In an ideal situation, the vibrations and fluidization will reduce the soil resistance. It is therefore a conservative assumption to say that the fluidization and liquefaction fully eliminate the shaft resistance.

The crane will carry some of the total weight in order to keep the monopile in position and submerged weight of the monopile is less than its weight in air. This reduces the load on the Vibro-drill on the bottom of the monopile. The weight of the Vibro-drill would compensate this however and thus this load can be simplified with the following equation in which g is the gravitational acceleration.

$$F_{gravity} = m_{monopile} * g = 1400e3 * 9.81 = 13.7MN \tag{3.1}$$

The dynamic load is caused by multiple eccentric weights rotating counter directional. The resulting force from this movement is a sinusoidal force which changes between up and down at the same frequency the axis of the weights are turning. For this load case the eccentric weights will be assumed to turn at a frequency of $23.3Hz$ or $147rad/s$, which is similar to what is being used in vibratory hammers of equal capacity. The product of the weights and their eccentricity is estimated, using Equation 3.2, to be around $1200kgm$ as this would provide a large enough force to vibrate the monopile and liquefy the surrounding soil. A lower frequency requires a higher eccentric moment and vice versa. The relation between these two parameters is given below.

$$F_{eccentric} = m * e * \omega^2 * sin(\omega * t) = 1200 * 147^2 * 1 = 25.9MN \tag{3.2}$$

As stated in Equation 3.3 the total load on the connection is the sum of $F_{gravity}$ and $F_{eccentric}$ which equals $39.6MN$. In Chapter 4 all possible connection mechanisms will be checked on this load.

$$F_{total} = F_{gravity} + F_{eccentric} = 13.7 + 25.9 = 39.6MN \tag{3.3}$$

Load case 2

When the Vibro-drill is hanging inside the monopile it is attached to the umbilical. This connection is along the axis of the monopile, while a lot of the weight is located away from the center, close to the wall of the monopile. The gravity pulls this weight down and creates a large moment on the structure. This requires the structure to be rigid even though it is not supported from the bottom as it is during operations. This load case is fully static as the Vibro-drill is not yet operational. No soil or water is involved at this point. The only forces acting on the Vibro-drill are gravity and resulting internal moments. The structure is therefore modeled as a simple static system based on the design of the Vibro-drill concept. This model, including the values for the loads, will be discussed during the structural analysis as the concept is to be designed in Chapter 4.

Load case 3

Figure 3.7 depicts the Vibro-drill during installation of the monopile. The soil is surrounding the monopile and Vibro-drill and thus on all surfaces soil resistance (in yellow) is to be expected. As the soil conditions which the Vibro-drill will encounter are uncertain, assumptions need to be made. At the same time the Vibro-drill is operating in order to reduce these forces, but the effectiveness of these methods are unknown. The gravity (in red) acts on the monopile as well as the Vibro-drill and finally the eccentric weights cause an internal load (in orange) on the Vibro-drill structure and the connection between the Vibro-drill and monopile.

The result is that the exact loads cannot be determined, but a statistical approximation must be made. In this process the distributions for the parameters above are determined based on literature and test results. This will be done in the next subsection.

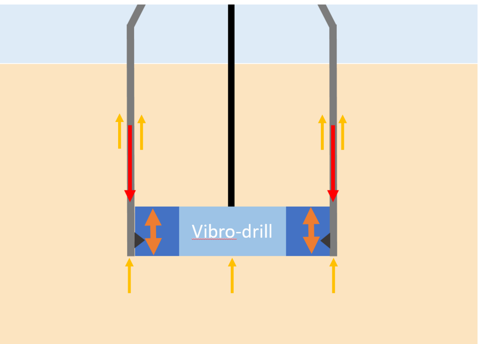


Figure 3.7: Schematic drawing of Load case 3

Just like the previous load case the magnitude and contact surfaces follow from the geometry of the concepts. These geometries will be elaborated on in Chapter 5 and the loads will be determined in Chapter 6.

3.2.2. Statistical load estimation

As explained in the problem description, Section 1.1, the uncertainties in the loads prevent the use of basic equations to determine the loads on the structure. The types of loads acting on the structure are the same for all concepts, but the point where the load acts on the structure and the magnitudes may differ. It is however possible to determine a set of parameters, to be used as input for the structural analysis. Figure 3.8 displays such a set of parameters. In the structural analysis these parameters will then be combined with the geometry to calculate the forces on the structures of the concepts chosen in Chapter 4. These parameters are identified and described below.

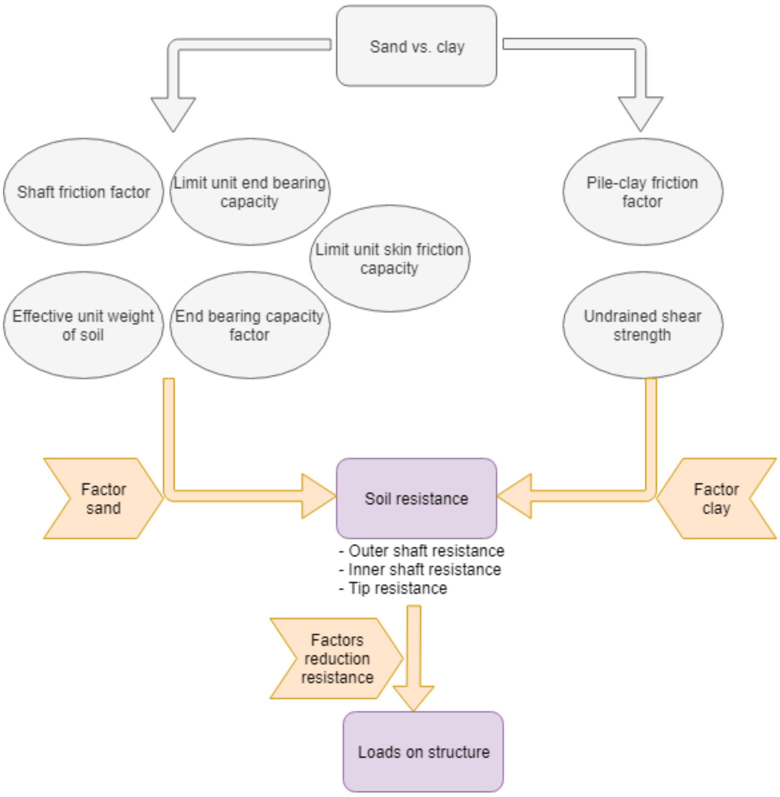


Figure 3.8: Schematic drawing load analysis

The maximum theoretical soil resistance a surface could produce overestimates of the real world values. A purely theoretical worst case scenario would lead to an over dimensioned design which would cost a lot of material and money. When using empirical loads on the structure they may be underestimating the worst case the Vibro-drill will encounter. A compromise between a low risk of failure and a lean non-over dimensioned structure needs to be made. This is done by calculating a probability of failure based on the input parameters. The probability of failure should be within a reasonable level and to determine this load a statistical simulation is used.

A one stage Monte Carlo simulation will be used with the following parameter for the determination of the loads on the structure which are caused by the soil resistance. The soil properties are determined based on a literature study. Values for the reduction in soil resistance however, cannot be found in literature as they do not exist. Assumptions, based on logic and previous test results from GBM, are used to determine a conservative distribution for these parameters. First the parameters are described and afterwards the distributions are repeated in Table 3.2.

- Initial unit external skin bearing capacity
- Reduction in unit external skin bearing capacity
- Initial unit internal skin bearing capacity
- Reduction in unit internal skin bearing capacity
- Initial unit end bearing capacity
- Reduction in unit skin bearing capacity

The initial unit external skin bearing capacity is the bearing capacity per unit of area of the outside of the monopile. This resistance needs to be overcome before the soil moves along the outside of the monopile. Due to different soil properties on all locations and even at the same locations, it is impossible to calculate a single value. For clay and sand different equations are used, additionally for layered soils this becomes even more complex. Following Equation 3.4 and 3.5 with the parameters in Table 3.1 it can be concluded that very dense sand gives the highest shaft resistance of these soil types. According to Dean the layers with high unit skin friction values will lose some strength when weaker layers are near. As the weaker layer can not carry the load acting upon by the stronger layer, it will reduce the overall resistance in the border region of the two layers. Therefore it is assumed that very dense sand will cause the highest loads on the structure.

The reduction in unit external skin bearing capacity follows from the liquefaction of the soil. Vibrations are passed onto the monopile with a certain frequency and amplitude and make the soil particles vibrate as well. The result is that the soil directly surrounding the monopile becomes liquefied. As there are vibratory hammers already on the market using this principle its effects can be proven. The effect of damping of the soil on the vibrations is not taken into account here. Because current vibratory hammers are mounted on top of the monopile the energy needs to pass through the full length of the monopile to get at the tip where the resistance is the greatest. Along this path the vibrations get dampened by the elasticity of the monopile and the soil surrounding it. As a worst case, the vibratory hammer can be used to determine the reduction in soil resistance on the outside of the monopile.

Just like the external friction the initial unit internal skin bearing capacity is also uncertain due to soil properties. The soil on the outside and inside of the monopile, before adjusting for the reduction due to fluidization and liquefaction, can be taken as equal and therefore the distribution for the parameter can be taken as equal.

The reduction in unit internal skin bearing capacity is mostly caused by fluidization. The water flow within the monopile is great enough to lift the particles and eliminate the cohesion, for clay, and shear strength between particles, for sand, making it behave like a fluid. In an ideal situation the resistance on the inside of the shaft would be equal to the drag due to a water flow with a velocity equal to the terminal falling velocity in water. In reality the effectiveness of this method has not been tested extensively enough to determine a factor. The combined effect of vibrations and fluidization on the inside of the monopile also requires testing before any conclusions on the failure of probability of the Vibro-drill can be drawn.

Also for the initial unit end bearing capacity the unpredictable soil properties cause uncertainties. The soil at the end of the monopile has a higher internal pressure and can have different soil properties than the soil along the shaft of the monopile. Due to the choice of worst soil conditions very dense sands are used in this load case.

Jetting in sandy soils will increase the soil pressure, reduce the internal friction and flush away some soil. In clay however it cuts the soil in smaller bits and flushes away some of it. Although jetting is proven to be effective in previous offshore operations as explained in Chapter 2. The nozzles, flow and soil properties are yet to be designed and the downwards facing area of the Vibro-drill might influence the effectiveness as well.

With the soil inside and outside the monopile having the same properties, the same values will be used for these parameters. This leaves five distributions to be determined.

Table 3.1: Soil characteristics (very dense sand hard clay) (Verruijt,1994)

Parameter	**Symbol**	**Value**	**unit**
Effective unit weight of soil	γ	9500	kN/m^3
Undrained shear strength	s_u	200	kN/m^2
End bearing capacity factor	N_q	50	–
Limit unit end bearing capacity	q_{lim}	12000	kN/m^2
Limit unit skin friction capacity	$f_{s,lim}$	115	kN/m^2
Shaft friction factor	β	0.46	–
Pile-clay friction factor	α	0.6	–

Equations for calculating the soil resistance for clay in which A_tip (frontal area tip of the monopile and Vibro-drill) is to be determined using the geometry of the Vibro-drill (Vugts, 2013).

$$f_{clay} = \alpha * s_u \tag{3.4a}$$

$$q_{clay} = 9 * s_u \tag{3.4b}$$

$$Q_{skin,clay} = f_{clay} * A_{skin} \tag{3.4c}$$

$$Q_{tip,clay} = q_{clay} * A_{tip} \tag{3.4d}$$

$$Q_{tot,clay} = Q_{skin,clay} + Q_{tip,clay} \tag{3.4e}$$

Equations for calculating the soil resistance for sand (Vugts, 2013).

$$p_0 = \gamma * z \tag{3.5a}$$

$$f_{sand} = min(\beta * p_0, f_{lim}) \tag{3.5b}$$

$$q_{sand} = min(N_q * p_0, q_{lim}) \tag{3.5c}$$

$$Q_{skin,sand} = f_{sand} * A_{skin} \tag{3.5d}$$

$$Q_{tip,sand} = q_{sand} * A_{tip} \tag{3.5e}$$

$$Q_{tot,sand} = Q_{skin,sand} + Q_{tip,sand} \tag{3.5f}$$

During the geotechnical survey a Cone Penetration Test (CPT) will be performed in many locations at the wind farm to determine the soil properties and potential bearing capacity of the monopile. The results from these tests can only give an estimate of the actual soil conditions the monopile will encounter during installation. It is therefore unknown what exact loads will be acting on the Vibro-drill and monopile. To cope with this uncertainty a distribution is taken from CPT measurements at wind farm Hollandse Kust (Klein, 2017). The sample with which the distribution is determined is small, but it will give a decent estimate for the range of soil properties to be found in a wind farm. The model is built such that all of the input parameters can easily be changed to check whether the design will meet the structural requirements in other locations as well. It should be noted that the probability of failure is not reliable until the input parameter is revised with new measurements.

From the CPT data from Hollandse Kust, the cone resistance has been measured at 69 locations. For the Monte Carlo simulation the distributions needs to be normally distributed in the form (Beck and Melchers, 2018). Therefore the CPT data from Hollandse Kust has been ranked and normalized. The statistical descriptive can be found in Appendix A. Again it needs to be stressed that the sample is too small to perform a proper simulation, but for this design exercise it will suffice as the model is build up such that new data, when available, can be used as input to recalculate the probability of failure.

The parameters for the Monte Carlo need to have independent distributions (Canfield et al., 2006). As for the soil this comes down to one single parameter which affects the total soil resistance. Also the parameters for the methods used to reduce the soil resistance do not influence each other resulting in three independent distributions for the reductions in soil resistance.
These reduction factors are arbitrarily taken between 10% and 80% truncated normal distribution within these limits. Test results with a medium scale prototype will provide data on which a more accurate estimate can be made.

3.2.3. Conclusion Loads

For the first load case a basic estimate has been made to determine the force on the connection between the Vibro-drill and the monopile that should be able to withstand the applied loads while the Vibro-drill is in operation. The force acting upon this connection is $39.6MN$. In the next chapter possible solutions will be checked for this requirement and others as stated in 3.3.

The second load case fully depends on the mass and thus design of the concept as its own gravitational pull will cause the loads on the structure. It can therefore only be determine in Chapter 6 after the geometries are determined in Chapter 5.

Finally the same applies for the third load case. The frontal area as seen from below and shaft area of the monopile will affect the exact soil resistance and with that the load. The soil characteristics and the parameters for the Monte Carlo have been determined in this chapter already and apply only to this load case. Table 3.2 shows the values for these parameters with the distribution for the soil resistance calculated based on the CPT measurements at Hollandse Kust and the other based on estimates.

Table 3.2: Distributions simulation parameters

Parameter	Min value	Max value	Mean	Standard deviation
Soil resistance	-	-	1	0.5062
Reduction jetting	0.1	0.8	-	-
Reduction vibrations	0.1	0.8	-	-
Reduction fluidization	0.1	0.8	-	-

3.3. Design requirements

The structural requirements, mentioned in the load analysis above, focus on the connection between the Vibro-drill and its structure. These will be displayed first as these are meant for selecting the potential solutions in the next chapter. Besides this, several requirements need to be met concerning the geometry, jetting, fluidization and liquefaction, moving parts and other components. These requirements will be listed afterwards. The practical requirements which are necessary for the dimensions and functionality of the Vibro-drill are used in a later stage of the design. From Section 3.1 the following requirements are identified as necessary for the connection:

- The connection must withstand the forces acting upon it before, during and after installation of the monopile. The first load case sets this benchmark.
- The connection must withstand the vibrations during operations and must pass these on to the monopile without too much losses.

- The connection can be disconnected remotely.
- The Vibro-drill must be repairable and re-usable after the installation of a monopile.

For the design of rest of the Vibro-drill the following requirements are set. They will be attributed to the concepts in Chapter 5.

- The Vibro-drill has to resist the loads acting on it as stated in Load case 2 and Load case 3.
- The Vibro-drill may not lose its functions before, during or after installation of the monopile.
- All eccentric weights must work in phase.
- The Vibro-drill needs to able to fit through the smaller top diameter of the monopile.
- The Vibro-drill must be able to attach itself on the larger bottom diameter of the monopile.
- The jetting holes must cover the tip area of the Vibro-drill and monopile.
- The fluidization holes must cover the top of the Vibro-drill.
- An umbilical cord must feed the Vibro-drill with the required electricity, hydraulic fluid, water and air.

3.4. Design criteria

As there are many more properties to the Vibro-drill that can enhance the efficiency and/or lower the cost of the Vibro-drill this chapter will end with these design criteria in this section. These criteria will be used to determine which concept is the most suitable in Chapter 7

- The probability of a loss of the monopile due to the failure should be a minimum.
- Vibro-drill must be cheap to manufacture.
- Minimal time/cost needed for repair.
- The frontal area of the Vibro-drill must be kept to a minimum to reduce the tip resistance.
- Easy fabrication (no rare machinery required).
- Easy and fast mounting procedure for the Vibro-drill.
- No or minimal adjustments to the monopile are preferred.
- Redundant connection system/back-up is preferred.
- Redundant release system is preferred.
- Use of proven technology is preferred.

4

Potential Solutions

In the first part of this chapter possible solutions for the connection of the Vibro-drill and other components will be checked based on the design requirements as stated in the previous chapter. The potential solutions that follow from this check are input for the Multi-Criteria Analysis (MCA) in the second part of this chapter. For the MCA it is useful to split the Vibro-drill into four separate components, namely the connection mechanism, the connection surface, the structure type and the number of arms. In Section 4.1 these components will be explained and a re-cap of the relevant design requirements is given. Afterwards four subsections will elaborate on the possible solutions for each of the components and the checks will be performed.

Section 4.2 uses the selection of alternatives and a set of criteria to determine the two most suitable combinations of alternatives. Subsections 4.2.1 and 4.2.2 explain the criteria to which the alternatives are scored and gives these scores. Finally some major issues will eliminate certain alternatives and/or combinations and two most suitable concepts are picked for further investigation.

4.1. Design check

As the Vibro-drill is retrievable the connection between the Vibro-drill and the monopile should be temporary and re-usable. At the time it has to withstand the loads acting upon it during operations. The first load case states that the total force acting on the connection mechanism equals $39MN$. This load is spread over the number of arms or number of connections. Finally the connection should also be solid as the vibrations must not be dampened. Because the Vibro-drill needs to fit inside the monopile and needs to be handled on the installation vessel its size and/or weight should not exceed practical limits. This is also looked into during the requirements check in 2.1.

The connection surface to which Vibro-drill attaches itself does not necessarily need to be the monopile. When the complexity of the design can be reduced by making the Vibro-drill mount onto a box or hook welded on the inside or bottom of the monopile, this option is to be regarded as well. The downside of adjusting the monopile obviously need to be outweighed by the benefits. Because the connection mechanism and connection surface are related to each other not all alternatives may be compatible with one another. This will be accounted for during the MCA.

Choosing the structure of the Vibro-drill is an important design choice as this forms the basis for the structural rigidity, but also possibilities for redundancy of parts and geometry limitations. It should be noted that also the structure is dependent on the other components and not all combinations might be possible. The most important feature of the structure is that it has to allow for the Vibro-drill to fit through the narrow top of the monopile while also being able to mount on the wider bottom.

The shape of the Vibro-drill will most likely consists of several connected units which each its own connection to the monopile and a number of eccentric weights. Because these weights must be driven with hydraulic engines. They must also be kept in phase with each other for which a physical connection is needed between either the units themselves or a central hub. The requirements for the Vibro-drill state a minimum required eccentric force, but depending on the number of arms this force is divided over a number of units. The Vibro-drill can consist of multiple units which are each connected to the monopile having their own connection and eccentric weights. By spreading the load over several units the connection can be made smaller and the eccentric weights can be divided. Making the Vibro-drill more complex is the downside however and thus a compromise must be made.

4.1.1. Connection mechanism

Each possible connections mechanism is discussed separately and only solutions that have the potential to connect the Vibro-drill and monopile are taken into account for the MCA. It should be noted that the load as stated in the first load case is in vertical direction along the monopile shaft and will therefore cause a shear stress on the connection. For the following connection mechanisms the required surface and size of the machine are important factors which are reviewed.

- Suction cups
- Bolted connection
- Hydraulic gripper
- Friction based grip
- Pins and holes
- Magnets

Suction cups

Industrial grippers often use suction cups to pick up flat item or sheets. By increasing the size and number of these cups this solution might be implemented on the Vibro-drill as well. Using Equation 4.1 from Giering et al. a surface area and required pressure can be estimated (2016). As the combined resultant force should equal $39MN$ the total area of the suction cups should be large enough and the pressure difference great enough to cope. Assuming four units with each a surface area of $4m^2$ completely covered in suction cups the pressure difference needs to be $3.8MPa$. Even under ideal circumstances this is not a practical solution due to the size and chances of damage and/or leakage and therefore this alternative will not be evaluated any further.

$$\Delta p = \frac{F_{res}}{A * \eta} * \left(SF_1 * cos(\beta) + \frac{SF_2 * sin(\beta)}{\mu} \right) \tag{4.1}$$

In which:

- Surface area vacuum cup $A = 16m^2$ (assumption)
- Efficiency vacuum $\eta = 1$ (assumption)
- Resultant force $F_{res} = 39.0MN$
- Security factor $SF_1 = SF_2 = 1$ (assumption)
- Angle to normal force $\beta = 90 degrees$
- friction coefficient $\mu = 0.64$ ([13])

Bolted connection

Bolts can be used to connect the Vibro-drill to the monopile. The force must be carried by all the bolt (or pins) in the vibration units, but also the material surrounding the holes should not yield. By calculating the amount of shear stress a single bolt and the boundary material can handle the minimum number of bolts can be determined. An important assumption here is being made concerning the vibrations: The bolt will not be undone by the vibrations. This assumption might be fulfilled by automatic fasteners or automated bolt wrenches. The latter of which is most likely to be used for removal anyway.

For the required number of bolts several diameters have been determined to prevent failure in either the bolt, the surrounding material of the monopile or the surrounding material of the Vibro-drill. For the bolts and Vibro-drill a yield strength of $800MPa$ is used and an average monopile is made of steel with a yield strength of $460MPa$. For the area of the bolt the cross-sectional area has been used and for the monopile and Vibro-drill the diameter of the bolt multiplied with the length of the section of the bolt.

$$A = \frac{F_{shear}}{\sigma_{yield}} \tag{4.2}$$

Table 4.1: Required number of bolts

Bolt diameter	**Bolt failure**	**Monopile failure**	**Vibro-drill failure**
M20	156	43	49
M30	69	29	33
M42	36	21	24
M52	23	17	19
M64	16	13	16

It is clear to see that the number of bolts is large, but by using large diameter bolts this problem can be solved. Bolting the Vibro-drill on the monopile can be done in the installation vessel, but removing the bolts after the installation of the monopile needs to be done in-situ tens of meters below the seabed. This is the most crucial phase and needs to work 100% of the time as the Vibro-drill can't be retrieved with one bolt failing. Due to practical difficulties in removing twenty bolts on a depth of forty meters below the seabed, the risk of the module not being able to be retrieved is too high. Therefore this alternative is not taken into account in further analysis.

Hydraulic gripper

As stated in Chapter 2 hydraulic grippers are widely used for vibro-hammers. As the product specification are openly available their capacity can be taken for this. With a listed clamping force of $3.5MN$ 12 hydraulic clamps would suffice (Dieseko-Group, 2014). As this alternative is already used in the industry and able to resist the force acting on it, it will be used in the MCA.

Friction based grip

When the Vibro-drill has extending arms, these arms can be used to create an enormous normal force on the monopile wall. If the normal force is large enough the friction keeps the Vibro-drill in place. The required normal force can be calculated using 4.3 with the friction force equal to $39MN$. As the steel on the bottom of the monopile is most likely to have an oxide layer on the surface the friction coefficient is taken as 0.27 [13]. This is a conservative choice as the friction coefficient for dry steel on steel is 0.78 [13]. As explained in Appendix B the vibrations of the eccentric weights do not influence the friction coefficient.

$$F_n = \frac{F_f}{\mu} \tag{4.3}$$

Table 4.2: Required normal force per arm [MN]

Number of arms	$\mu = 0.27$	$\mu = 0.78$
2	72	25
3	48	17
4	36	13
8	18	6.3

With both ends constraint but the beam not supported, the arms are sensitive to buckling. Euler buckling is an idealised situation in which the soil load is equal to zero and therefore a cylinder with a diameter of $200mm$ and a wall thickness of $50mm$ can carry the buckling stress (Equation 4.4). Here the yield is the limiting factor, but once again, by increasing the diameter this problem can be solved.

$$P_{crit} = \frac{4 * \pi^2 * E * I}{L^2} \tag{4.4}$$

Especially with the soil load acting on the beam the probability of the failure of one arm is rather high and with the loss of the Vibro-drill and monopile as a consequence a risk that is unacceptable. To prevent buckling due to soil loads supports in the vertical direction must be added. Decreasing the section of the beam that is free to buckle increases its capacity.

There is one issue that is inevitable though. By pushing a against the wall of the monopile with a large force will bend the monopile resulting in a loss of friction force. The contact area is limited the monopile will deform outwards at these point. The monopile also can't be deformed such that it doesn't stretch anymore because that would make it harder to drive it in the soil and would require an even larger normal force from the arms. For this reason this alternative is not reviewed in the MCA.

Pins and holes

As stated in Subsection 4.1.1 the removal of the bolts, when the monopile is installed and the Vibro-drill is deep under the seabed, is an uncertain process. Risking the Vibro-drill if one, or more, bolt(s) won't come out is unwanted, but can be avoided when pins are used instead of bolts. By inserting pins in dedicated holes in the monopile with hydraulic pistons, the same shear resistance can be achieved with a mechanically simpler system. As the forces on the Vibro-drill act in the plane of the monopile wall, the pin are able to carry them as long as there combined area is large enough. The same equation apply here as for the bolted connection. Therefore the same results as in Table 4.2 can be used.

Because the pistons in a hydraulic cylinder can be put under pressure as well when the pins need to be retracted, a large force can be applied to remove them. This makes a pins-and-holes system suitable for further analysis.

Magnets

Besides suction cups, also magnets are often used to temporarily attach things or pick up objects. When magnets of $1.5 Tesla$ are being used to for keeping the module in place on the monopile, it will pull with a force equal to approximately $800 kN/m^2$ (Magnetics, 2010). In Subsection 4.1.1 the required friction force was calculated to be between $50 MN$ and $144 MN$. This results in a surface area between and $63 m^2$ and $180 m^2$. This is unrealistic and therefore this alternative will not be evaluated any further.

4.1.2. Connection surface

The Vibro-drill needs to attach itself to the monopile with one of the connection mechanism described above. The connection surface however is very important as a connection may require adjustments in terms of holes, a certain shape, surface roughness or thickness. For this it might be necessary that an attachment is added to which the Vibro-drill connects. Four alternatives ranging from minimal to very large adjustments to the monopile are described below.

The very first alternative is the simplest one with the Vibro-drill attached directly on the monopile. The material might need roughening or a special coating, but the design of the monopile itself remains then same. This alternative also includes drilling holes for the bolts or pins to attach to, but it should be noted here that the yield strength of the material surrounding these holes should be checked before implementation.

When the connection mechanism requires a certain shape to fold around or some sort of attachment makes it easier to connect to, an additional piece of material can be welded to the monopile. The downside is that an extra step needs to be taken before the Vibro-drill can be mounted on the monopile, but the connection mechanism might become cheaper or quicker to install. Depending on the connection mechanism the attachment will be designed such that the surface is optimal. Again limits need to be put on the attachment in terms of shape and size as this attachment itself should also make a solid connection with the monopile. Attention should be paid to this when designing it.

When such an attachment is needed but be welded directly on the monopile an additional ring can be welded to the bottom of the monopile. This ring can then also include nozzles and pipelines for the jetting. Including this functionality and the attachment can make the design of the Vibro-drill easier, but makes the preparation of the monopile difficult. The extra material and fabrication costs are downsides and again a comparison is needed between the advantages and disadvantages.

Finally an attachment can be designed such that it will provide some structural rigidity to the Vibro-drill as well. By adding a bridge across the whole diameter of the monopile it can support the Vibro-drill when it is attached to it. It might even eliminate the need for an extending structure. The adjustments made to the monopile are massive however and a lot of work and time goes into a piece that will stay behind in the seabed. As adding the functionality of retrieval makes the Vibro-drill more economically viable this is something that needs to be looked at.

4.1.3. Structure type

Because of the difference in diameter the Vibro-drill needs to expand and contract before and after the installation of the monopile. This requires a dynamic structure to which the connection mechanism, the eccentric weights and other parts are attached. Because of internal loads it is preferred to have the eccentric weights near the monopile and thus the connection mechanism. Three different types of structures are discussed below and compared in the next chapter.

Using a set of hinged arms that can rotate up and down an extending structure can be formed. When the arms and the hinges are placed correctly a parallelogram can be formed to keep the connection mechanism and housing of the eccentric weights in the right orientation. The folding parallelogram can be controlled by an additional arm. When the Vibro-drill is mounted on the monopile the arms are fixed in place and a truss system is created to provide the necessary structural rigidity.

An alternative that doesn't require hinges is a central hub with arms that extend along their axis. By extending the arms the connection mechanism and eccentric weights are moved outwards towards the wall of the monopile. Bending and buckling are issues that must be investigated in the structural analysis and also the length of the arms in extended and contracted position should be calculated to fit the dimensions of the monopile.

The third and final structure type is based on the cross shaped connection surface. With a cross already in place the Vibro-drill can mount itself on there without first having to extend. The Vibro-drill can be made into one solid structure with the eccentric weights inside. The downside of the structure type is that the internal load caused by the eccentric weights need to be carried by the cross itself.

4.1.4. Number of arms

Theoretically the number of arms can be unlimited, but for the sake of stability the minimum is two. Above eight it also becomes difficult because the structure then becomes to complex. For this reason it is opted for four alternatives, namely: 2, 3, 4 and 8 arms.

4.1.5. Conclusion design check

For each of the components a number of alternatives has been selection. All of the alternatives have been checked on the design requirements, making them potential solutions for the final design of the Vibro-drill. These alternatives will be scored and compared in the MCA in Section 4.2.

For the connection mechanism the Hydraulic gripper and the Pin-and-hole system will be used as input for the MCA.

The connection surface can be directly on the monopile, but also attachments are being looked into further. A ring below the monopile, like a pile shoe, can have a specially designed contact point. A third alternative is to attach only such a contact point straight on the monopile. The last option is the most rigorous and consists of a diameter spanning cross in the monopile on which the Vibro-drill can attach itself.

The structure type needs to be adapted to fit with the connection mechanism and vice versa. For this there are three types of structure, namely a folding parallelogram, extending arms and cross shaped structure which fits on the cross described for the connection surface.

Then finally the number of arms needs to be determined with the minimum being 2, for stability, and the maximum being 8, due to complexity and 3 and 4 as intermediate alternatives.

4.2. Multi-Criteria Analysis

In order to evaluate potential concepts a MCA is performed on the all alternatives. The MCA scores the alternatives on five criteria which are defined such that they express all important characteristics of the concepts. The scores are quantitative and per concept to give the final score. Below the five criteria are explained after which the score per alternative is given. In the last section the two most suitable concepts are described.
From the design check the following alternatives will be input for the MCA:

Table 4.3: Morphological overview (mp = monopile)

Component	**Alternative 1**	**Alternative 2**	**Alternative 3**	**Alternative 4**
Connection mechanism	Hydraulic gripper	Pin-and-hole	-	-
Connection surface	Directly on the mp	Ring below mp	Attachment on mp	-
Structure type	Folding parallelogram	Extending arms	Fixed cross	-
Number of units	2	3	4	4+

4.2.1. Criteria MCA

To compare all potential connection mechanisms, they will be evaluated based on criteria. For the drivability of the monopile and Vibro-drill it is important that the soil resistance is minimal. As the tip resistance is directly related to the frontal area of the system as seen from below, it is crucial that this area should be small. A larger area would mean a larger loads on the structure, but also larger eccentric weight are required to drive the monopile in the soil. This criteria is therefore chosen for the MCA.
The reliability of the Vibro-drill should be high enough to install a wind farm worth of monopiles without needing large repairs or having a large risk of failure. The use of proven technology is an advantage. Due to the difficult environment with large vibrations, sea water and inaccessibility complex technologies might not be suitable and simple solutions are more desirable. Also redundancy is preferred for critical components.
On the installation vessel the Vibro-drill is lifted into the monopile before it is upended and after up-ending is it lowered into position to mount on the monopile. These two handling procedures limit the Vibro-drill in terms of size and weight.
Finally the costs of the Vibro-drill are important to make the product economically viable. The material and parts for the Vibro-drill itself should not be rare, expensive or hard to obtain, but more crucial are the costs of the parts left behind on the monopile. The Vibro-drill itself will be retrieved and re-used many times over, but the attachments to the monopile are lost after installation. This will drive up the cost of installation for every single monopile and is therefore an important factor.

The following design criteria are used in the MCA:

- Minimum frontal area as seen from below
- High level of redundancy/reliability
- Easy handling (in terms of size and weight)
- Low production cost

4.2.2. Scores

During this preliminary design stage no detailed designs and/or calculations are made. The grades will therefore be qualitative instead of quantitative based on estimates and assumptions. The scores for each of the alternatives are motivated below, but first the method for determining the scores is explained.

The frontal area as seen from below is an estimate based on rough dimensions. Because the eccentric weights have a large frontal area it is important to note that the connection mechanism will be underneath there. The result of this is that the frontal area of the connection mechanism is only important when it is larger the frontal area of the housing for the eccentric weights.
Reliability and redundancy are based on the amount of new and/or complex technology used in the design and the ability to build in redundancy for failure sensitive components.
Handling is based on size (when folded in) and a rough estimate for the weight. Multiplying rough dimension with the density of structural steel and weights of comparable product are used to make these estimates.

Finally the costs are difficult to estimate as the manufacturing requirements and detailed design aren't known yet. The cost of raw material, comparable products and estimates for manufacturing form the basis for these scores. Most important are the costs of the parts that are left behind after installation.

Connection mechanism
Hydraulic grippers have been used for vibrohammers before and therefore there are off-the-shelf product with proven functionality. The reliability and costs can be easily be given a good score, but the number of hydraulic grippers needed causes the frontal area to become quit large and the handling may be difficult as well.

For the pins-and-holes system can be designed such that they do not exceed the frontal area of the housing of the eccentric weights, but alternatively they can be integrated in the housing. Therefore this alternative scores good on the first criterion. As the number of pins can be increased the redundancy is possible and the reliability of a hydraulic piston is great again a high score is awarded. The only downside here is the risk of a pin getting stuck or deformed. In the first case the hydraulic pressure can be increased to free the pin and redundancy will allow for one pin to not work.
The handling will not be affected unless the system is excessively large or heavy and finally the costs can be kept low as the system itself is very basic and only holes are required in the connection surface.

Connection surface
Mounting the Vibro-drill directly on the monopile will not increase the frontal area besides the parts needed for jetting. These parts need to cover the monopile tip and no more is required. With no attachments only the monopile itself will have to take the loads and the surface, shape and material may not be ideal for this. The Vibro-drill will most likely become more complex as it has to adjust to the monopile shape instead of having an ideal surface to which it can connect.

Adding an attachment on the monopile to mount the Vibro-drill on does not necessarily have to increase the frontal area as this attachment can be small are placed underneath the Vibro-drill itself. It will also be build for the sole purpose of keeping the Vibro-drill in place which increases the reliability of the connection, but at the same time it needs to be connected to the monopile itself as well. This connection is most likely a weld and may still be a critical part. The handling of the Vibro-drill will not be affected a lot. Finally the cost may come down as the attachment must be small and could make the design of the Vibro-drill less complex.

A ring under the monopile, shaped such that the Vibro-drill can easily attach itself to it, might have a larger frontal area, but just like the attachment it will cover the monopile and parts of the Vibro-drill increasing the total tip resistance only slightly. As the connection surface can be integrated in the ring and reinforcements can be made the structural rigidity is most likely to be larger. The ring itself also has enough area to be properly welded to the monopile and therefore the reliability is a strong point.
The ring can take some of the loads on the Vibro-drill reducing the internal loads in the Vibro-drill as well as the monopile and the jetting and fluidization holes can be placed in the ring. The Vibro-drill might lose some weight due to this. This last argument also causes for the cost of the Vibro-drill to decrease, but losing one ring on every installed monopile diminishes this benefit.

Finally the fixed cross will increase the frontal area as this large surface is added to the tip. The structural rigidity that this cross will bring is a great benefit though. Similar to the ring under the monopile, this cross can reduce the internal loads in the Vibro-drill resulting in a lighter and smaller Vibro-drill. The cost of such an object will be large and this a the major disadvantage of this alternative.

structure type
The parallelogram structure has a central hub and arms going from this center to the monopile wall. Because the parallelogram itself is vertical, the frontal area as seen from below is not that large. The hinge on the folding arms in combination with the large vibrations cause the reliability to be less though. The resistance of the arms and hinges against sideways forces is mediocre and this results in a low score for reliability.
The folding arms also cause a downside when folded in. Because the arms need to be long enough to reach the monopile wall they will extend far when the Vibro-drill is folded in and needs to be handled. The costs of this relatively simple system is not that high. Not a lot of material is needed.

The extending arms will have a larger frontal area than the folding arms, as the housing must be wider. This does help improve the structural rigidity and the reliability is better because all moving parts can be shielded from water and sand. The compact design will be easier to handle and this alternative is technically simple. On both costs and handling the extending arms score well.

The fixed cross has a huge benefit due to the fact that it is completely covered by the structure it attaches itself to. The design of the Vibro-drill can therefore be made smaller, simpler and lighter making it score high on reliability and handling. Here again the costs are higher due to the cross being left behind on the monopile.

Number of arms
The number of arms on the Vibro-drill is a compromise between the load per arm, the redundancy with extra arm, the number of components that can fail and the complexity of the system. The frontal area of each arm is almost equal for all alternatives and thus a small number of arms is beneficial. The advantages of having multiple arms is that the other arms can carry the load if one fails. For both handling and costs less arms score better because of simplicity of the design.

Table 4.4: Scores Multi-Criteria Analysis

Alternative	Frontal Area	Reliability/redundancy	Handling	Costs
Connection mechanism				
Hydraulic gripper	-	+	+-	+
Pin-and-hole system	+	+-	+	-
Connection surface				
Directly on the monopile	+	-	-	+
Attachment on the monopile	+-	+-	+-	+
Ring below the monopile	+-	+	+	+-
Fixed cross in monopile*	-	+	+	- -
Structure type				
Folding parallelogram	+-	-	-	+
Extending arms	-	+	+	+
Fixed cross*	+	+	+	- -
Number of units				
2	++	- -	+	+
3	+	-	+-	+-
4	-	+-	+-	+-
4+	- -	++	-	-

*With a cross fixed in the monopile the only structure type that fits is another cross on top of it and thus no other combination can be made.

4.2.3. Major issues

For certain combinations major issues can be identified. These are problems which can not be solved within reasonable measures. For example using components an order of magnitude larger than currently available will lead to very high production costs and/or time. As a result some alternatives are rejected for potential concepts further along the project. Other issues are marked as risks to be looked into in the detailed design. The issues which have been identified in this early stage already are summed up below.

- A fixed cross in the monopile takes a lot of material and with that a lot of costs for component that are left behind. Also, it is not possible to positions the eccentric weights parallel to the monopile for torsional vibrations. As this is required feature, this alternative is rejected.

- More than four arms will make the structure complicated and large. Besides this keeping the eccentric weights in phase in all of the different housings becomes more difficult as well. To prevent the frontal area on the bottom of the monopile to be open and leave room for larger objects like rocks to pass though it is useful to keep the number of arms low. Therefore four is the largest number of arms considered in the concept selection.

- Three or four blocks with eccentric weights is sufficient for providing the eccentric force. Having only two blocks would mean that both of them would become very large and the forces acting on the monopile are not divided properly. It should be checked whether the Vibro-drill and monopile can withstand these loads.

- For the folding parallelogram it is difficult to connect the eccentric weights with each other as the arms (through which the axles run) fold. The angle that the joints need to make in combination with the high RPM and high torque require specialized industrial equipment. The size of these components makes them difficult to fit in the design and may limit the functionality.

- Also the arms between the torpedo and unit need to be able to resist high loads in order to keep the alignment of the connections in place. With the long thin arms the risk of sideways loads bending and/or buckling them is serious and needs to be addressed accordingly.

4.3. Conclusion preliminary design

From many potential solutions two combinations of alternatives are selected for further investigation. The first concept is a compact and sturdy design with three extending arms and hydraulic grippers which attach to a ring below monopile. This concept is reliable and robust, but also heavy. This concept will be referred to as the Vibro-Boxer. The second option is lighter folding structure. Again three arms will attach to a ring below the monopile but this time with a pins-and-holes system. This concept is named the Vibro-Polyp.

The Vibro-Boxer and the Vibro-Polyp are the best-scoring concepts and are therefore chosen to be worked out in more detail. A more detailed model and a structural analysis will provide profound arguments for which of these two designs is the most suitable. The design requirements will be kept as a benchmark for this comparison. First Chapter 5 will elaborate on the geometry and dimension a simple design for both of the concepts.

II

Concept selection

5

Concept Design

From the preliminary design stage two concepts emerged. Both of these concepts will be worked out into simple designs in this chapter. The designs will include all of the general geometry and initial dimensions for the structural components. In Chapter 6 these components will be checked for the load cases as described in the design requirements.
First some general design choices are explained as they are relevant for both concepts. Afterwards, the Vibro-Boxer is discussed and its geometry is explained. Finally the same is done for the Vibro-Polyp.

5.1. General design choices

For both of the concepts the housing for the eccentric weights, the pile shoe and the umbilical are the same. Therefore these will be presented first and some characteristics, which are needed to determine the geometry, are given.

5.1.1. Housing eccentric weights

The housing for the eccentric weights is assumed to be identical for both concepts. The number of arms is equal and with therefore also the size and number of housings for the eccentric weights are the same. For the geometry of the Vibro-drill it is important that the machine will fit through the top of the monopile, which has a diameter of $5.5m$. At the same time is also needs to attach to the pile shoe, which has a diameter of $8.5m$. The housing of the eccentric weights is located close to the connection for structural rigidity and the most effective path for vibrations. The length and width of the housing should therefore be small as these affect the outer diameter the most.
The dimensions of the housing need to be a minimum of $LxWxH = 2100x630x2300mm$ in order to fit the eccentric weights. These dimensions will fit three rows of three weights. The internal layout of the housing will be determined in the detailed design, but for now the dimension and the total weight are important for geometry and structural analysis. Based on the eccentric weights, axis and raw material that is needed, an estimate of $6000kg$ is made for the total weight of the housing. Figure 5.1 shows the green box used in the simple design to obtain the required geometry.

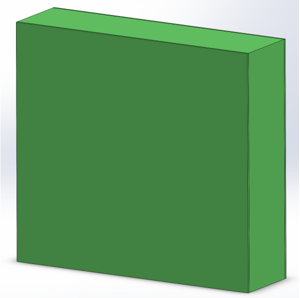
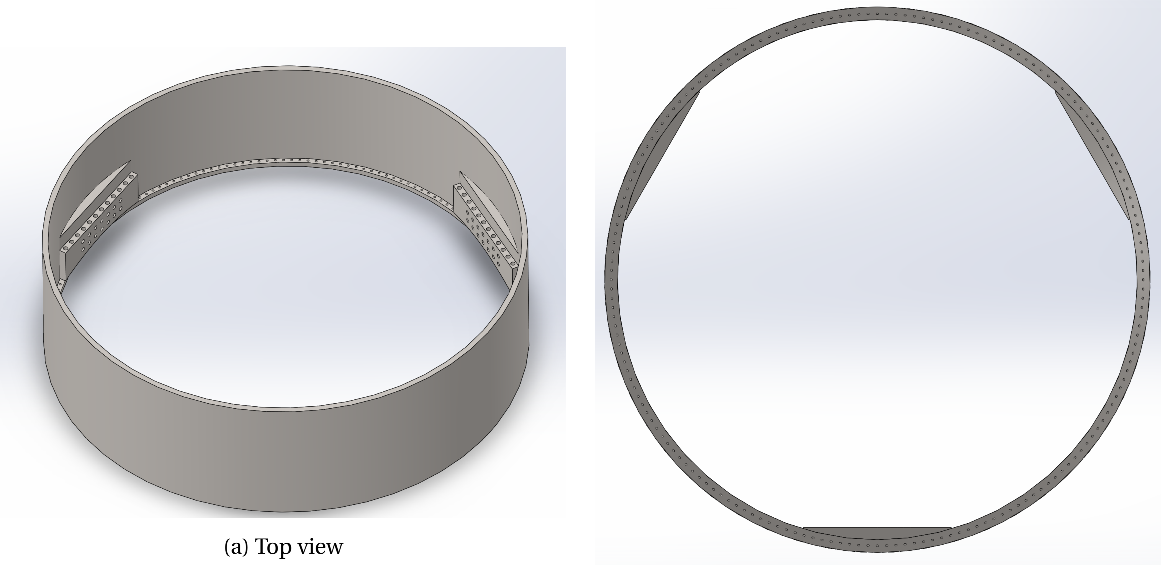


Figure 5.1: Housing for the eccentric weight

5.1.2. Pile shoe

The ring below the monopile will be referred to as pile shoe during the remainder of this report. The pile shoe is welded underneath the monopile and has a height of $2500mm$. It has the same diameter and wall thickness as the bottom of the monopile, respectively $8.5m$ and $100mm$. On the inside of the pile shoe there is a ring with pipes and holes for jetting and fluidization which can be seen clearly in figure 5.3. The attachments for the connection with the Vibro-drill will be fitted above this ring. In Figures 5.2 and 5.3 the attachment for the Vibro-Polyp is placed on the pile shoe.


(a) Top view

(b) Bottom view

Figure 5.2: Pile shoe with attachment for the Vibro-Polyp

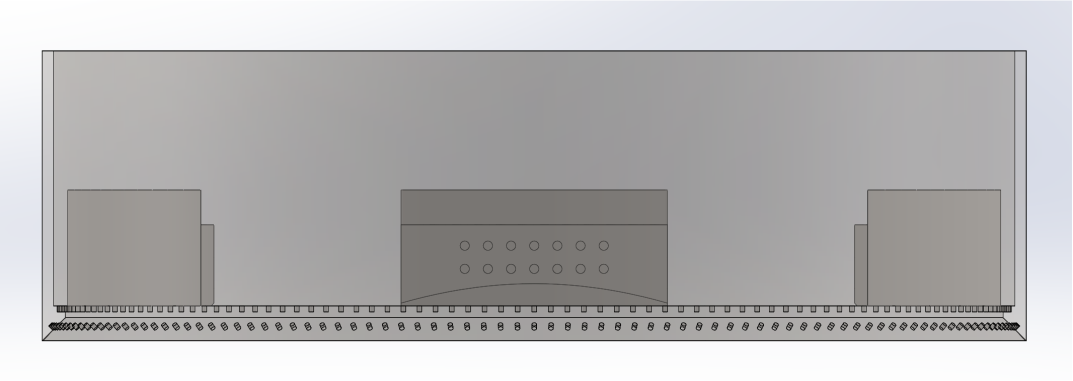


Figure 5.3: Pile shoe with attachment for the Vibro-Polyp side view

5.1.3. Umbilical

The eccentric weights are driven by hydraulic motors and also other components require pressurized hydraulic fluid. For jetting and fluidization water is needed and finally some cables for monitoring the system run down from the installation vessel to the Vibro-drill. In order to protect all of these lines, an umbilical is used to keep all of them together. This umbilical is connected to the top of the Vibro-drill where all lines meet. Because the requirements for the hoses and cables are not known yet, it is assumed that it will be a single line attached to the top of the central hub.

5.2. Vibro-Boxer

The Vibro-Boxer has three extending arms which allow it to mount onto the pile shoe. In Figure 5.4 the Vibro-Boxer is depicted in contracted position. The inner part of the extending arms (orange) is inside the outer part (blue). The hydraulic gripper can be found right underneath the housing for the eccentric weights. As one of the arms is transparent, the internal structure can be seen. Two beams run the full length of the arms and extend beyond the housing. In Figure 5.5, these arms form the connection with the inner part of the extending arm (orange). Four large sprockets along those beams are turned by motors and move the inner part of the extending arms outwards. When the arm is extended four clamps will fix the position of the beams relative to the inner part of the arms.

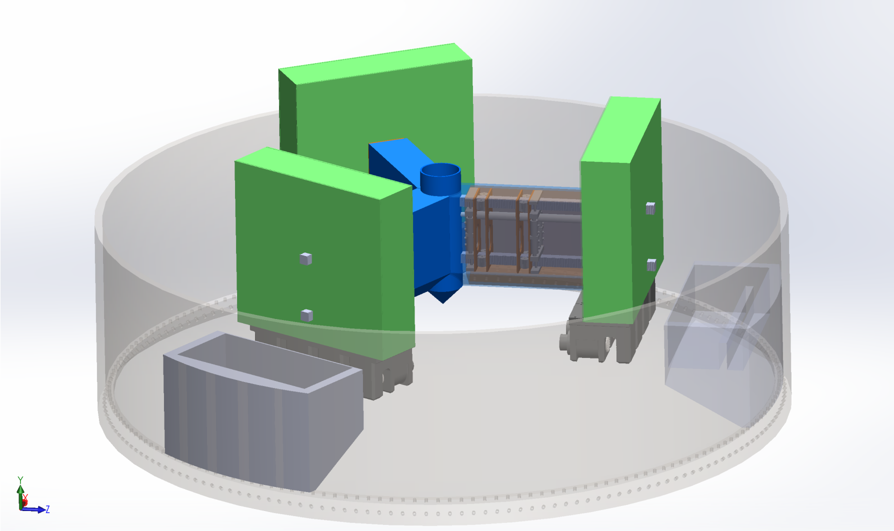


Figure 5.4: Top view of the Vibro-Boxer with pile shoe (contracted)

Once the Vibro-Boxer is in place the two beams are connecting the central hub and the inner part of the extending arms (orange). The outer part is only protecting the beams from impacts from rocks and keeps the seawater and soil outside. The soil resistance acts on this section, but the structural rigidity is provided by the beams. The connection between these beams and the central hub, the connection between these beams and the inner part of the extending arm and the connection between the inner part and the housing of the eccentric weights are critical areas and need to be investigated in the structural analysis.

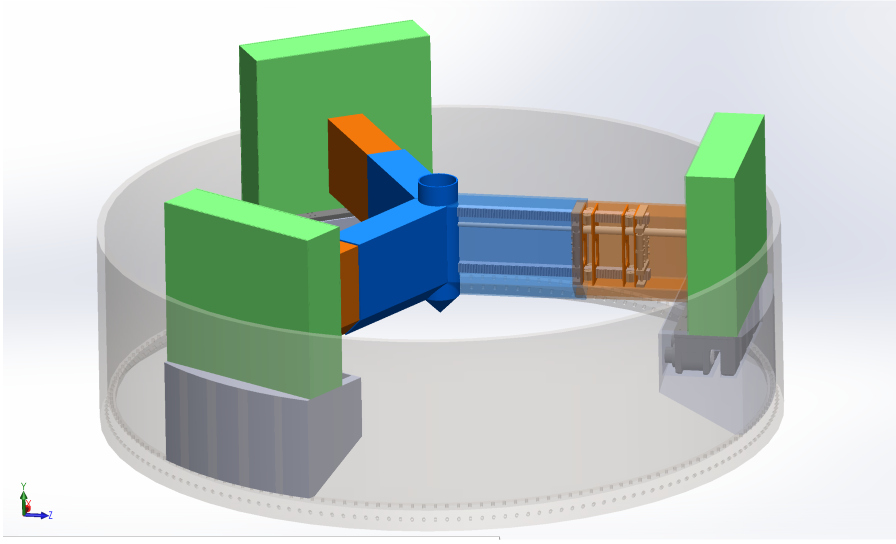


Figure 5.5: Top view of the Vibro-Boxer with pile shoe (extended)

The attachment on the pile shoe is meant to surround and protect the hydraulic grippers. The large box shaped attachments take the soil resistance and protect the grippers from impact from rocks and the flow of seawater and soil. In Figures 5.6 and 5.7 this function is clearly visible as the hydraulic grippers can not be seen from the bottom.

Inside these boxes reinforced plates are made for the Vibro-Boxer to grab onto. As these attachments are designed specifically for this type the gripper, the material and surface can be optimized for grip. The bottom of the attachment will be covered with holes for jetting as well just like the bottom of the extending arms.

5.2.1. Geometry Vibro-Boxer

The range of the extending arms should allow for the Vibro-Boxer to contract such that the outside corner of the housing is within a $5.5m$ diameter, but can also reach the wall of the monopile when extended. The beams and inner part have an overlapping length of $750mm$. On this section the four clamps and the four sprockets will connect the two parts of the extending arm. This part of the inner arm is reinforced with partitions, as can be seen in Figure 5.5, to cope with the loads acting upon it.

The width of the arms should allow room for not only the beams, but also the axle for phasing the eccentric weights in the different housings and the hydraulic and water lines. The downside of a greater width is a higher soil resistance and thus a larger eccentric force needed to overcome this soil resistance. A compromise has therefore been made at $W = 500mm$.

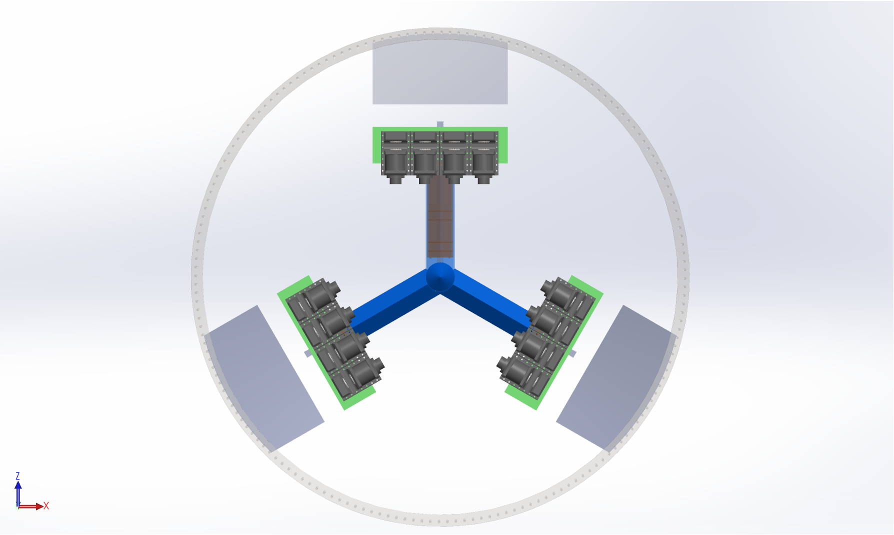


Figure 5.6: Bottom view of the Vibro-Boxer with pile shoe (contracted)

The height between the beams determines its resistance against bending in the vertical plane. As this is the most important load due to the soil resistance, buoyancy, the weight of the Vibro-drill and axial vibrations, this height should be sufficient. For an initial estimate the second load case, extending the Vibro-Boxers arms, has been used as this creates the largest moment on these beams. A height of the $378mm$ allows the two beams to carry the load and still provides room for the axle for phasing to fit in between. The total height of the arm is $1250mm$, but in the detailed design additional plates may be added to the bottom to prevent damage after rock impacts.

Because the hydraulic grippers are larger, they will extend outside the housing for the eccentric weights when viewed from the bottom. This causes additional soil resistance. By aligning the gripper vertically, such that they grab onto a horizontal plate, this problem would be eliminated. Then the loads acting on the connection would be directed along the pistons of the hydraulic gripper. This results in the pistons taking the load and affecting the pressure in the hydraulic lines. Besides this, movement of the pistons would dampen the vibration that should pass from the Vibro-drill to the monopile. This option is therefore neglected.

The length of the attachment is the same as for the housing: $L = 2300mm$ and the width is $1050mm$ on the sides and $1212mm$ in the center due to the curvature of the monopile. The height at the monopile wall is $1291mm$ and the surface is inclined to help the flow of water and soil particles.

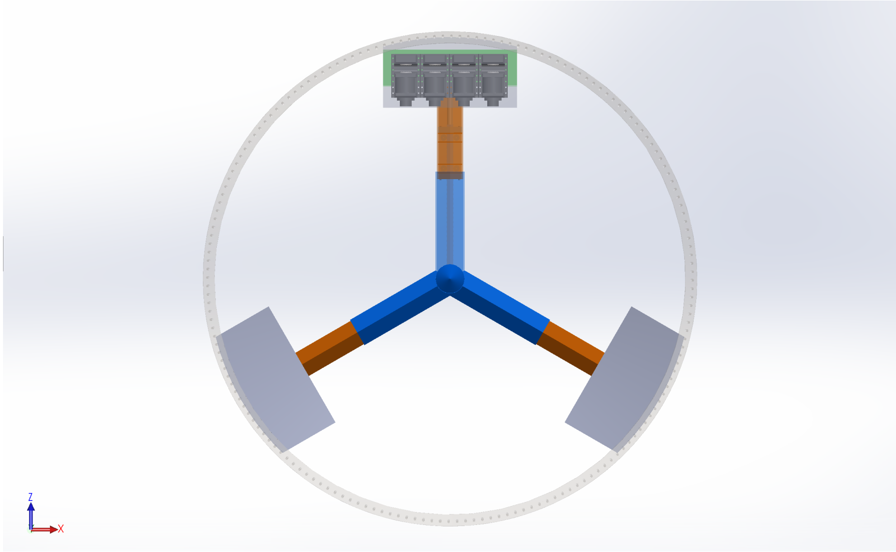


Figure 5.7: Bottom view of the Vibro-Boxer with pile shoe (extended)

5.3. Vibro-Polyp

The Vibro-Polyp also has three sets of arms, three housings and three connections with the pile shoe. Instead of extending arms the Vibro-Polyp has a parallelogram and an additional arm with which the central hub is connected to the housings. The parallelogram has two parallel arms of equal length which keep the housing for the eccentric weights and the connection mechanism in the correct orientation. When it is folded, as depicted in Figure 5.8, the angle between the arms and horizontal is large. When the Vibro-Polyp needs to extend, the ring on the top of the central hub is pushed upwards by three pistons. This raises the ring and pulls up the top arms. The middle and bottom arms folds upwards and outwards due to their geometrical layout. The connection mechanism is pushed out against the attachment on the pile shoe and the connection is made. Figure 5.9 shows the Vibro-polyp in extended position.

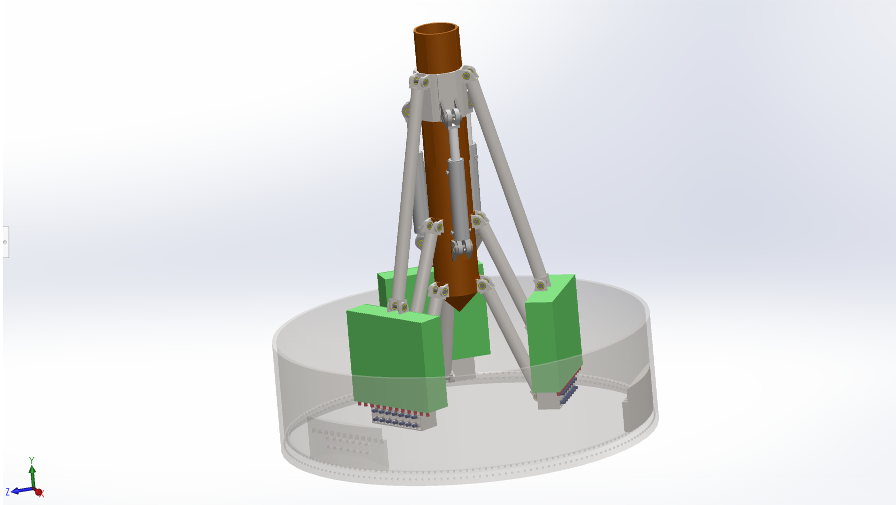


Figure 5.8: Top view of the Vibro-Polyp with pile shoe (contracted)

The angles of the arms have been calculated such that the Vibro-Polyp fits through the top of the monopile and it can connect to the attachment on the pile shoe. At the same time, the length of the arms should remain small in order to keep the length of the Vibro-Polyp small in contracted position. This improves the handling of the machine.

The ring on top of the central hub is fixed when the Vibro-Polyp is extended. This prevent the arms from folding in when the soil resistance and buoyancy push the central hub upwards during operations. This is

needed as the arms of the Vibro-Polyp are directed downwards when seen from the central hub. The upwards loads on the hub pulls the arms away from the monopile wall. Horizontal pins on the bottom housing for the eccentric weights and the fixture on the ring on the top of the hub prevent the arms from contracting.
The arms themselves consist of hinged cylinders and are free to move within the limits of the geometry. The middle arm also houses the axle which keeps the eccentric weight in phase and the hydraulic and water lines run through the bottom arm. The downside of the axle running through an arm is that it needs a double joint to cope with the angle under which it has to operate. Industrial joints will be used, but these are integrated in the detailed design.

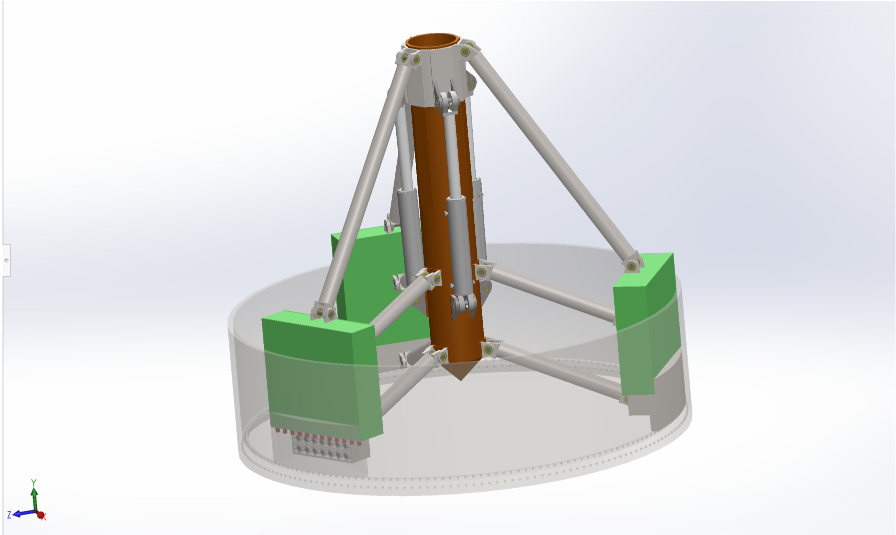


Figure 5.9: Top view of the Vibro-Polyp with pile shoe (extended)

The housing for the pins-and-holes system is located underneath the housing for the eccentric weights for a simplicity. The attachment on the pile shoe is therefore a slim box with holes and connections for the water lines in it. In a further design process the pins can also be distribution along the housing for the eccentric weights to spread the load. The downside of this is that a larger attachment on the pile shoe is needed and the holes will be likely to penetrate the pile shoe making the fabrication more difficult and reducing the structure rigidity of the pile shoe.
In Chapter 4 the number of pins was calculated, but for this basic design fourteen pins are used for the in-plane forces (along the pile shoe wall) and twelve pins for the out-of-plane forces.

5.3.1. Geometry Vibro-Polyp

For the geometry of the Vibro-Polyp the angles between the arms and the horizon are important for the loads acting on the arms and hinges. For the Vibro-Polyp the largest force is expected in the top arm during second load case while extending. The larger the angle Alpha between the top arm and the horizon the more force is required to pull the arm up as is need to move outwards first. At the same time this angle is limited by the geometrical requirements for the size in contracted position. A large angle of Alpha makes the Vibro-Boxer too large to fit through the top of the monopile.
Another limitation is the angle Beta between the bottom or middle arm and the horizon. This angle should also be as large as possible to minimize the load while folding outwards, but at the same time can't be near zero in extended position as this prevents the arms from carrying vertical loads during operations. Table 5.1 show the angles Alpha and Beta in contracted and extended position.

Table 5.1: Angles of arms on Vibro-Polyp

Angle	Contracted	Extended	Unit
Alpha	75	56	Degrees
Beta	68	21	Degrees

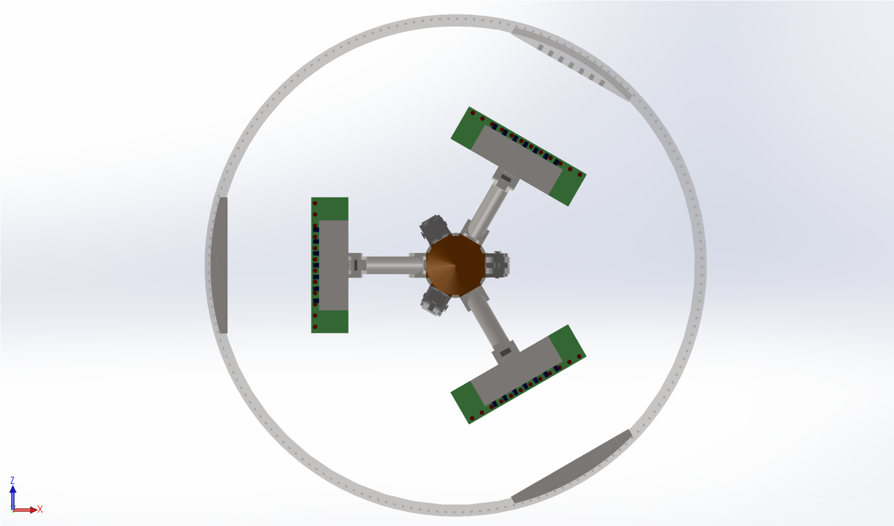


Figure 5.10: Bottom view of the Vibro-Polyp with pile shoe (contracted)

The lengths of the bottom two arms from hinge to hinge is 2464mm and the top arm measures 4736mm. The outer diameter is 300mm, but might be increased if the structural analysis requires to do so. The cylinders are hollow and allow for the axle and hydraulic and water lines to run through them.

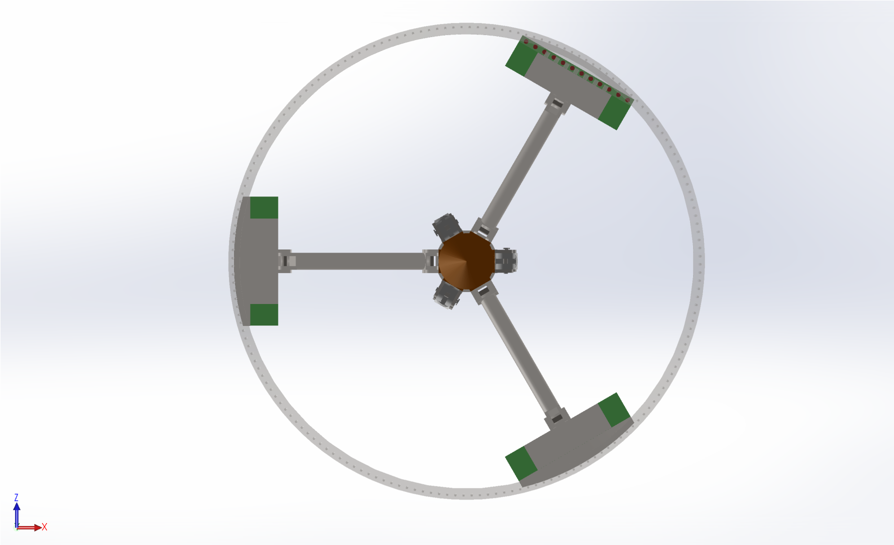


Figure 5.11: Bottom view of the Vibro-Polyp with pile shoe (extended)

6

Structural Analysis

In the structural analysis, the structures of both Vibro-drill concepts are reviewed and checked on structural capacity relative to the demand on component level. To do this, a simplification of the structure has been made to which quasi-static force is applied. During operations the loads on the structure will cause dynamic behaviour, but time constraints do not allow for this to be included in this thesis. The consequence of this simplification will be discussed in Chapter 9.
First an overview of the structural analysis is given and requirements for the input are discussed. Then the loads acting on the structure will be displayed. The way they are modelled and result in internal loads will be explained. When the loads are calculated the structural checks are performed and finally the results will be presented. In Chapter 5 some non-structural components were identified as critical. These will be mentioned in Section 6.6. A conclusion will summarise the important findings before a comparison is made between the two concept in the next chapter.

6.1. Analysis build up

For the structural analysis a Monte Carlo simulation is used to calculate the loads on the system. As discussed in Chapter 3 the soil resistance is calculated using a sample taken from a distribution. The unit soil resistance is determined 10,000 times for the simulation. The number of samples is a compromise between accuracy and computation time. For every sample the load analysis and structural checks are performed. In Section 3.2 will be explained how these external loads are turned into internal loads using the geometry of the structure. When the internal loads are known the geometry of the components are used as parameters for the failure mode checks. Dimensions such as the wall thickness, width or height can be increased when a component fails a check.

Figure 6.1 shows the full analysis. Using Matlab, a tool is made in which the parameters can be adjusted and the structural analysis is automatically performed. The benefit of this tool is that it allows the user to improve the structural rigidity of the Vibro-drill to such extend that the probability of failure is less than 0.005 (0.5%). Especially because the requirements for the Vibro-drill may change when test results are available.
The costs of losing a monopile is immense and therefore the probability of failure has to be low. The probability of 0.005, as stated above, is for the installation of one wind farm with approximately 100 wind turbines. If the structural analysis tool is correct on the probability of failure it is still possible other failure modes such as leakages or broken bearings can hinder the installation of the monopile. It should therefore be noted that the overall probability of failure for the Vibro-drill is higher than calculated for just the structural analysis. As the sum of the probability of all failure modes should very low, it is, arbitrarily, determined that for the structural analysis a probability of 0.005 is a conservative but reasonable value.

Besides for this structural analysis, the tool can also be used for optimization of the Vibro-drill to save material and decrease the costs and weight. By reducing the dimension of component which have a large capacity left the design of the Vibro-drill can be improved. Due to limited time an optimization has not been performed in this project.

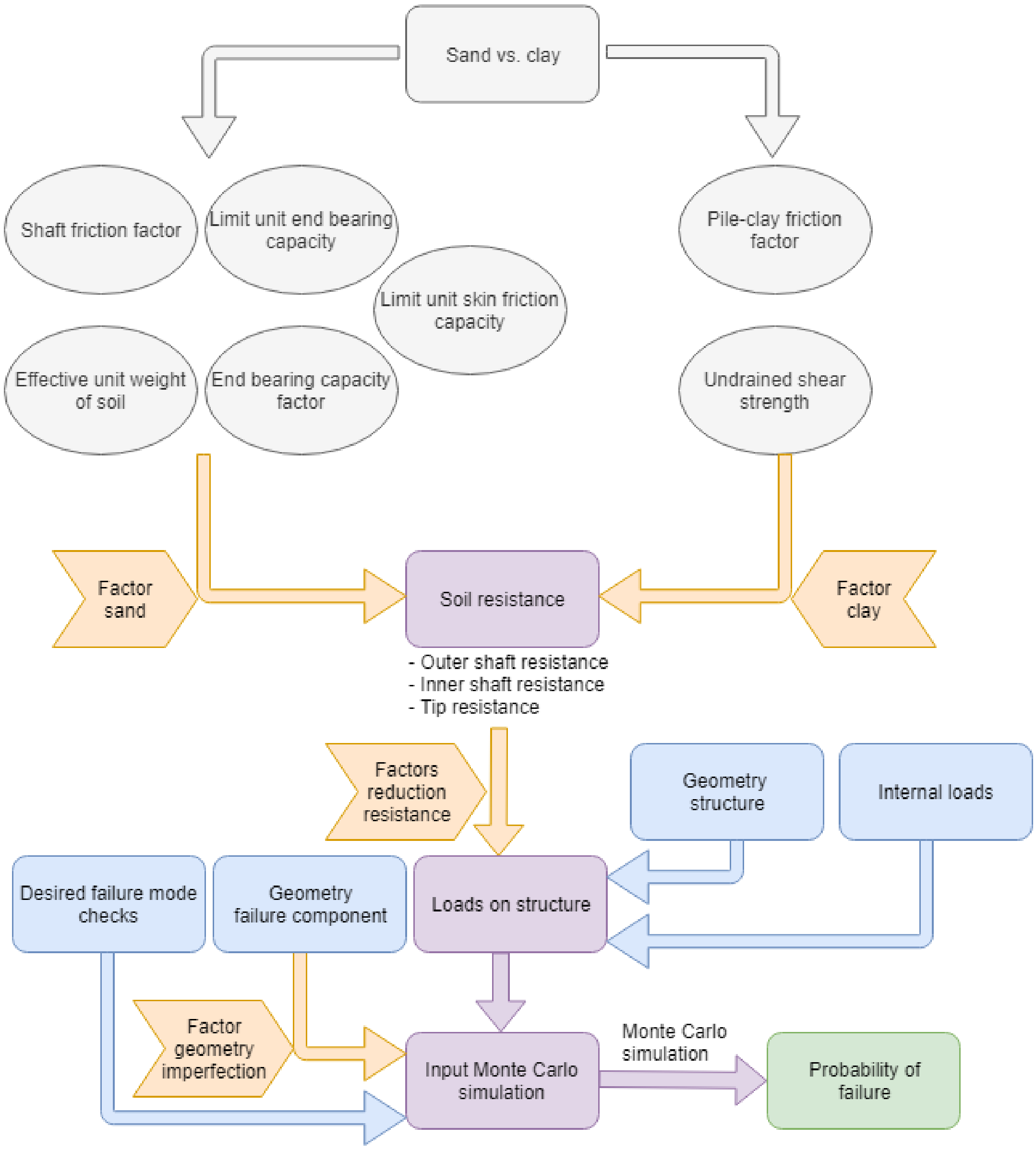


Figure 6.1: Schematic drawing structural analysis

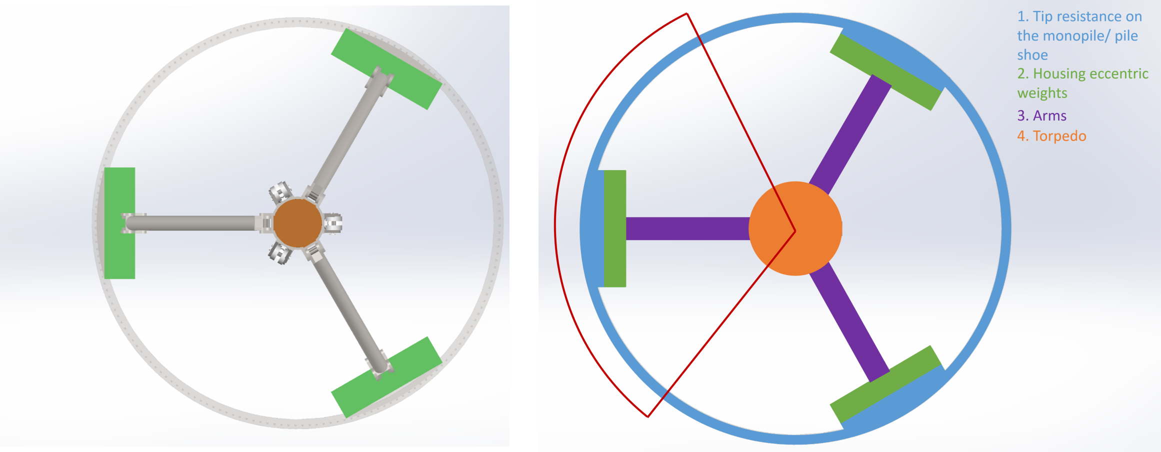
6.2. Loads

The loads acting on the Vibro-drill are described in Section 3.2.1. As the loads and structures of both concepts are modelled some assumptions had to be made. First, the assumptions are discussed which are valid for both concepts and afterwards the concept specific assumption are explained. Afterwards the external loads are depicted on the structure and the internal loads are extracted. This is also where the components for the failure mode checks are listed. The loads for the last load case are discussed separately for each concept. Note that only the second and third load case from Section 3.2 are used for this analysis.
For the structural analysis the following assumptions were made:

- For the structural analysis an important simplification has been made. By modelling the loads as quasi-static even though the system is dynamic. Inertia is therefore not included in the internal loads. The consequences of this is that stresses will be underestimated when dynamic amplification and resonance occurs in the system. To compensate for this, a Dynamic Amplification Factor (DAF) is used during the structural checks. This factor γ_{DAF} is set a 1.3 as this value is prescribed by the International Organization for Standardization (ISO) for Offshore structures (DIN, 2007). This assumption and its consequences for the reliability of this structural analysis is discussed in Chapter 9.

- $F_{eccentric}$ is a sinusoidal force, but it is simplified such that the absolute maximum force is taken from the situation with $F_{eccentric}$ is positive and $F_{eccentric}$ is negative. For the fatigue check the strain difference between these two situation is used in the calculation of the fatigue damage.
- In an ideal situation the resultant force in the system is used to accelerate the monopile and vibrate it to cause liquefaction.
- The structural analysis will be performed on a section of the system as both concepts are symmetrical along the longitudinal axis. As depicted in Figure 6.2b one third of the Vibro-drill and monopile is modelled and the results of this section apply to all three.

As stated above only a third of the total system will be modelled. In Figure 6.2b the red line marks the area with the arm running in the middle of it. For the loads it is important that the weight and buoyancy of the monopile, pile shoe and torpedo are divided equally among these sections. Furthermore the soil resistance on these parts is split. The rest of the soil resistance is divided on the components as their projected area is multiplied with the calculated unit soil resistance. By modelling only one third of the system calculations can be simplified and run time can be decreased.



(a) Frontal area

(b) Frontal area divided per region

Figure 6.2: Area on which the soil resistance acts

In the structural analysis two load cases are used. The first to be tested is the extension of the Vibro-drill while hanging inside the monopile. The central hub (or torpedo) is connected to the top of the monopile while the arms are extending until the connection mechanism is in place and the Vibro-drill can connect itself to the pile shoe. During this operation the eccentric weight are not turning and there is no soil resistance. The only force acting on the structure is its own weight. The gravitational force leads to internal forces and the resulting loads in the arms and hinges are calculated. The process of extending the arms is slow and controlled, it is therefore assumed to be a static system.

For the second load case the system is described as a quasi-static system. The external loads are static, but the eccentric force, which is sinusoidal in time, change direction. During a cycle all internal loads are calculated and their peaks correspond with the maximum and minimum of the eccentric force. Of these peaks the absolute maximum are used as the maximum load on the component. Besides this the minimum and maximum load on the components are further used to determine the strain range for the for the fatigue check. The difference between the minimum and maximum is calculated for every component separately and the number of cycles is equal to the installation time for a complete wind farm of monopiles. This is elaborated on in Section 6.3.
The loads acting on the systems are discussed separately for the concepts. For the Vibro-Boxer the following assumption has been made in order to model the structure.

- The connection between the two beams and the inner part of the extending arm (orange) is modelled as a fixed hinge. It is assumed in can resist the moment acting upon it. This assumption is to be checked in Chapter 8.

As can be seen in Figure 6.3 the soil resistance acts on the torpedo (blue), inner (orange) and outer (blue) parts of the extending arms, the pile shoe (including attachment) and the monopile. Also the weight and buoyancy act on these components as well as the housing for the eccentric weights. Because the housing is shield by the attachment on the pile shoe, the soil resistance does not act on its bottom surface. The eccentric weight inside the housing are the source of the final load acting on the system.

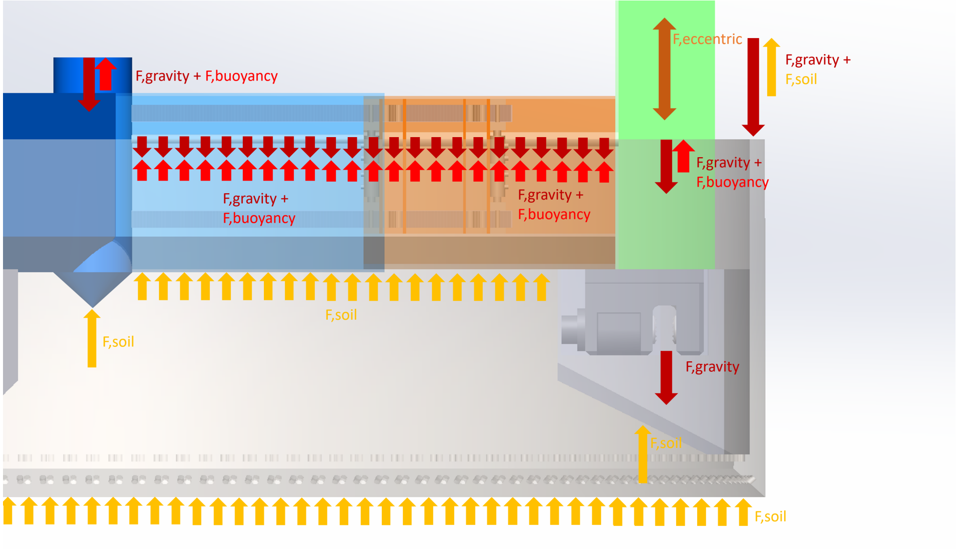


Figure 6.3: External loads on the Vibro-Boxer during operations

The structure of the Vibro-Boxers arms is modelled as two beams with a hinge in between. The left beam is constructed of two actual beams while the outer part of the extending arm does not carry any of the internal loads. The two beams are clamped and form a hinge at the end of the inner part of the extending arms. This part is also modelled as a beam. Due to the reinforced end with the clamped beams a rectangular structure is created. This resists the moment acting on the hinge and prevents the arm from bending.

The inner part of the extending arm is connected to the housing and the housing sits on top of the hydraulic grippers which pass the loads on to the pile shoe and monopile.

During the structural analysis the beams on the left of the hinge are checked both separately as well as combined. As both beams together resist bending, but both experience shear and tension/compression individually. Besides this the outer and inner parts of the extending arm are also checked.

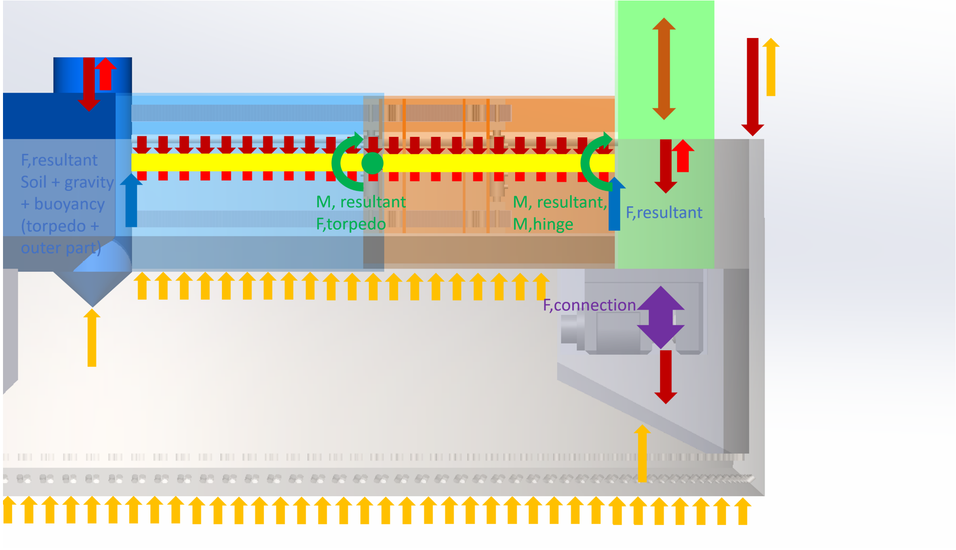


Figure 6.4: Internal loads on the Vibro-Boxer during operations

The Vibro-Polyps structure consists of a torpedo, three arms connected with hinges and the housing with the connection mechanism underneath it. When the Vibro-Polyp is mounted on the pile shoe and the guidance ring on the top of the torpedo is fixed in place, the structure can be modelled as a fixed truss system. The angles and lengths of the arms are known and therefore the force acting in the system can be calculated using trigonometry.
Before the loads are determined an important assumption has been made concerning the soil resistance.

- The soil resistance acts on all of the three arms of the parallelogram. Even though the bottom arm has to distort the soil it is not reduced to zero for the two arms above it. Due to the soil being fluidized it will partially move back to its original position. The movement of the soil will weaken it so therefore the soil resistance on the middle and top arms is assumed to be less than on the bottom arm. Because an exact value cannot be calculated, resistance on the middle and top arms assumed to be equal to 70% of the soil resistance on the bottom arm.

The soil resistance acts on all main components of the Vibro-Polyp with a lower unit soil resistance for the middle and top arm. Also the gravity and buoyancy acts on all components and finally the eccentric force is inside the housing for the eccentric weights.
The arms are all in tension as the torpedo is being pushed up by the soil and buoyancy. The resulting forces on the hinges have a horizontal component as well and on the torpedo these internal loads are cancelled out by the other two arms, but on the connection the pins-and-holes system transfers this load to the pile shoe.
Due to the geometry of the arms the top arm will carry the highest load from the torpedo. These loads are divided based on the angle of the arm.

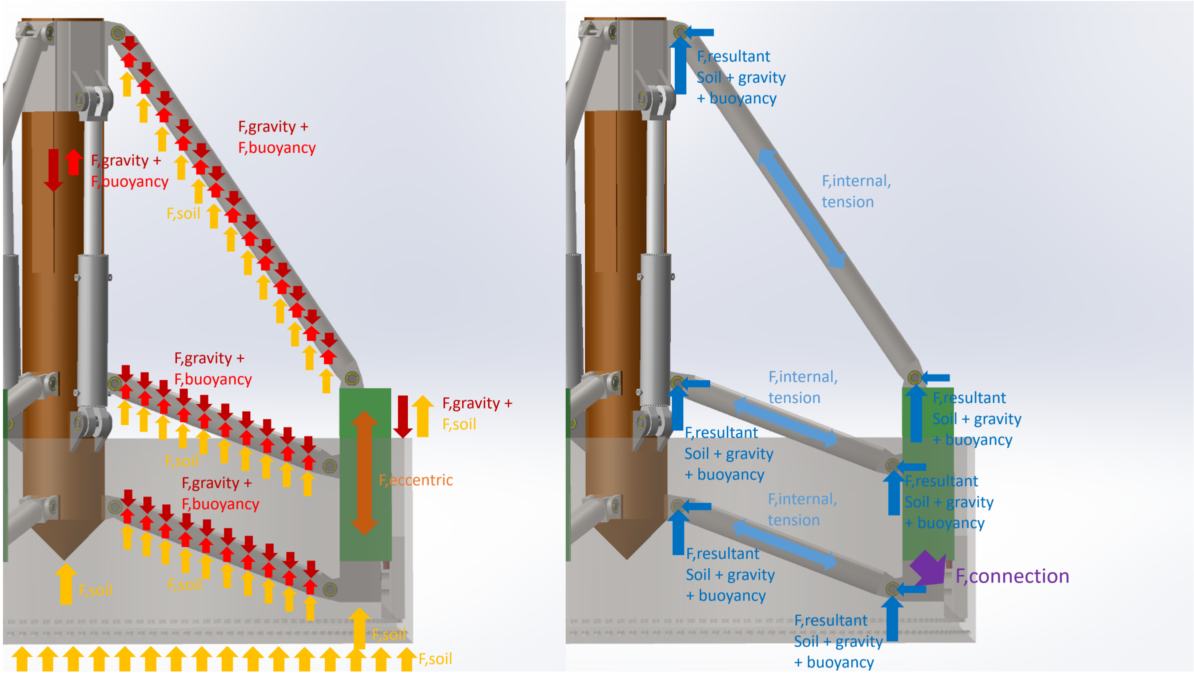


Figure 6.5: External and internal forces on the Vibro-Polyp during operations

For the second load case when the piston on the torpedo has to push the guidance ring up in order to lift the housing with the connection system, the rod of the piston needs to be checked. Besides this the three arms and all their hinges are modelled as well as the pins in the connection mechanism. The hinges, the pin of the hinge with the greatest total force acting on it is checked as the other pins will have lower stresses. For the connection mechanism the pins which take the horizon load and the pins that take the vertical load are checked separately as they only carry load in one of those two directions.

6.3. Structural checks

After the stresses on the components are calculated, the failure modes need to be determined before the structural checks can be performed. The components that need to be checked and the failure modes they are checked for, are displayed in Tables 6.1 and 6.2. The limit states for the checks are discussed afterwards.

For the Vibro-Boxer shear is only checked for the third load case, as those stresses are higher. Fatigue is neglected as the process of extending the arms happens only one time per installation. During operations the number of cycles is multiple magnitudes larger and therefore all components are checked for fatigue in the third load case.

Table 6.1: Structural checks performed on Vibro-Boxer

Component	Bending	Buckling	Shear	Tension/compression	Fatigue
Second load case					
Bottom beam		X			
Top beam				X	
Combined beams	X				
Outer part					
Inner part	X				
Third load case					
Bottom beam			X	X	X
Top beam		X	X	X	X
Combined beams	X				X
Outer part			X		X
Inner part	X		X		X

For the Vibro-Polyp especially the middle and top arm are critical during the second load case as these two carry all the load. The gravitational force is overcome by the top arm which pulls the housing up, while the middle arm is compressed and pushes the housing outwards. That is why buckling needs to be checked on this component.

Due to the gravitational pull and soil resistance and buoyancy pushing the arms up a distributed load acts on them in the third load case. This load causes a bending moment while at the same time the tension in the arms also causes stress. For the fatigue the squared sum of these stresses is used to determine the strain range per cycle.

Finally, the pins from the hinges and connection mechanism are checked for shear and fatigue.

Table 6.2: Structural checks performed on Vibro-Polyp

Component	Bending	Buckling	Shear	Tension/compression	Fatigue
Second load case					
Bottom arm					
Middle arm		X		X	
Top arm				X	
Hinge pin			X		
Connection pins (F_{hor})					
Connection pins (F_{vert})					
Piston rod				X	
Third load case					
Bottom arm	X			X	X
Middle arm	X			X	X
Top arm	X			X	X
Hinge pin			X		X
Connection pins (F_{hor})			X		X
Connection pins (F_{vert})			X		X
Piston rod					

To check the components on their failure modes the following equations have been used. The stresses on the component may not exceed the materials capacity divided by the factor γ_{DAF} used to compensate for assuming a quasi-static system.

For bending two methods have been applied depending on the load. The first being the moment acting on the component:

$$\sigma_{moment} = \frac{M * c}{I_x} < \frac{\sigma_{yield}}{\gamma_{DAF}} \tag{6.1}$$

The second type of load is distributed load due to gravity, buoyancy and soil resistance causes a bending moment. The maximum bending stress is calculated with the following set of equations.

$$q_{total} = \frac{F_{soil} + F_{buoyancy} - F_{gravity}}{L} \tag{6.2}$$

$$M_{bending} = \frac{q_{total*L}}{12} \tag{6.3}$$

$$F_{bending} = \frac{M_{bending}}{S} \tag{6.4}$$

$$\sigma_{moment} = \frac{F_{bending}}{A} < \frac{\sigma_{yield}}{\gamma_{DAF}} \tag{6.5}$$

For buckling Critical Euler buckling is used for which the following limit state can be used.

$$F_{compression} < \frac{\pi^2 E * I}{L^2} * \frac{1}{\gamma_{DAF}} \tag{6.6}$$

Basquin's relation is used to determine the fatigue damage with a strain-life method. The fatigue can be classified as mid-cycle with the number of cycles being $10e6 cycles$. For the fatigue strength exponent b a value of -0.118 is used ([14]). The fatigue strength coefficient σ'_f is taken as 1.6 times the yield strength of the material ([14]). The number of cycles follows from the Vibro-drill installation a wind farm of $100 monopiles$ in $15 minutes$ each at a frequency of $23.3 Hz$ for the eccentric weights. As stated in Section 3.2 the strain range is calculated using the difference in strain during the maximum and minimum load during each cycle of the eccentric weights. This is calculated for each component separately at the peaks of the sinusoidal eccentric force.

$$Cycles = \frac{t * f * n}{2} = \frac{15 * 60 * 23.3 * 100}{2} = 1,048,500 \tag{6.7}$$

Basquin's relation for the strain-life is as follows (Den Besten, 2018):

$$log(\Delta\epsilon) < log\left(\frac{\sigma'_f}{E}\right) + b * log(2 * N) \tag{6.8}$$

Finally for shear, tension and compression the following equation is used to determine whether the component will fail or not:

$$\sigma_{load} = \frac{F}{A} < \frac{\sigma_{yield}}{\gamma_{DAF}} \tag{6.9}$$

6.4. Verification

For the verification of the structural analysis tool two separate checks are performed. The first tests whether the loads on the structure are calculated properly from the geometry provided. The resultant force on the connection calculated by the tool should equal the sum of force on the Vibro-drill. The second check will verify the calculations of the Monte Carlo simulation. By reviewing a histogram of the load on the certain component with its capacity against the calculated probability of failure the result of the Monte Carlo simulation can be checked.

6.4.1. Verification check 1

For this verification check only the external loads on the Vibro-drill and the eccentric force are used. No internal loads are calculated below. The sum of these loads should equal the resultant load on the connection mechanism. As this check is for the calculation of the internal loads only the Monte Carlo simulation is left out and the soil characteristics as stated in Table 3.1 are used without any reduction due to jetting, liquefaction or fluidization.

The attachment is not taken into account for the soil resistance as this load gets passed on to the pile shoe directly instead of the connection. Besides this the horizontal resultant force of the Vibro-drill cancels out against the two other arms so only the vertical resultant force is important. Forces acting in upwards direction are positive and force acting in downwards direction are negative in this calculation. The areas and volumes used in the calculations where taken from Matlab and/or Solidworks.

$$\sum F_{Vibro-drill} = F_{Soil} + F_{Buoyancy} - F_{eccentric} - F_{gravity} = F_{connection} * 3 \tag{6.10}$$

Verification calculations Vibro-Boxer

For the Vibro-Boxer the total soil resistance is equal to the unit tip resistance multiplied with the frontal area of the Vibro-Boxer which is not covered by the attachment on the pile shoe. This includes only the central hub and the three arms. The gravity and buoyancy do act on the housing of the eccentric weights as well. Because the attachment on the pile shoe shields the housing of the eccentric weights from the soil resistance the total frontal area (in m^2) is very small. Using the soil characteristic stated in Section 3.2.2, the unit tip resistance acting on the Vibro-Boxer is $12\frac{MN}{m^2}$.

$$\begin{aligned} F_{Soil} &= A_{Vibro-Boxer} * min(N_q * p_0, q_{lim}) \\ &= 4.0543 * 12e6 \\ &= 48.652MN \end{aligned} \tag{6.11}$$

For the buoyancy the dimensions of the main components have been determined in Solidworks and their volumes (in m^3) are calculated. It is clear to see the large housing contributes a lot to the buoyancy of the system.

$$\begin{aligned} F_{Buoyancy} &= V_{Vibro-Boxer} * \rho_{water} * g \\ &= (V_{torpedo} + (V_{inner} + V_{outer} + V_{housing}) * 3) * \rho_{water} * g \\ &= (0.2945 + (0.0808 + 0.0860 + 3.0429) * 3) * 1025 * 9.81 \\ &= 0.099784MN \end{aligned} \tag{6.12}$$

The eccentric weights produce in total $25MN$ of force acting downwards.

$$F_{Eccentric} = 25MN \tag{6.13}$$

The masses (in kg) of some components have been calculated in Solidworks where others are estimated.

$$\begin{aligned} F_{Gravity} &= m_{Vibro-Boxer} * g \\ &= (m_{torpedo} + m_{beams} + m_{inner} + m_{outer} + m_{housing}) * g \\ &= (5000 + (630 + 905 + 762 + 6000) * 3) * 9.81 \\ &= 0.29323MN \end{aligned} \tag{6.14}$$

The resultant force as calculated by the simple sums of forces is positive. It should be noted that the soil resistance on the monopile shaft and the weight of the pile shoe and monopile are not taken into account here and therefore this value gives no insight apart from this verification.

$$\begin{aligned} \sum F_{Vibro-Boxer} &= 48.652 + 0.099784 - 25 - 0.29323 \\ &= 23.459MN \end{aligned} \tag{6.15}$$

The vertical force on three connections as calculated by the model is equal to the sum of forces. This result verifies the calculation of the internal loads.

$$\begin{aligned} F_{Connection} * 3 &= 7.3965 * 3 \\ &= 23.459MN \end{aligned} \tag{6.16}$$

Verification calculations Vibro-Polyp

For the Vibro-Polyp the general equation remains the same, but the internal components for the buoyancy and gravity do differs as can be seen below.

Unlike for the Vibro-Boxer the housing of the Vibro-Polyp is not shielded from the soil and therefore the total frontal area for the Vibro-Polyp is larger than for the Vibro-Boxer. For the total frontal are of the completely system this is the other way around, but this will be elaborated on in Section 7.1.4.

$$\begin{aligned} F_{Soil} &= A_{Vibro-Polyp} * min(N_q * p_0, q_{lim}) \\ &= 5.5269 * 12e6 \\ &= 66.323MN \end{aligned} \tag{6.17}$$

For the volumes of the components again Solidworks is used to determine the dimensions.

$$\begin{aligned} F_{Buoyancy} &= V_{Vibro-Polyp} * \rho_{water} * g \\ &= (V_{torpedo} + (V_{bottomarm} + V_{middlearm} + V_{toparm} + V_{housing}) * 3) * \rho_{water} * g \\ &= (4.1257 + (0.1742 + 0.1742 + 0.3348 + 3.0429) * 3) * 1025 * 9.81 \\ &= 0.15389MN \end{aligned} \tag{6.18}$$

$$F_{Eccentric} = 25MN \tag{6.19}$$

The gravitational force for the Vibro-Polyp consist of the masses of the central torpedo, the bottom, middle and top arm and the housing multiplied with the gravitational acceleration g.

$$\begin{aligned} F_{Gravity} &= m_{Vibro-Polyp} * g \\ &= (m_{torpedo} + m_{bottomarm} + m_{middlearm} + m_{toparm} + m_{housing}) * g \\ &= (15000 + (489.0691 + 489.0691 + 940.0289 + 6000) * 3) * 9.81 \\ &= 0.38018MN \end{aligned} \tag{6.20}$$

The total resultant vertical force on the connection of the Vibro-Polyp significantly higher than for the Vibro-Boxer as a result of the higher soil resistance acting on the structure. Again the resultant force does not include the loads acting on the monopile and pile shoe.

$$\begin{aligned} \sum F_{Vibro-Polyp} &= 66.323 + 0.15389 - 25 - 0.38018 \\ &= 41.097MN \end{aligned} \tag{6.21}$$

During the calculations for the internal loads a difference of $0.01MN$ between the values for the resultant force on the connection was created. As this error is insignificant to the order of magnitude of the forces it is assumed to be a rounding error.

$$\begin{aligned} F_{Connection} * 3 &= 13.699 * 3 \\ &= 41.096MN \end{aligned} \tag{6.22}$$

6.4.2. Verification check 2

The Monte Carlo simulation uses several parameter distributions to calculate the probability of failure. This is done by taking a sample from a distribution for every parameter and calculation the load and capacity of the component. When the load is higher than the capacity the component will fail and it will be counted. Therefore for both designs the bottom arm is chosen for which the probability of failure and capacity are calculated. By plotting the distribution and capacity an estimate can be made for the probability of failure.

Verification comparison Vibro-Boxer

The bottom arm of the Vibro-Boxer is loaded in tension as the center of the structure is pushed up by the soil. The capacity of the beam is calculated to be 6.624MN and is depicted in Figure 6.6 with a red line. It is clear to see that the peak of the load distribution is higher than the capacity of the component. The probability of failure that corresponds with the tension force on component equals 0.9824. By looking at the distribution this value is in accordance with the histogram, but an improvement to the capacity of the component is required. The distribution of the load is close to a normal distribution which is logical as the input for the soil resistance is determined using normal distribution as well.

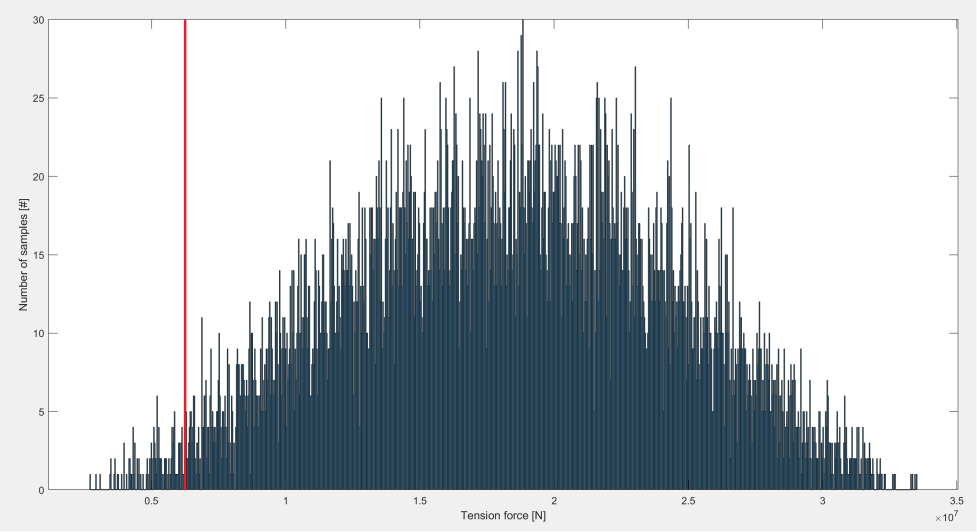


Figure 6.6: Vibro-Boxer bottom arm load and capacity

Verification comparison Vibro-Polyp

The load distribution for the Vibro-Polyp has a number of outliers which shift the peak to the left of Figure 6.7. The number of outliers is extremely low as the calculated probability of failure is 0.0024. Due to this shift, the bins of the histogram in Figure 6.7 are larger than in Figure 6.6 and the distribution is more smooth.

The capacity of the component, depicted in red, is 11.706MN and includes almost all cases and is located above the peak of the distribution. The histogram clearly shows only few cases for which the load on the component is greater than its capacity. The probability of failure, as calculated by the Monte Carlo simulation is in accordance with the load distribution.

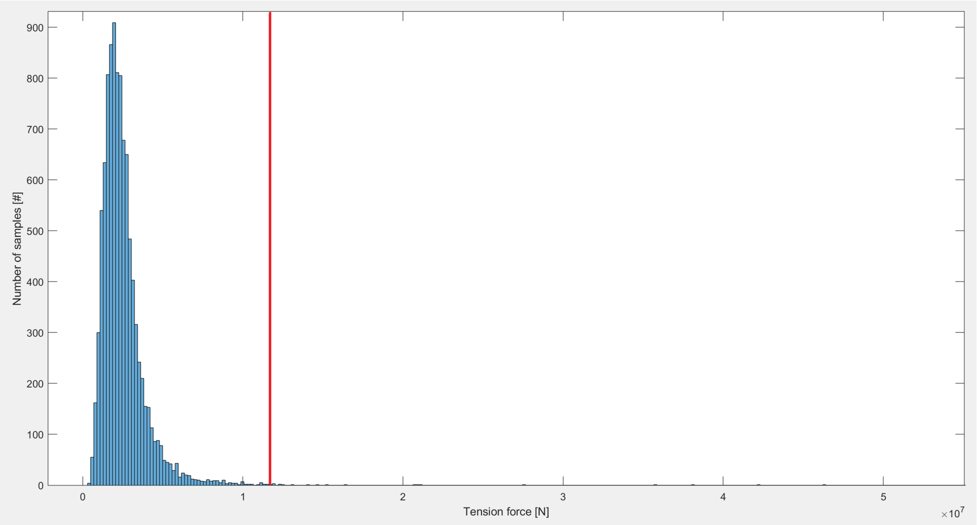


Figure 6.7: Vibro-Polyp bottom arm load and capacity

6.5. Results

After the performing all the checks the probabilities of failure of all the components and the Vibro-drill as a whole are calculated. As these probabilities are over 0.005 for both concepts, adjustments to the dimensions are needed in order to proof the reliability of the designs. For the Vibro-Boxer and Vibro-Polyp some parameters have been changed and these changes are shown in Tables 6.3 and 6.4.

For the design of the Vibro-Boxer two checks failed and therefore some parameters needed to be changed. Both beams in bending (combined) and tension/compression. The bottom beam also failed in buckling and finally the inner part of the extending arm failed in bending. Therefore the width and height of both arms are increased as well as the wall thickness of the inner part of the extending arm. As both arms are identical and tension/compression was their leading failure mode, their probabilities of failure and the required adjustments are equal as well.

Table 6.3: Changed parameters Vibro-Boxer

Parameter	Failure mode	Old value	New value	Unit
Width bottom beam	Tension	120	270	mm
Height bottom beam	Tension	120	270	mm
Width top beam	Compression	120	270	mm
Height top beam	Compression	120	270	mm
Wall thickness inner part extending arm*	bending	-	-	mm

* The wall thickness at the bottom, top and sides of the inner part of the extending arm were increased as these contribute to the moment of inertia. The outer fibers are more effective and therefore these have been increased the most.

The probability of failure of the Vibro-Boxer after the improvements to the design is 0.0020. As this is below the desired level, no further adjustments have been made. The probability of failure as stated above is the change of one failure out of all checks. On an individual level the bottom and top beam are still the most critical components as their probabilities of failure due to fatigue are higher than the others. Because these probabilities are below the desired level there is no need for more changes to the design.

For the Vibro-Polyp many components are fatigue sensitive and needed to be adjusted. As the bottom and middle arm have the large strain ranges, they are critical component for fatigue damage. For both arms large changes needed to be made. The top arm is less critical, but also needed a slight increase in diameter. Finally the pins for the hinges as well as for the connection mechanism needed large adjustments in order to cope with the massive strain range that act upon them. Because the strain range is much larger compared to the maximum stress the fatigue-life of these components is short.

Table 6.4: Changed parameters Vibro-Polyp

Parameter	Failure mode	Old value	New value	Unit
Diameter bottom arm	Fatigue	300	600	mm
Wall thickness bottom arm	Fatigue	30	70	mm
Diameter middle arm	Fatigue	300	500	mm
Wall thickness middle arm	Fatigue	30	60	mm
Diameter top arm	Fatigue	300	500	mm
Diameter hinge pin	Fatigue	180	270	mm
Number of pins (F_{hor})	Fatigue	12	20	-
Number of pins (F_{vert})	Fatigue	14	30	-
Diameter pin	Fatigue	73	120	mm

After the changes to the design the probability of failure is 0.0036. This is below the desired limit and therefore no further adjustments are made to this design. It should be noted that the dimensions of the arms and the pins have increased a lot. The results for the design is that the frontal area as seen from below is also greater.

The increase in total frontal area is 24% which increases the soil tip resistance by the same percentage making the Vibro-drill install monopiles less efficient.

Some of the changes affected the frontal area of the Vibro-drill and with that the soil resistance, but almost all alterations done after the structural checks added weight to the Vibro-drill resulting in higher internal loads. The final internal loads acting on the structural components are displayed. These loads are the averages calculated over all 10,000 samples of the Monte Carlo simulation.

Table 6.5: Average internal loads acting on Vibro-Boxer components

Component	**Load type**	**Dominant failure mode(s)**	**Force [MN]**	**Moment [MNm]**
Second load case				
Bottom beam	Normal stress	Buckling	1.1465	-
Top beam	Normal stress	Tension	0.125	-
Combined beams	Bending moment	Bending	-	0.21670
Outer part	-	-	-	-
Inner part	Bending moment	Bending	-	0.12307
Third load case				
Bottom beam	Normal stress	Tension	14.05	-
Top beam	Normal stress	Compression and buckling	14.05	-
Combined beams	Bending moment	Bending	-	2.655
Outer part	Shear stress	Shear	5.128	
Inner part	Bending moment	Bending	-	7.922

It is clear to see that the third load case is driving for the design. Especially the bending moment on the inner part of the extending arms is large due to the long arm on which all the forces act. The inner part of the extending arm has been reinforced to cope with these loads though.

Table 6.6: Average internal loads acting on Vibro-Polyp components

Component	**Load type**	**Dominant failure mode(s)**	**Force [MN]**
Second load case			
Bottom arm	-	-	-
Middle arm	Normal stress	Compression and buckling	0.11232
Top arm	Normal stress	Compression	0.16823
Hinge pin	Shear stress	Shear	0.16823
Connection pins (F_{hor})	-	-	-
Connection pins (F_{vert})	-	-	-
Piston rod	stress	Compression	0.17398
Third load case			
Bottom arm	Normal stress	Tension	3.2450
Middle arm	Normal stress	Tension	2.3211
Top arm	Normal stress	Tension	1.1072
Hinge pin	Shear stress	Shear	3.2751
Connection pins (F_{hor})	Shear stress	Shear	5.8105
Connection pins (F_{vert})	Shear stress	Shear	14.876
Piston rod	-	-	-

For the Vibro-Polyp it is clear to see that the force on the connection pins outweighs all other forces acting on the system. When looking at the geometry it is logical to conclude that the soil drags the central hub upwards which in turn pulls on the connection pins. It also explains why the number and diameter of these pins need to be increased. Note that the force acting on the connection pins is higher than stated in the verification as result of the increased diameter of the arms and thus the increased soil resistance.

6.6. Special load cases

For each of the concept certain structural risks have been identified that have not been considered during the structural analysis. Because the detailed design is required in order to check the component for these special loads, these checks will be performed in Chapter 8. Among these special loads are as follows:

- A rock hitting the bottom arm of the Vibro-drill. For this load a rock with a diameter of $1.0m$ is supported by soil an hits small area on the bottom arm. This can be modelled as a point load acting on a plate in which the point load is equal to the soil resistance of A_{rock}.
- The axle that connects the eccentric weights needs to keep them in phase. The torque acting on this axle is unknown, but an assumption must be made in order to perform the check. The maximum torque can be approximated by the power of the hydraulic pump divided by the rotational velocity of the weights. This equals $10kNm$ of torque. The axles length and diameter are determined in the detailed design phase and afterwards the check is performed.
- Finally for the Vibro-Polyp the housing for the eccentric weights needs to be checked for the forces acting on it. Unlike for the Vibro-Boxer, one arm is attached to the top of the housing acting loads on it. The housing should not collapse under these loads.

6.7. Conclusion

The results from the structural analysis show the the Vibro-Boxer has a lower probability of failure and only the beams are critical components in buckling. For the Vibro-Polyp fatigue plays an important role as the top arms and pins are all sensitive to failure. The dimension have been increased which may require further adjustments to the design in order to fit the larger component.

For some checks the additional capacity of the component can be reduced in order to save weight and costs. Room for improvement is larger for the Vibro-Boxer than for the Vibro-Polyp due to its short fatigue-life time. An optimisation can be added to the tool, but it should be noted that the tool is limited due to simplification to a quasi-static system. Only when the Vibro-drill is modelled as a dynamic system should optimization be performed.

Test results must be used to validate the soil resistance on the system and the required eccentric force. These loads play a major role in the structural analysis and the uncertainty of current data lowers the reliability of the tool. After updating the soil characteristic and distributions for the soil reduction factor, the design can be checked again and should be adjusted accordingly.

7

Concept comparison

To determine which of the two concepts is the most suitable a comparison is made based on the design criteria as stated in Section 3.4 and other risks that were found during the concept design phase. First the design criteria are repeated below and discussed for both concepts. Then Section 7.2 will present the risks that haven't been mentioned in the design criteria and finally a choice will be made between the two concepts.

7.1. Design Criteria

Per design criteria both concepts will be discussed and, when possible, the most suitable concept will be named. The most relevant argument for one or another concept will be used to determine which concept will be used for the final design.

- The probability of a loss of the monopile due to the failure should be a minimum.
- Vibro-drill must be cheap to manufacture.
- Minimal time/cost needed for repair.
- The frontal area of the Vibro-drill must be kept to a minimum to reduce the tip resistance.
- Easy fabrication (no rare machinery required).
- Easy and fast mounting procedure for the Vibro-drill.
- No or minimal adjustments to the monopile are preferred.
- Redundant connection system/back-up is preferred.
- Redundant release system is preferred.
- Use of proven technology is preferred.

7.1.1. Risk loss of monopile

The risks of losing a monopile due to failure of the connection or Vibro-drill should be ALARP. For both concepts the structural analysis reveals some critical components of which the severity is worse for the Vibro-Polyp. For the loss of a monopile some structural failure are allowed, as long as the eccentric weights are attached to the monopile and they are powered the installation might continue, but retrieval is in nearly all cases impossible. Therefore the lower probability of failure is an important benefit for the Vibro-Boxer. For the connection mechanism the Vibro-Polyp has a slight advantage as the pins will stay in the holes when the system fails where the hydraulic clamp will simply let go of the attachments when the pressure drops. Overall the Vibro-Boxer scores better on this criteria due to the structural analysis.

7.1.2. Costs

To make an estimate for the concepts is difficult as no comparable machines exist. As both concepts do not require excessive amounts of material or exotic material the cost of raw steel will not make a big difference. The same goes for the fabrication costs as the components due not require specialist tools or machinery. The twelve hydraulic gripper for the Vibro-Boxer combined might be expensive but so are the large hydraulic pistons that push up the guidance ring of the Vibro-Polyp. All together there is not enough information available at this point in the design process to pick a winner for this criteria.

7.1.3. Repair and/or maintenance

No repairs or maintenance should be required in between installations. This is the same for both concepts and therefore this design criteria doesn't make a difference.

7.1.4. Frontal area

The total frontal area for the Vibro-Boxer including pile shoe is $15.97m^2$ and the area for the Vibro-Polyp is (after the structural analysis) is $13.32m^2$. Figure 7.1 shows the different frontal areas relative to the monopile wall, the pile shoe, the old Vibro-Polyp design and the total frontal area of the monopile. It should be noted here that for the Vibro-boxer there is still room for improvement by narrowing the extending arms were as for the Vibro-Polyp the frontal area can not be decreased anymore while keeping the probability of failure below the limit. The extra soil resistance needs to be compensated by the eccentric weights. The difference in total soil resistance is approximately 10%. This is a huge benefit for the Vibro-Polyp.

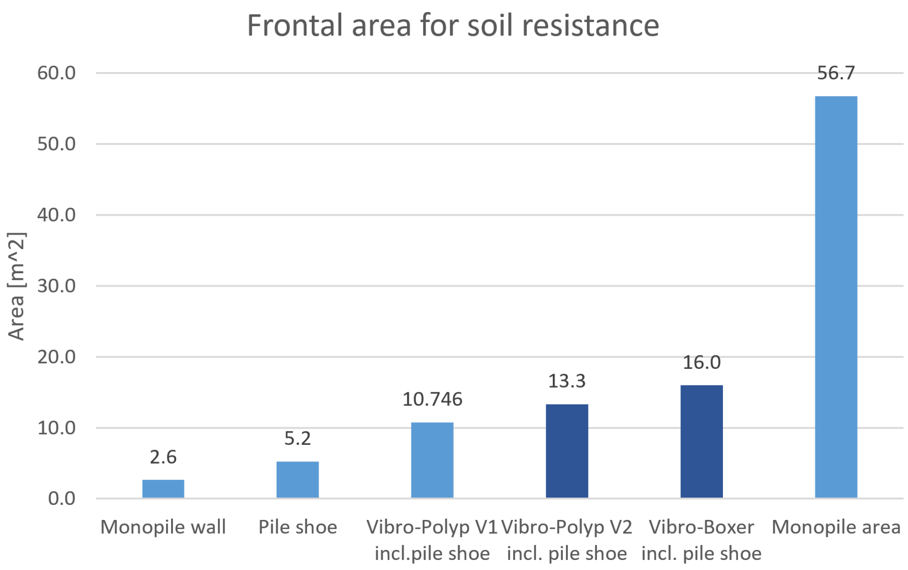


Figure 7.1: Frontal area on which the soil resistance acts

Besides looking at the frontal area for the total soil resistance it is also important to look at where these loads act on. The Vibro-Polyp takes almost all of the soil load on the hub, arms and housing, where as the housing is covered by the attachment on the pile shoe for the Vibro-Boxer. This means that a large part of the soil resistance acts on the pile shoe instead of on the structure of the Vibro-drill. This reduces the internal loads and the resultant force that acts on the connection mechanism. In vertical direction the force on the hydraulic grippers is $6.551MN$ versus $14.876MN$ for the Vibro-Polyp. During the structural analysis this point was proven by the amount of adjustments required to the Vibro-Polyp, where as the reliability of the Vibro-Boxer is higher. In conclusion the Vibro-Polyp has an advantage as it causes less soil resistance, but for the reliability as stated above, the Vibro-Boxer wins.

7.1.5. Fabrication

Similar to the costs also the fabrication time is difficult to estimate. During the detailed design phase changes can be made to make fabrication easier and quicker. For the Vibro-Polyp the pins for the connection mechanism will be time consuming to manufacture in such numbers. The beams of the Vibro-Boxer have serrated sides which are also difficult to produce, but besides those no significant differences can be identified between the two concepts. Once again no argument can be made for one or the other concept.

7.1.6. Mounting procedure

After the monopile is upended and the Vibro-drill is lowered, it needs to extend its arm and attach itself to the pile shoe. As this process can only right before the installation of the monopile it must be done quickly to reduce the installation time. The Vibro-Polyp is most likely to fold open in less time then it takes the Vibro-Boxer to extend its arms. The required precision for the pins-and-holes system to align and secure causes this benefit to be counteracted. The hydraulic grippers will grab onto the attachment instantly and the alignment does not need to be as precise.

7.1.7. Adjustments to the monopile

As both concepts require a pile shoe with an attachment on it, the changes made to the monopile can be considered equal. In both cases a pile shoe needs to be welded to the monopile before transport to the installation vessel.

7.1.8. Redundancy connection

The twelve hydraulic grippers of the Vibro-Boxer provide some redundancy, but the two hundred and forty pins the Vibro-Polyp uses to connect itself to the pile shoe provide a lot more. Besides this, a failure in the hydraulic system will release the Vibro-Boxer instantly where the Vibro-Polyp will stay in place, but won't come off. The loss of the Vibro-drill has a smaller consequence then the loss of a monopile. As this exact risk is already used for the first criteria, only the number of pins versus the number of hydraulic clamps gives the Vibro-Polyp is slight advantage.

7.1.9. Redundancy release mechanism

In contrary to the connection mechanism the redundancy for release mechanism is better for the Vibro-Boxer. As stated above, the Vibro-Polyp needs hydraulic pressure to pull all of the pins out before it can be retrieved, where the Vibro-Boxer can be pulled up as soon as the pressure is released.

7.1.10. Use of proven technology

From Chapter 2 can be concluded that the combination of hydraulic grippers and vibratory hammers is used frequently. The use of pins to hold an object in place is less common. Especially in the offshore industry, the use of proven technology benefits the certification process of new equipment. The extending arms as well as the folding arms have no exact resemblance in the current offshore industry. The Vibro-Boxer has an advantage over the Vibro-Polyp for this criteria.

7.2. Risks

Not all risks have been covered by the design criteria discussed above. The risks that are important to mention are summed up below.

- The clamp mechanism in the extending arm of the Vibro-Boxer can fail. When this happen extending or contracting the Vibro-Boxer will become impossible. In the detailed design this part of the design can be made such that this does not happen.

- The axle for keeping the eccentric weights in phase runs through the middle arm and need to operate under an angle. When the joints fail the eccentric weights can start to counteract the vibrations rendering the Vibro-drill useless. The combination of the high torque, high rotational speeds and vibration is hard on the joints especially considering the number of cycles.

- For both concepts the axles for phasing the eccentric weights need to be connected with a gearing system. This system must be strong enough to handle the torque of all the eccentric weights and at the same time fit in the central hub.

Especially for the Vibro-Polyp the a failure in the connection between the eccentric weights to keep them in phase is a high risk. As the vibrations are one the main functionalities of the Vibro-drill, this risk plays an important role in the decision for the most suitable concept.

7.3. Design choice

When all arguments are compared and weighted against each other it can be concluded that the Vibro-Boxer is the most suitable concept for the Vibro-drill. The simple design has only few moving parts and even though it has a larger frontal area, causing more soil resistance, a large part of this load acts on the attachment on the pile shoe. Therefore the loads on the Vibro-Boxer is low, reducing the risk of a structural failure. This is clearly visible as the probability of failure is lower for the Vibro-Boxer than for the Vibro-Polyp.

In conclusion it is the higher reliability and a proven technology for the connection of the Vibro-Boxer that is more important than the increase in soil resistance. The test results at GBM must verify that the increase in soil resistance can be overcome with the methods for reducing the soil resistance. The total eccentric force required to install a monopile using the Vibro-Boxer must result from these tests in order to determine the true viability of this design.

III

Final Design

8

Detailed design

For the detailed design of the Vibro-drill some parts of the Vibro-boxer, discussed in Chapter 5, are improved and/or worked out in more detail. In this chapter, the requirements for these parts are discussed separately before a solution is described. First a general overview of the final design is given.

8.1. General characteristics

The final design of the Vibro-Boxer is displayed in Figure 8.1. In this new design the outer part of the extending arm (blue) has been reinforced and both parts of the extending arm are higher to cope with the loads during a rock impact. The beams are increased in width and height and resulting from this change the clamp system for the beams has been modified. The new system is explained in Section 8.3. Also the eccentric weights have been placed in the housing.

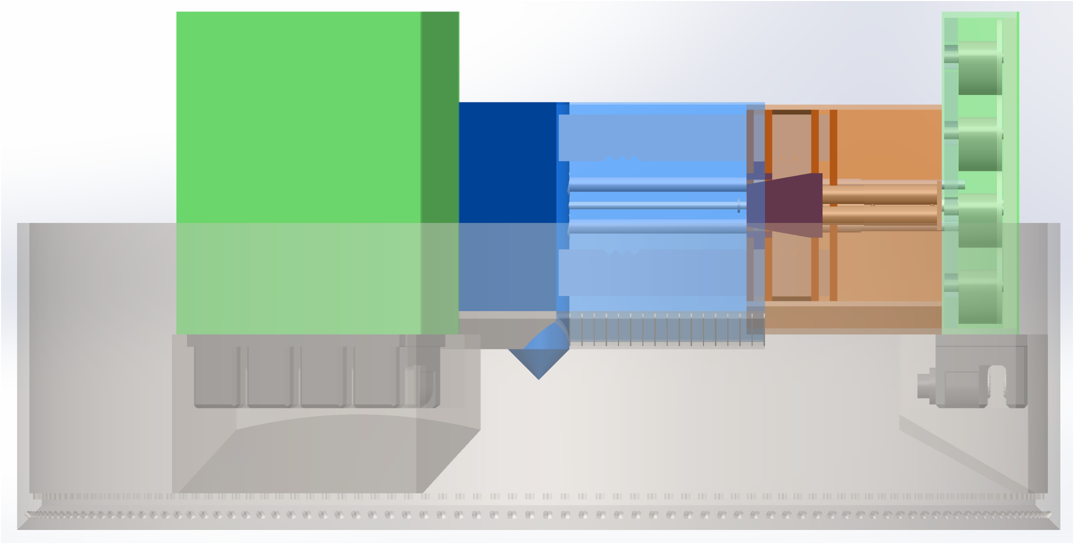


Figure 8.1: Side view final design Vibro-Boxer

The height of the extending is 2010mm and the height of the housing is 2628mm. The total height of the Vibro-Boxer, measured from the bottom of the hydraulic gripper to the top of the housing of the eccentric weight is 3228mm. The radius of the extended Vibro-Boxer at the most outer fiber equal to the inner diameter of the monopile; 8300mm.

The total weight of the Vibro-Boxer as depicted below is 160$tonnes$. Without the pile shoe and attachment the Vibro-Boxer itself weighs 69$tonnes$. This is the mass that needs to be retrieved after installation. It should be noted though that the additional parts such as hydraulic lines, bearings, cables and sensors will add more weight to the Vibro-Boxer.

8.2. Rock impact analysis

As stated in the structural analysis a rock impact is a potential threat to the Vibro-drill. The extending arms of the Vibro-Boxer are critical parts in this specific load case and therefore it is analysed what an impact of a rock hitting the outer part of the extending arm would do. As depicted in Figure 8.2, the rock will hit halfway the the outer part of the arm causing a large bending moment in this component. Besides this also the bending moment in the two beams and the inner part of the extending arm will increase.

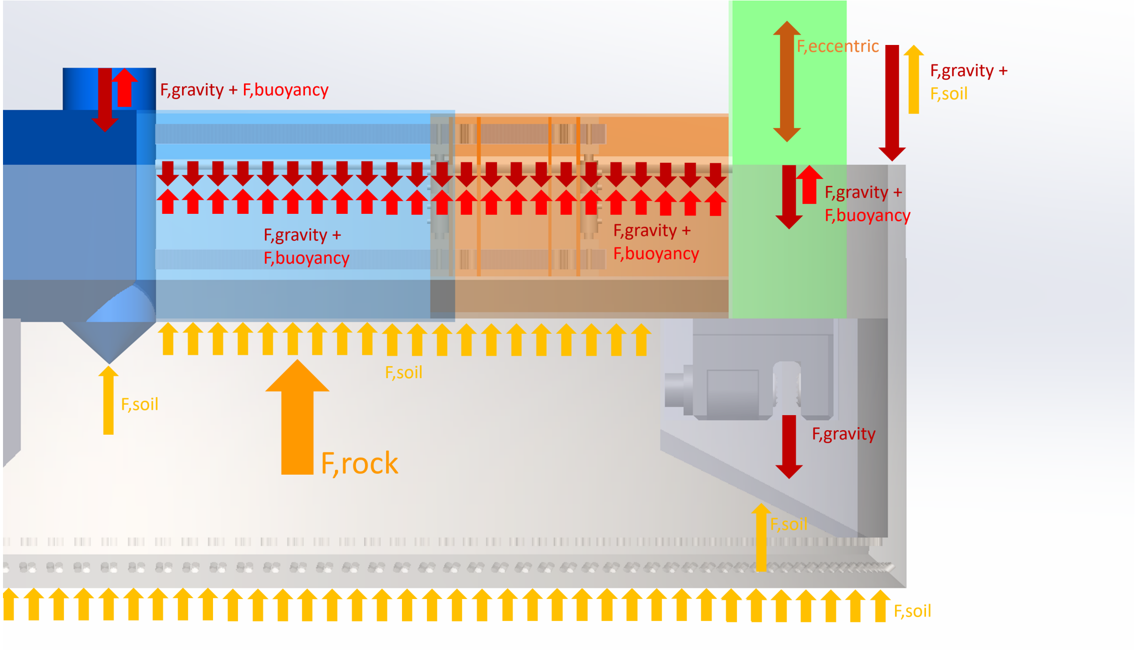


Figure 8.2: External and internal forces on the Vibro-Boxer during operations with a rock impact

As large rocks can be detected during the geotechnical survey these are avoided when deciding on the location of the wind turbine. Therefore only small rocks are to be encountered. A rock of $1.0m$ in diameter can still cause a high load on the structure. As the rock is supported the maximum load it can apply equal the soil load acting on its frontal surface area. Because the rock must be positioned under the arm in order for it to load it, it will reduce the soil resistance on the arm as well. After adding this load the structural analysis was repeated. The results show that the Vibro-Boxer fails several structural checks.

For the rock impact analysis the outer part of the extending arm needs to be checked for bending as well and the bending moment is too large for the component. An increase in height adds bending strength. As a dent in this part would prevent the inner part of the arm to retract, an external reinforcement plate is added as well. The latter only prevents the bottom of the arm from denting.
Besides the outer part of the extending arm, also the inner part fails in bending. As the outer part is increased in height the inner part is heightened as well. Apart from the fact that the inner part is already strengthened by a greater wall thickness, a dent is also less critical. Therefore no reinforcements are needed here. Finally the beams are made wider and higher to cope with the larger tension and compression forces.

Table 8.1: Changed parameters Vibro-Boxer for rock impact

Parameter	**Failure mode**	**Old value**	**New value**	**Unit**
Width bottom beam	Bending and tension	270	380	mm
Height bottom beam	Bending and tension	270	380	mm
Width top beam	Bending, buckling and compression	270	380	mm
Height top beam	Bending, buckling and compression	270	380	mm
Height inner part extending arm	bending	920	1660	mm
Height outer part extending arm	bending	960	1700	mm

Due to the method used for the design that follows from this analysis is conservative. By calculating the loads on one arm and then applying the values to all three arms, this model suggests that all three arms will hit a rock at exactly the same time. For the bending moments in the arms are therefore higher then to be expected in reality.
After the improvements the following internal loads were calculated for the structural components during operation and a rock impact. Note that during the second load case the Vibro-drill is still hanging in the crane and therefore a rock impact is not possible.

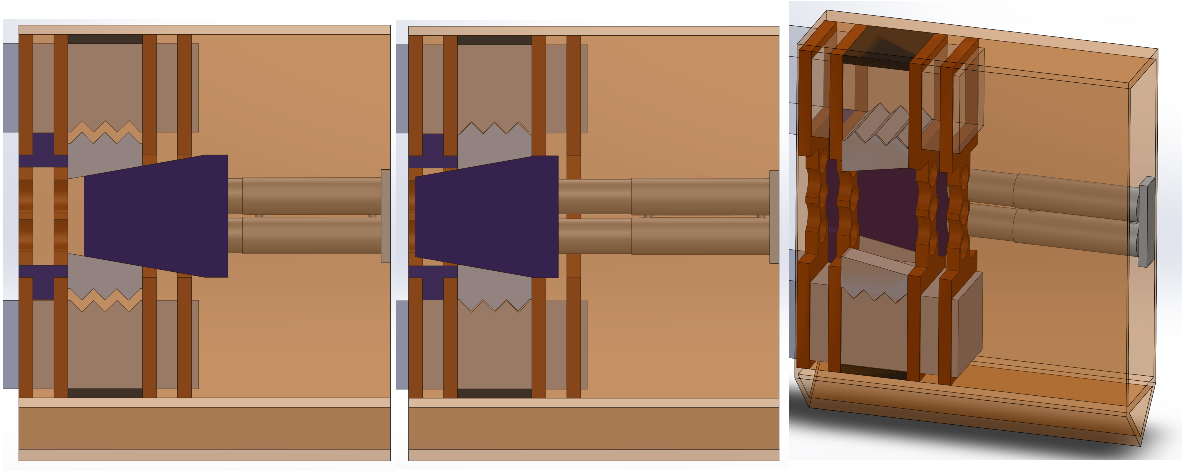
Table 8.2: Average internal loads acting on Vibro-Boxer components with rock impact

Component	**Load type**	**Failure mode(s)**	**Force [MN]**	**Moment [MNm]**	**Increase [%]**
Bottom beam	Normal stress	Tension	22.3256	-	59
Top beam	Normal stress	Compression and buckling	22.3256	-	59
Combined beams	Bending moment	Bending	-	4.395	66
Outer part	Bending moment	Bending	-	0.86550	-
Inner part	Bending moment	Bending	-	13.179	66

The new probability of failure for the Vibro-Boxer with reinforced arms is 0.0022 for a rock impact during operations. The weight has increased due to the height of the arms, but the frontal area of the Vibro-Boxer remains the same.

8.3. Clamps beams

The beams in the extending arm of the Vibro-Boxer have been increased in width and height in order to cope with the rock impact. As a result of this the clamp system needed improvement as well. Because the vibrations will move up and down, it is sensible to align the piston in a different direction. By using a wedge to clamp the beams in position the piston force can be kept low as long as the stroke is long enough. In the design, shown in Figure 8.3, the wedge has an angle of 5 degrees and the piston have a stroke of 300mm. Besides the clamp system also the phasing axle and two pistons for extending and contracting the arm are placed inside this component. By using two wedges and two sets of pistons on the sides, the middle of the arm allows room for the other components.


(a) Open clamp system

(b) Closed clamp system

(c) Closed clamp system (only single sided)

Figure 8.3: Side view clamps beams extending arms

By using the tension force calculated in Section 8.2 the required force on the clamp should be $22.33MN$. Due to the angle of the wedge the pistons need to produce $1.953MN$ per beam. Using two pistons at $500bar$ with a rod diameter of $160mm$ this requirement can be met.

$$F = P * A = 50e6 * 2 * \pi * \frac{0.160^2}{4} = 2.011MN \tag{8.1}$$

8.4. Housing eccentric weights

GBM estimates that the eccentric weights need to provide $1200kgm$ of moment. In Section 3.2 this was calculated to provide sufficient eccentric force. Due to the beams running into the housing of the eccentric weights, the lay-out must be avoid the center and bottom. The number of eccentric weights must be even as otherwise a horizontal resultant force cause unwanted loads on the system.

$$m = W * \pi * r^2 * \rho = 0.315 * \pi * 0.320^2 * 800 = 380kg \tag{8.2}$$

$$H_{cog} = \frac{4 * r}{3 * \pi} = 0.132m \tag{8.3}$$

$$M_{eccentric} = * H_{cog} * 0.8 = 40kgm \tag{8.4}$$

The resulting lay-out is shown in Figure 8.4. Ten eccentric weights are placed on toothed wheels which act as gears. The radius of $320mm$ and width of $315mm$ provide an mass of $380kg$. The height of the center of gravity is $132mm$ and the eccentric moment per weight is $40kgm$. With ten weights per housing and one housing on each of the three arms, the total eccentric moment of the Vibro-Boxer is $1200kgm$. The central weight is aligned with the phasing angle.

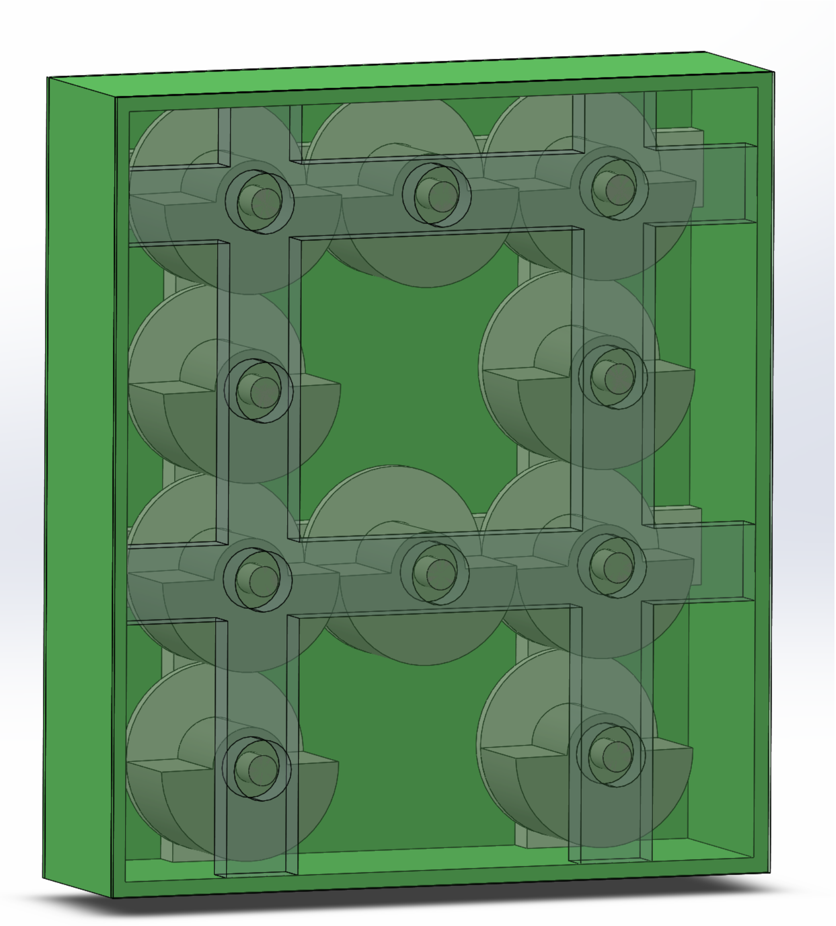


Figure 8.4: Housing with the eccentric weights

In order to fit ten eccentric weights, the height of the housing is increased from $2100mm$ to $2628mm$. This change does not affect the soil resistance and therefore the structural design can be kept the same. Also the total mass of the housing was estimated for the structural analysis, but as this was done based on calculated for the eccentric moment, the difference between the estimate and actual mass of the housing is 2.7%, which does not affect the structural analysis.

8.5. Phasing axle

This section is on the axle that connect the eccentric weights in the different housings in order to keep them in phase. The axles run through the extending arms and are connected by a bevel gear in the central hub. By mechanically connecting the eccentric weights they are kept in phase, but in case of a failure at one the eccentric weights the connection needs to be broken to prevent a larger failure. Below five possible situations are specified to determine at what torque the phasing axle should be disconnected. A torque decoupler, designed for this torque, should be included in the system to do so.

- All units are operational, the axle needs to transfer a certain percentage of the torque produced by a hydraulic engine in order to correct the eccentric weights when going out of sync. $T_{decoupler} >= 20\%$ of T_{engine} depending on the precision of the controller of the hydraulic engine.*
- One engine fails and eccentric weights are free to rotate, the axle needs to transfer torque from the other three engines to the non-powered eccentric weights. $T_{decoupler} >= 67\%$ of T_{engine} (in case of 3 arms).
- One engine fails and eccentric weights are blocked, the axle needs to break to allow the other unit to keep on functioning. $T_{decoupler} <= 100\%$ of T_{engine}.
- One of the eccentric weights jams and blocks a unit, the axle needs to break to allow the other unit to keep on functioning. $T_{decoupler} <= 100\%$ of T_{engine}.
- The bevel gears blocks, the machine is rendered useless as the eccentric weights will run out of phase.

* The engines run on hydraulic fluid and cannot be controlled instantly to be kept in phase with the other two engines. The amount of torque required to correct the imbalance is unknown and depends on the friction and dynamics of the system. This is to be tested but of this design exercise it is assumed to be 20%.

It can be concluded that preferably the decoupler should break at 80 to 90 percent of the torque produced by a hydraulic engine. This allows the axle to power the eccentric weights in case one hydraulic engine fails.

IV

Conclusion

9

Conclusion

As the current design of the Vibro-drill is not meant for retrieval every monopile installation would require a new Vibro-drill. Besides the practical and environmental issues this would bring, the economical viability of the Vibro-drill would be very low. As adjusting the current design is not possible, a new design has been made such that it can be retrieved after installation without losing its functionalities during installation. In this report the design exercise is described and the main findings are summed up below. Some arguments and design choices are discussed in Section 9.2. Finally some recommendations for GBM Works are given to further improve the design of the Vibro-drill.

9.1. Conclusions

During the process analysis three load cases were found. The most critical loads on the Vibro-drill structure are the forces from the eccentric weights and the soil resistance. From CPT data a distribution for the soil resistance is made. For the effectiveness of jetting, fluidization and liquefaction, no test results are experimental data is available. Therefore it is assumed that these methods reduce the soil resistance by a percentage between 10% and 80%. These values should be updated after the test results from GBM are available.
The focus of the design is on the connection mechanism and the structure of the Vibro-drill. Besides resisting the loads on the system the Vibro-drill must be able to change its size. Because the Vibro-drill attaches itself at the bottom of the monopile and needs to be retrieved through the top of monopile which has a smaller diameter.

Of many possible solutions a selection has been made based on the design requirements. On this selection of potential solutions a MCA was performed. The result were two concepts for the Vibro-drill that meet the design requirements. The Vibro-Boxer has extending arms which allow it to fit through the narrow top of the monopile after installation. The Vibro-Polyp uses three sets of folding arms to modify its size such that retrieval is possible.
Both of the concepts use the pile shoe, a ring below the monopile, to connect the Vibro-drill with. Especially for jetting and fluidization and structural rigidity the pile shoe scored well in the MCA.
The connection mechanism is also different for these concepts. The Vibro-Boxer has, commonly used, hydraulic grippers which attach to the pile shoe. The Vibro-Polyp has a system with pins that are pushed in holes in the pile shoe.

The structural analysis, which was performed on both concepts, showed that some component were unable to resist all the loads and especially fatigue would be causing failures. The components with a high probability of failure were improved and the structural analysis tool was used to recalculate the probability of failure. After several improvements, the structural analysis tool estimates the probability of failure for both concepts to be less the 0.005, which is the desired value.
The dimensions for multiple components of the Vibro-Polyp are increased resulting in a higher frontal area and thus soil resistance. At the same time the design is still sensitive to fatigue.

Especially due to the higher structural reliability the Vibro-Boxer is a more suitable design. Another argument for the Vibro-Boxer is use of hydraulic grippers which are already used in the offshore pile driving sector. The frontal area, and with that the soil resistance, is smaller for the Vibro-Polyp, but the risks concerning its structural rigidity, connection mechanism and other components are too high for this concept. Therefore the final design is based on the principles of the Vibro-Boxer.

The detailed design focuses on some critical parts that hadn't been addressed yet. Also some details have been improved, but no major changes have been made to the design of the Vibro-Boxer. The Vibro-drill has been re-designed such that it can connect to and disconnect from the monopile. The extending arms make it possible to be pulled through the top of the monopile after installation and the eccentric weights and holes for jetting and fluidization have been placed on the ends of these arms and on the pile shoe. With that the Vibro-drill is designed such that it can be retrieved after installation without losing its functionalities during installation.

9.2. Discussion

The uncertainties in the load analysis have required the need for assumptions. Some of these assumptions are estimates based on literature or experimental data, but others have scientific ground. The latter are mentioned below and their consequence on the reliability of the design is discussed. Besides this also some crucial design choices are elaborated on.

- The distribution for the soil resistance is taken from CPT data measured at Hollandse Kust while the mean unit soil resistance is calculated with the properties of very dense sands. As a lot of locations have better soil conditions for monopile this is a very conservative choice. When it is known that the Vibro-drill will be used in weaker soil only, the loads on the structure will be smaller and the design can be optimized in terms of size and weight.

- With the simplification from a dynamic system to a quasi-static system the reliability of the results of the structural analysis has its limitations. Including the inertia of the system and damping of the soil will change the internal loads. Especially when dynamic amplification occurs these loads can be much larger than calculated in the structural analysis. In this case the components in which loads are amplified should be adjusted such that their Eigenfrequencies shift away from the frequency of the induced vibrations.
  Higher loads and greater strain ranges are to be expected when inertia is included in the analysis. After a structural analysis including the dynamics of the system some components might fail due to increased loads and/or fatigue stresses.

- During this design exercise several design codes have been reviewed, but none were guiding during this process. Because the Vibro-drill is a new piece of equipment to perform a new installation method no design code fits all of its characteristics. Certification of the current design is therefore difficult.

- As stated in the introduction of this report, the monopiles of the new generation of wind turbines are larger. The monopile used to design to Vibro-drill for has a length of $80m$ and a bottom diameter of $8.5m$. It is assumed that within the next two years, in which the Vibro-drill is still under development, this diameter will be commonly used. If this assumption is incorrect the design of the Vibro-Boxer can be adjusted easily such that it will fit the correct monopile diameter.

9.3. Recommendations

Resulting from this design exercise, some recommendations can be done for GBM Works to improve the design of the Vibro-drill. With the uncertainties concerning the soil loads on the system, the methods to reduce these and consequently the forces required to drive the monopile, the design can be optimized after more data is available. This is where most of the recommendation below will contribute to.

- Perform scale test and use the test results to determine the effectiveness of jetting, liquefaction and fluidization. By updating the distribution used in the structural analysis, the range of the loads on the structure decreases enormously and the design can be improved in terms of reliability, size and weight.

- By modelling the Vibro-drill and monopile as a dynamic system the loads on the structural components are calculated more accurate to their real world values and therefore components can be optimized accordingly. Since the Vibro-drill has to produce large vibration it is recommended to perform a study on the dynamic behaviour of the components in order to prevent dynamic amplification. Besides this the internal loads will change and may increase to levels above the structural capacity of the component. For all structural components the eigenfrequency must be determined and compared to the frequency of the eccentric weights, inertia must be included in the structural analysis and the structural checks, as stated in Chapter 6, must be performed with the new loads.

- The current design is based on the assumption that the monopile is installed using an installation vessel with a crane. As a floating installation would bring forward new challenges especially concerning the process of attaching the Vibro-drill to the monopile underwater, it is interesting to investigate the performance of the current design. Due to the Vibro-drill adding weight on the bottom of the monopile the behaviour of the monopile under hydrodynamic loading might be improved, resulting in an advantage for the GBM Vibro-drill over competitive installation methods.

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V

Appendices

A

CPT measurements Hollandse Kust

This appendix elaborates on the distribution for the soil resistance in the Monte Carlo simulation. The CPT measurements were obtained as raw measurement data filed per CPT location. Only the cone resistance was used as the shear strength data was incomplete at some locations. The values for the cone resistance at target depth of $40m$ are incomplete as well and therefore the measurements taken at $20m$ depth are used. A check has been performed to see whether the standard deviation at different depths differs. At this is not the case, taking the standard deviation at $20m$ instead of $40m$ provides representative values.

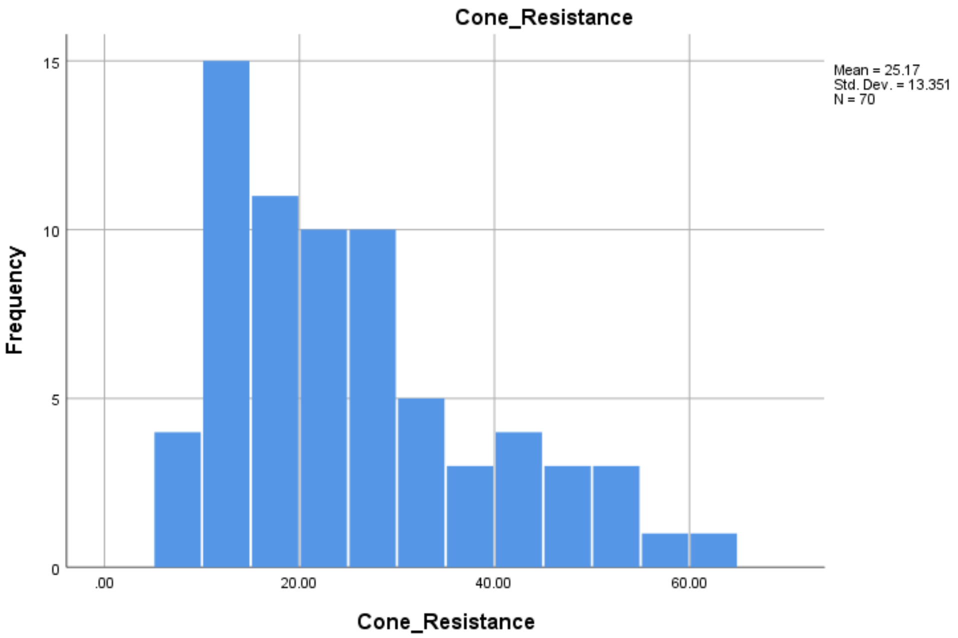


Figure A.1: Cone resistance at $20m$ depth in 70 locations of Hollandse Kust

The cone resistance has a average value of $25.17MPa$ with a standard deviation of $13.35MPa$. It is clear to see there is a large spread, but mostly positive of the average. The Monte Carlo simulation requires a normal distribution of the cone resistance and therefore the data set is normalized.

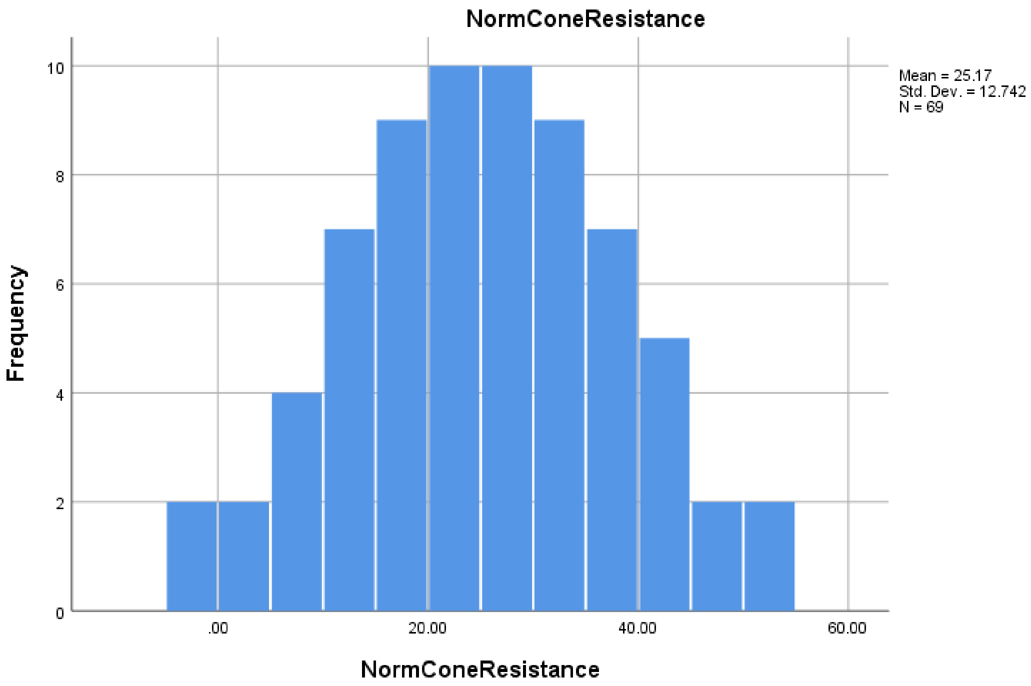
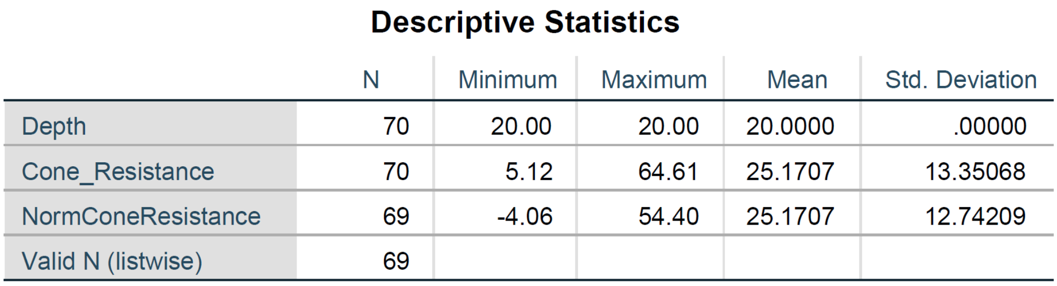


Figure A.2: Normalized cone resistance at 20m depth

The process of normalizing the data set takes one data point out of the data set which explains that there are sixty nine cases left. The descriptive statistics in Table A.1 show the mean is equal for both variables and the standard deviation is almost equal as well.
It can be noted that the maximum cone resistance is lower for the normalized data than the maximum cone resistance for the non-normalized data set. When a larger number of samples is used with the distribution provided some data points will exceed the maximum maximum cone resistance for the non-normalized data set.

Table A.1: Cone resistance statistical descriptives



Descriptive Statistics

	N	Minimum	Maximum	Mean	Std. Deviation
Depth	70	20.00	20.00	20.0000	.00000
Cone_Resistance	70	5.12	64.61	25.1707	13.35068
NormConeResistance	69	-4.06	54.40	25.1707	12.74209
Valid N (listwise)	69				

By dividing both the mean as the standard deviation with the mean a distribution around 1 is created which can be used for the soil resistance parameter in the Monte Carlo simulation. For this parameter the mean becomes 1 and the standard deviation becomes 0.5062.

B

Friction Coefficient

The hydraulic gripper is supposed to create enough friction on the monopile to keep the Vibro-drill in place before and during installation. This static friction is caused by the normal force a hydraulic piston perform on the monopile. Determining the required normal force means a friction coefficient is needed. As the friction force is dependent on many factors, of which vibrations are the most critical, this appendix is dedicated to the literature study on these effects. For the gripper only the static friction coefficient is important because movement of the Vibro-drill is not allowed. This appendix will end with a conclusion concerning whether or not to adjust the friction coefficient.

B.1. Surface roughness

According to Bogey, Chang and Etsion [3] the surface roughness has a negative effect on the friction coefficient as the true contact area decreases. High stresses will cause plastic deformation in the asperities on a metal surface. A high surface roughness will become smooth due to high normal pressures which are required for the friction force. As more material deforms the true contact area increases and the stress lowers. At a certain stress level an equilibrium develops. The hardness, or yield strength, and contact area of the material determine at what stress the equilibrium is set. As the surface area will be minimized based on the structural force acting on the beam, the stress on the true contact surface will start off higher than the yield strength of the metal. It therefore is expected that the deformation will take place, an increase in the true contact area will happen and the surface roughness will decrease. This process takes place when the module is mounted on the monopile and at this point there will be no vibration running through the system yet. This allows the material to deform and settle before the maximum friction force is required.

Surface roughness is interdependent with material hardness and normal force as these influence the deformations.

B.2. Lubrication

As the Vibro-drill is submerged during installation sea water is expected to immerse the system. Sea water can affect the friction coefficient when it acts as a lubricant in between the two surfaces. For the Vibro-drill the impact of the presence of sea water is less however. This is due to the fact that the Vibro-drill is mounted on the monopile while it is still on the deck of the installation vessel. The contact area, which will deform on local scale while applying pressure, is dry when the material settles. This makes it difficult for penetrate the contact area and reduce the friction coefficient. It is therefore assumed that the lubricating effect of sea water on the friction coefficient can be neglected.

B.3. Normal force

Also from Bogey, Chang and Etsion [3] can be derived that the normal force acting on the surface has a negative effect on the friction coefficient. Even tough the friction force does increase when a greater normal force is applied, the friction force increase with a smaller rate than the normal force. This effect is not significant though to alter the friction coefficient.

B.4. Vibrations

Cited from Broniec and Lenkiewicz [4]: *"External vibration in the contact surface on the static friction joint in the direction of the tangential force decreases the static friction force."*

For the hydraulic cylinder it is important to prevent vibrations along the strike of the cylinder as this might cause unwanted movement. As the piston is restricted in all directions but the strike direction, the alignment should be well thought about. The vibrations are mostly characterized by three parameters which are the frequency, the amplitude and the direction. Both the frequency and amplitude are found to have a negative relation with the the friction coefficient. It should be noted that the frequency range for which the friction coefficient is affected is one order of magnitude higher than the highest frequency the Vibro-drill is designed to operate at. Therefore this is no reason to alter the friction coefficient. The direction is also important as vibrations in the normal direction to the friction surface will reduce the normal force and with that the friction force. When the amplitude of the vibration is in tangential direction it takes up some of the friction force.

B.5. Dynamic friction

As stated, dynamic friction coefficient seems irrelevant as slip should be prevented to keep the module in place. However literature discusses whether in vibrations micro-slip occurs which includes the dynamic friction coefficient in the analysis [2]. Again here the research stated that the frequency is too high for the results to apply for the Vibro-drill.

B.6. Conclusion

From literature it can be concluded that many factor influence the friction coefficient, mostly in a negative way. Due to the characteristics of the Vibro-drill these effects do not apply and the friction coefficient does not need to be altered.