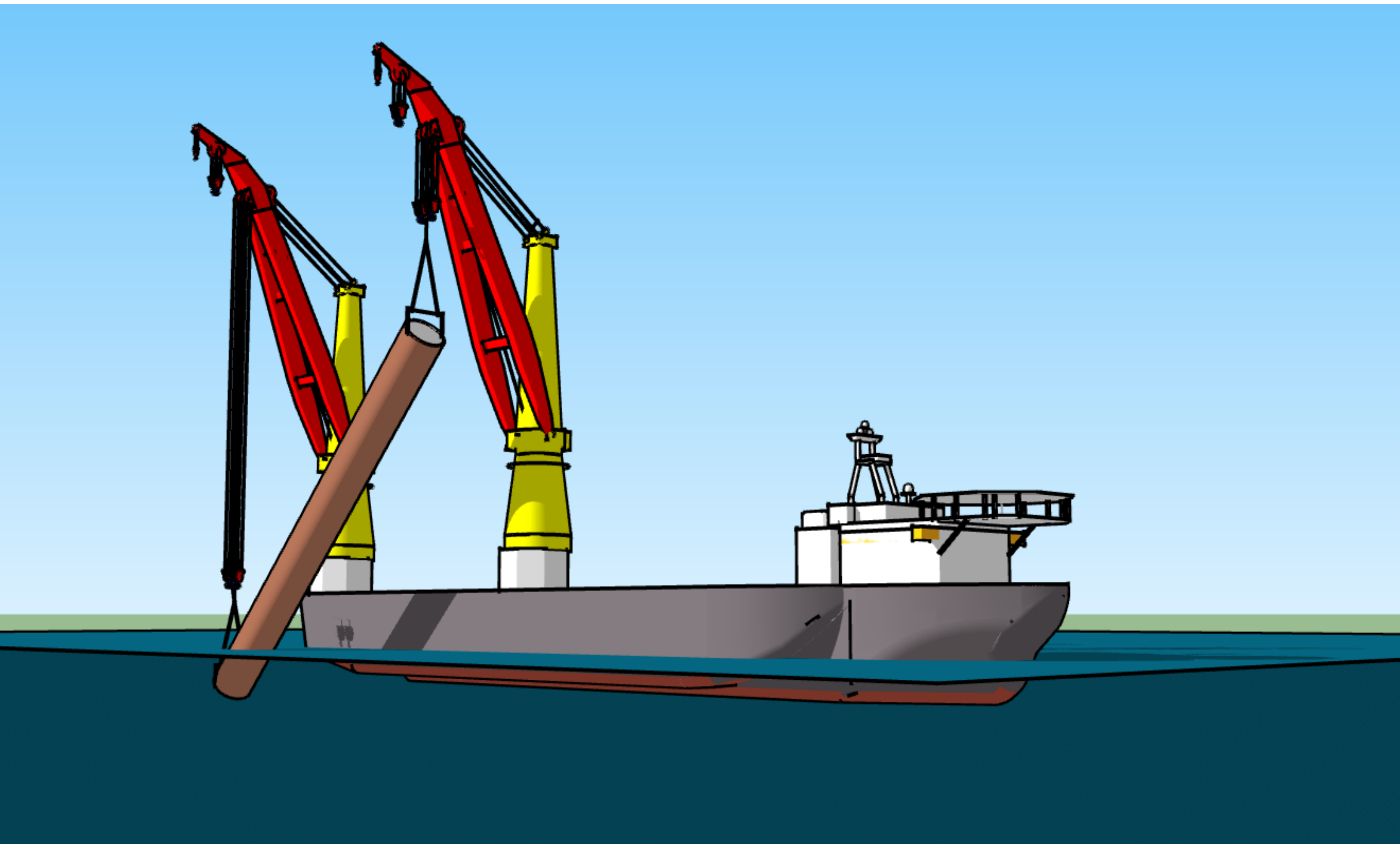

Coupled Dynamics of a Dual Lift Operation

Upending Method of an XXL Monopile

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Boskalis

Coupled Dynamics of a Dual Lift

Monopile Installation

by

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Preface

The last part of my master study, doing research and writing a thesis. I had the privilege to perform this at the company Boskalis. During my time at Boskalis, I became more excited about the contractor side of the offshore industry. The people around me at Boskalis where always very helpfull and interested in the work I was doing. I really would like to thank my supervisor of Boskalis in particular, Panagoitis (Panos) Antonakas. I could not have a better supervisor then him. He awared me to not "get lost in the woods" and gave me helpfull advise about the topic. I would also like to thank the other graduate students at Boskalis for the great time we had.

Working on a project for 9 months solo is a mental challenge as well. I want to thank my friends, family and Francien for the support. Not only during my thesis but also during my study. It was a great period of my life, and I hope I can use the knowledge I gained in my future job.

D.A. Jutten
Delft, June 2019

Abstract

Offshore wind farms moving to deeper waters, implying that larger substructures will need to be installed. The monopile is the leading substructure nowadays and this structure could be promising for use in deeper waters as well. Generally, jack up vessels are used for the installation of the substructures and turbines. However, the jacking up and down of these vessels is time-consuming and limiting with regard to the water depth. A solution to tackle these two drawbacks, is to conduct the installation with a floating vessel. However, this brings along other challenges due to the hydrodynamics of the floating vessel. Therefore, this thesis focuses on the installation of XXL Monopiles with a monohull vessel. An XXL Monopile is a monopile with approximately a length of 120m and a diameter of 10m.

Boskalis is expanding its position on the offshore installation market and has recently added a new heavy lift vessel to its fleet. This vessel is the Bokalift 1 which is a monohull vessel equipped with a crane of 3000 tons lift capacity. To strengthen their position on the installation market, Boskalis is thinking about a new kind of heavy lift vessel which will be an extended version of the Bokalift 1: the Bokalift 2. This vessel will be a monohull vessel as well but equipped with two cranes. The idea is to use this vessel for the installation of XXL monopiles with the use of the two cranes. Boskalis wants to understand the dynamic behavior of the lifted object and vessel during this kind of operation and to know whether the installation with this kind of vessel is feasible or not. This will be assessed by determining the Eigenmodes and corresponding natural frequencies of the system, including the dynamic coupling between the lifted object and the vessel.

The installation of the XXL monopile with the Bokalift 2 is a novel operation. The phase of interest during this installation is the upending of the monopile, which will be executed partly through the splash zone. Using a simplified model, the modes and corresponding natural frequencies are determined for the lifted object in air (no vessel included). This is done with a modal analysis. With a more complete model made in OrcaFlex, these modes and frequencies are verified. The model is expended by including the vessel and subsequently a modal analysis is done. The vessel and lift dynamics are determined for different installation methods and compared. It can be concluded that the so-called "Monopile Yaw" modes are within or near the wave excitation region with their natural frequencies. This requires necessary attention because of the occurrence of unwanted large motions caused by resonance. The most favorable work methods are analyzed using a time domain analysis. Since the submerged part of the monopile during the upending cannot be seen as a slender structure, the Morison equation is not applicable. To run the time domain simulations with the correct hydrodynamics included, the model is improved using a diffraction software AQWA. In this software, the hydrodynamic behavior of the submerged part of the monopile is calculated and these results are implemented in OrcaFlex. With this model, the dynamic coupling between the monopile and vessel is determined and it can be concluded that there is a strong coupling between the monopile's yaw motion and the vessel's roll motion.

Since an XXL monopile is being considered, it was decided to take a look at the piston mode. This is a phenomenon where the water inside the monopile moves up and down. The water column in the monopile will be excited by the waves and this results in a harmonic moving water plug inside the monopile. Since the monopiles are becoming bigger, the water volume is becoming bigger as well and might be of great concern. In previous studies this piston mode has been investigated, but never been applied in a complete model. In this preliminary study, this phenomenon is modeled in OrcaFlex and implemented into the monopile installation model. From the comparison between a model with and without the piston mode it is concluded that, generally, the Piston mode will result in larger responses. However, the approximation of the piston mode phenomena is done by making several assumptions. The coupling between the water in and outside is not considered for example and, therefore, further research is required to gain more knowledge on the effects of the piston mode.

The current thesis provides more insight into the behavior of the lift and vessel dynamics. Based on these results, it is known at which motions attention need to be paid. Also, the coupling between the monopile and vessel is established and could be reduced by the use of tuggers. A novel way of modeling the piston mode is developed and the effect of the piston mode is resolved to be increasing the motions of the system.

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Introduction and Problem Statement

The offshore marine heavy lift sector consists of installing and decommission structures. These structures are getting bigger and heavier. For instance monopiles and jackets. Therefore, larger installation vessels are required (1992, Wouts). Boskalis is developing new crane vessels in order to retain and expand its position in the marine heavy lifting sector. Recently the Bokalift 1 (BL1) shown in 1.1 is added to the fleet. This is a mono hull heavy lift vessel equipped with one crane (3000 ton lift capacity). A next step has already been made by designing a new ship: the Bokalift 2 (BL2).

Figure 1.1: Bokalift 1

The BL2 will be also a monohull vessel but in contrast to the BL1, it will be equipped with two cranes. This different configuration results in novel operations and dynamic behavior. Therefore, it will be interesting to analyze the responses of the object and vessel during a dual lift.

To execute a lift operation safely, it is very important to understand the motions of the vessel and the lifted object. For every vessel and lifted object, the behavior of the system will differ. Since the BL2 will be the first monohull vessel equipped with 2 large cranes for Boskalis, the main question is: how workable it is to do a dual lift with a monohull vessel. To determine the workability, a complete understanding of the dynamics of the complete dynamics of the system (vessel and object) is needed. Eventually, the weight of the lifted object will be considered so the coupled dynamics of the vessel and the object will be quite important (2017, Nam). Therefore, the purpose of this research is to investigate the coupled dynamics during a dual lift operation of the BL2. One of the purposes of the BL2 is to install support structures for wind turbines. These support structures could be jackets or monopiles. In this thesis, the main focus will be on the installation of large monopiles. Which is the predominant substructure of the offshore wind industry (2018, Remy).

Installing a monopile can be done in different ways but the purpose of the installation is always the same; put the monopile into the seabed in an upright position. As mentioned before, the monopiles are getting bigger and heavier and this requires different installation methods. The installation of a monopile consists of

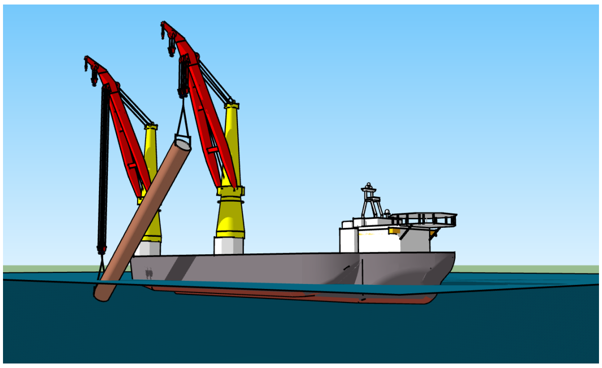

Figure 1.2: Sketch of BL2 upending a monopile

different phases. These phases could be: transporting, move monopile from horizontal to a vertical position, position the upended monopile in the gripper, lower the monopile and drive the monopile into the seabed. This research is executed regarding to move the monopile from a horizontal position to a vertical position shown in figure 1.3, also known as the upending of the monopile.

Figure 1.3: Upending monopile (2016, IHC)

1.1. A brief introduction on the upending methods

Upending the monopile can be done in different ways. For instance, relative short monopiles (max 30m) will be upended on deck using special equipment and will enter the water surface in a vertical position. When the monopile is relatively long (read 80 – 120 m) upending becomes more complex due to the relative compact workspace on deck. Due to the great lift capacity of the BL2, there is a possibility to upend the monopile without an upending frame. The upending will be done using the two cranes. Several ways to upend a monopile with the BL2 are explained in the following section.

1.1.1. Upending above deck

Upending the monopile above deck with two cranes is a work method of the BL2. In figure 1.4, a sketch is shown how the upending above deck will be done. An advantage of this method is the limiting roll motions caused by the lifted load. Since the monopile will be upended above deck, the heeling moment of the vessel will be restricted during this phase. However, there will be a moment where a mass of approximately 2000 is ton hanging above the deck which is a considerable risk. Since the monopiles will become longer, the max height of the cranes could be reached. To avoid upending above deck, upending through the water line (splash zone) is an option. This is explained in the next section.

1.1.2. Upending through splash zone

Another way to up end a long monopile is to use two cranes as shown in figure 1.5. The monopile is positioned on the long side of the vessel and upended with the two cranes. During the upending, the monopile will be moved in different positions and will go through the splash zone to get finally in a vertical position. Lifting through the splash zone is a complex phase of lift operations (2013, Gordon). At every position, the dynamics of the system will differ. The dynamics of the vessel and object are required to determine if there is coupling (2009, Ibrahim) and to check if the limits of the crane are reached or not during a certain condition.

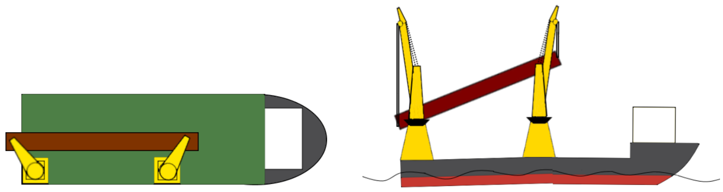

Figure 1.4: Upending above deck

This limits are the off and side-lead angles at the crane tip and are 4 degrees. The drawings are some rough sketches to get a good understanding.

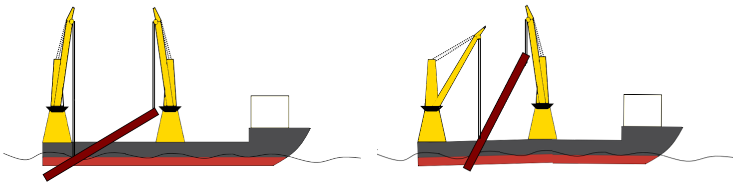

Figure 1.5: Longitudinal upending

Understanding the upending of the monopile through the splash zone is really important because in throughout similar operation large complications were experienced when the lifted object entered the splash zone. During the up ending the monopile will be partly submerged under an angle. Different forces are acting on the submerged part of the monopile and have influence on the motions of the system. An explanation of these phenomena will be given in later section of this chapter.

1.2. Research scope

The BL2 will be a new kind of heavy lift vessel in his class. With at least a configurations with 2 cranes of 3000 ton lift capacity, the installation of large monopiles is possible, specifically the upending of the monopiles as mentioned earlier. During this research the work method where the monopile will be upended long side the vessel will be analyzed. This work method can be executed at different manners, an objective is to find the most favorable work method for upending through the splash zone.

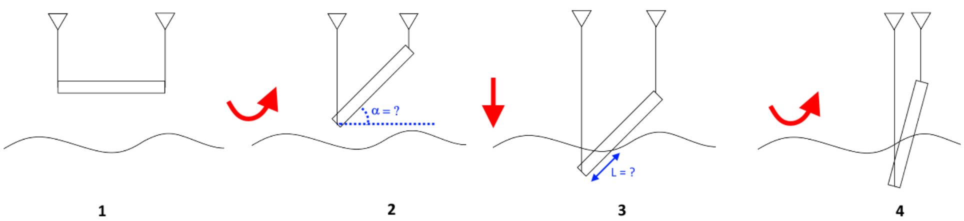

Figure 1.6: Research scope

All methods will be executed using two cranes, where for example the ”angle α ” or ”draft L” are the variables as shown in figure 1.6. The system will be consisting of the vessel the monopile. The outcome of this analysis will be the critical modes and frequencies. When a favorable work method is determined, this work method will be analyzed by doing time domain simulations and identify critical motions. When these motions are known, solutions to mitigate these motions will be recommend.

1.3. Description of the approach

The approach of the research will be step wise. In other words, a simplified model will be made first to become fully known with the problem. When there is enough knowledge about the situation, an extra phenomenon or aspect is added to the system. Finally, an acceptable model will be the result to calculate the motions of the system during a certain sea state. In the following section the steps are described.

1.3.1. Simplified model in air

As mentioned before, the purpose of the beginning of the thesis is to completely understand what is happening during a dual lift operation. Therefore, first the monopile will be seen as a free hanging beam in air where the vessel is considered fixed. For example, a simplified system is shown in figure 1.7. This system will have

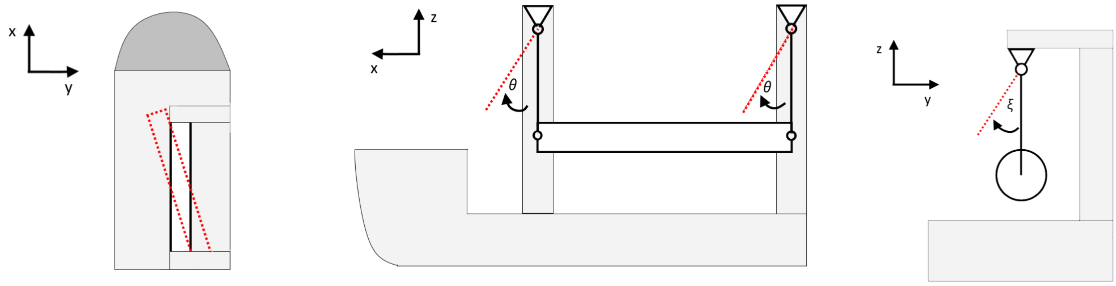

Figure 1.7: Example simplified model

N degrees of freedom. Natural frequencies and Eigenmodes will be determined with a modal analysis using Python. The system will consist of second order differential equations. The number of differential equations will depend on the degrees of freedom of the system. N degrees of freedom will result in N differential equations.
In the first phase of the approach the vessel is assumed to be fixed. Basically the simplified system is a variant of a pendulum. The equation of motion will look more like:

$$ml\ddot{\theta} + mgsin(\theta) = M \tag{1.1}$$

Where: m = mass monopile, $\ddot{\theta}$ = acceleration, l = Lenght of the rope/cable, θ = rotation, M = force/moment acting on the system

Solving the system will result in a very basic first impression. However, this already gives more insight in the behavior of the system.

1.3.2. Introducing Orcaflex

To approach the problem more accurate, a model will be made in Orcaflex. OrcaFlex is the world's leading package for the dynamic analysis of offshore marine systems. With Orcaflex the model will be expand so the following steps can be executed:

- Verify the simplified model
- Add vessel (Bokalift 2) to the system
- Determine coupling

When there are any problem regarding to the Orcaflex model, it is always possible to fall back to the basics of the problem as explained in the previous section.

Equation (1.2) is a general form of an equation of motion for ship hydrodynamics and will be the basis of the calculations.

$$(m+a)\ddot{x} + b\dot{x} + cx = F \tag{1.2}$$

Where: m = mass, a = added mass, $\ddot{x}$ = acceleration, b = damping, $\dot{x}$ = velocity, c = stiffness, x = displacement, F = forces acting on the system

1.3.3. Including splash zone part

If the the previous part is understand well, the next step can be made and the focus will be on upending the monopile through the splash zone. Extra aspects have to be taken into account in the model (2010, Sarkar). The forces on the monopile cause by the water. The monopile will be partly submerged and this part will be subjected to static and dynamic forces of the water. The new aspects that have to be taken into account are summarized below

1. **Hydrodynamic loads submerged part monopile**
   Orcaflex is not calculating the hydrodynamics on structures correctly in this particular case. To have a good estimation of the hydrodynamic loads, diffraction software AQWA is used to determine the added mass and response operators of the submerged part.

2. **Piston Mode Phenomena**
   Piston mode resonance is about the moving water inside of the monopile. Since in this thesis an XXL-Monopile is considered, it might be of XXL concerns. The water inside a 20m submerged part of this kind of monopile has already a mass of approximately 1570 tons.

1.4. Research objective

The objective of the research is to get a better understanding of the dynamics during a dual lift operation. This research is focusing in particular on the upending of an XXL- Monopile with a dual lift. To achieve this objective the following research question is formulated:

What would be the coupled dynamics of the vessel and monopile during the upending with a dual lift under various work methods?

The research question will be answered using the previously described approach. From basic to complex. During this approach the following sub questions will be answered:

- *What are the modes of the monopile in air?*

  During the first phase of this thesis, the focus will be on understanding the lift dynamics. A begin of solving this sub question will be done during this phase. The system can be seen as a rigid bar hangin on two constrains.

- *What are the modes and natural frequencies of the lift and vessel dynamics?*

  During the second step of the approach the vessel is added to the system with use of OrcaFlex. This affects the natural periods calculated in the first part.

- *Which work method is the most favorable in terms of inclination and draft?*

  With modal analysis the different work methods are analyzed. The inclination of the monopile before it will enter the splash zone and different drafts are researched. Comparing the natural frequencies with the wave excitation frequencies will give an impression about favorable and not favorable.

- *How does the monopile affects the motions of the vessel and vice versa, where and how storng is this coupling?*

  With use of time domain simulations the favorable work methods from previous part are analyzed. By comparing the solutions between the use of displacement and force RAO's, coupling will be determined. After this section it is clear where the coupling occurs and how strong this coupling is.

- *Is it possible to model the piston mode in Orcaflex, and how much does this phenomena affects the dynamics of the system?*

  With the use of preliminary research, the piston mode is modeled in OrcaFlex. Time domain simulations are ran to understand how much this affects the system. In the end it will be clear if the piston mode phenomena is of great concern or not.

1.5. Thesis Outline

The outline of the thesis is as follows:

- **Chapter 2:** This chapter is about the simplified model, the lift dynamics. It is explained how the simplified model is made and the difference between linear and non linear solutions are explained. This part only considers the lifted object, there is no vessel included in this chapter.
- **Chapter 3:** The third chapter of the thesis is about the lift and vessel dynamics. The simplified model of chapter 2 is verified and the vessel is added to the model. With use of OrcaFlex, modal analysis is done for different set ups and coupling is determined.
- **Chapter 4:** In chapter four the model of chapter 3 is expanded with an extra phenomena that occurs, Piston Mode. This phenomena is implemented and the affect of this is discussed.
- **Chapter 5:** Chapter 5 is about to determine the required crane capacity and the workability of the upending method.
- **Chapter 6:** In chapter 6 the conclusions about the dynamic behavior of the system are presented and recommendations are given.

2
Lift Dynamics

The main purpose of this thesis is to understand the coupled dynamics of a dual lift with a monohull vessel, with the focus on the upending of an XXL monopile. To understand the coupled dynamics, a basic start will be made in this chapter by using a stepwise approach. The structure of this chapter is shown in figure2.1. In this chapter, the goal is to obtain a first impression of the natural frequencies and the eigenmodes of the lifted object (XXL monopile). These natural frequencies are important to know in order to check if the natural frequencies will match the excitation frequencies or not. If the frequencies will match, resonance will occur and this is not favorable.

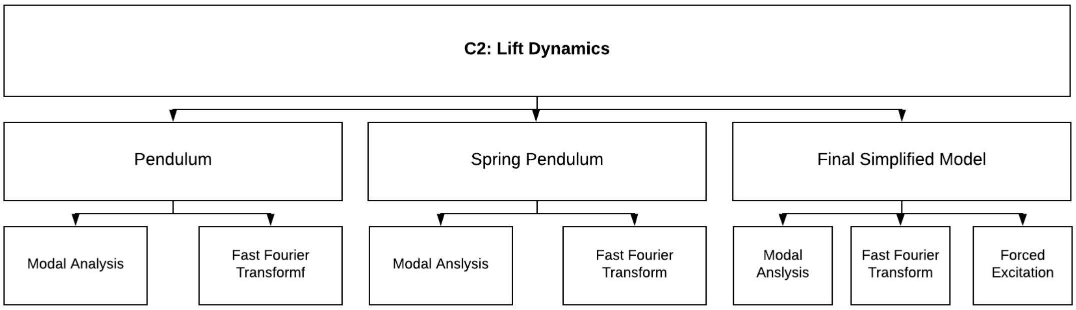

Figure 2.1: Chapter 2 structure

This chapter has three main parts which are called: Pendulum, Single Pendulum, and Final Simplified Model. Every part will consist of a Modal Analysis where, with the use of a decay test, a Fast Fourier Analysis will be executed. The last part (Final Simplified Model) will also consist of an additional section where the system is excited by crane tip motions. At the end of this chapter, a first impression about the natural frequencies is obtained. The steps during this chapter are shown in figure 2.2

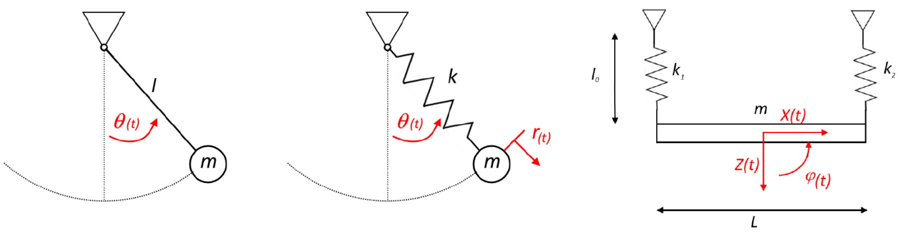

Figure 2.2: Models considered this chapter

The models will be solved analytically, linear and nonlinear. To solve this kind of dynamic systems, it is common to apply linearization because of its simplicity and increase calculation speed. The effect of linearization will be explained in this chapter. To understand how the system behaves, the equations of motion need to be solved. All the equations of motions in this chapter are formed by using the Euler- Lagrangian equation (2.1). This is based on the classical mechanics. The Euler - Lagrangian equation is a second order partial differential equation whose solutions are the function for which a given functional is stationary.

$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial L}{\partial \dot{q}} - \frac{\partial L}{\partial q} = 0 \tag{2.1}$$

Where:

$$L = T - V \tag{2.2}$$

During the following sections the Lagrange function L (2.2) will be given where T is the kinetic energy and V will be the potential energy. To create the plots and calculate the natural frequencies the same parameters are used unless mentioned otherwise. These parameters are based on an XXL- Monopile and presented in table 2.1.

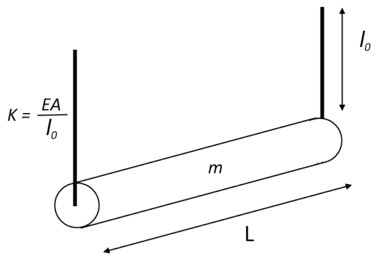

Figure 2.3: Definition parameters

L	90	[m]
g	9.81	[m/s^{2}]
m	2000	[ton]
l_0	40	[m]
EA	$3*10^6$	[kN]
$k_{1,2}$	7500	[kN/m]

Table 2.1: Parameters (2018, Boskalis)

Since the focus is on the upending of an XXL monopile, the final simplified model will consist of a rigid body (with the characteristics of this XXL monopile) constraint with two springs. These springs are the cables/tackles of the crane, connected to the rigid body and the possibly moving crane tips. This system can be approached as a combination of a pendulum and a spring-mass system. Consequently, the pendulum and spring pendulum are described and discussed in this chapter. An extra intermediate step is a spring, rotational spring pendulum, this step is explained in Appendix A.

2.1. Pendulum

The basics of the dynamics are explained briefly in this part. Initially, a single pendulum as shown in figure 2.4. During this part, the differences between the linear and non-linear approaches are discussed and the natural frequency of this kind of system is defined analytically and with a Fast Fourier Transform (FFT).

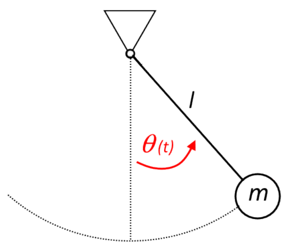

Figure 2.4: Single Pendulum

2.1.1. Modal Analysis

Before solving these systems, an important note is that the systems can be solved linear and nonlinear. Namely, approach the system with the idea that only small displacements will occur or not (2006, Klarbring). To give an idea about the effect of solving a system linear on nonlinear, the two solutions will be compared for the different systems. With this comparison, the effect on the (natural) frequency can be seen due to nonlinearity. "Natural" is put in brackets in the previous sentence because a nonlinear solution does not have one natural frequency, it depends on the amplitude of the motion.

Linear

With a linear approach is meant that non linear terms are linearized or neglected. The general reason this is done, is for the simplicity of the calculations. The Lagrange function (2.2) for this pendulum is as follows:

$$L = \frac{1}{2}m(l*\dot{\theta})^2 - mgl(1 - cos(\theta)) \tag{2.3}$$

where:

m = Mass [kg]

l = Length of the rigid massles bar [m]

θ = Rotation of the pendulum [rad]

$\dot{\theta}$ = Rotational speed of the mass [rad/s]

g = Gravitational acceleration [m/s^2]

When equation (2.3) is implemented and differentiated in equation (2.2), where $q = \theta$, the equation of motion will be the result. The way of linearization at this particular system is to assume that only small displacements will occur and the sin and cos functions are approximately: $\sin(\theta) \approx \theta$ and $\cos(\theta) \approx 1$. There is one degree of freedom is this system ($\theta(t)$). Hence, there will be also one equation of motion. The equation of motion is:

$$ml^2\ddot{\theta} + mgl\theta = 0 \tag{2.4}$$

The natural frequency of this system can be calculated with $\omega = \sqrt{\frac{g}{l}}$. However, for coupled and more complex systems the natural frequencies need to be defined with another method. More about these method (Fast Fourier transform) in the following section. To have an idea about the behavior of the system, an initial displacement is given. The motion of the system is displayed over time in figures 2.5 and 2.6 due to small and large initial displacements respectively.

ω_θ	0.495	[rad/s]
T_θ	12.69	[s]

Table 2.2: Theoretically calculated natural frequency

Non-Linear

In this section, the assumption of small displacements is not applicable and the sin and cos terms are kept. When this is done the following Lagrange equations and equation of motion are the results:

$$L = \frac{1}{2}m(l*\dot{\theta})^2 - mgl(1 - cos(\theta)) \tag{2.5}$$

$$ml^2\ddot{\theta} + mglsin(\theta) = 0 \tag{2.6}$$

With figures 2.5 and 2.6 the differences between the linear and non-linear solutions are displayed. However, when the system has a small initial displacement, there is no much difference. When the initial displacement is large, different behavior is noticed. This will result in a different natural frequency, more about this in the section about the Fast Fourier Transform (FFT).

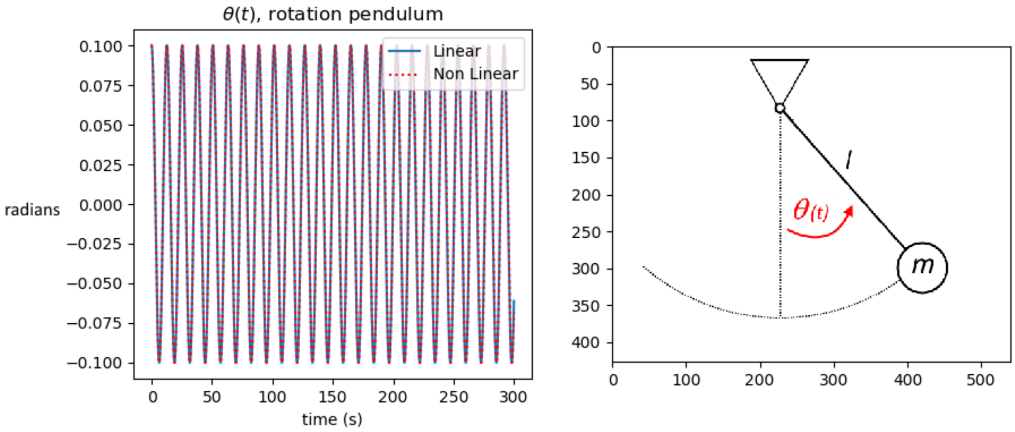

Figure 2.5: Motion of the linear and non-linear pendulum due to a small initial displacement ($\theta_0 = 0.1rad$)

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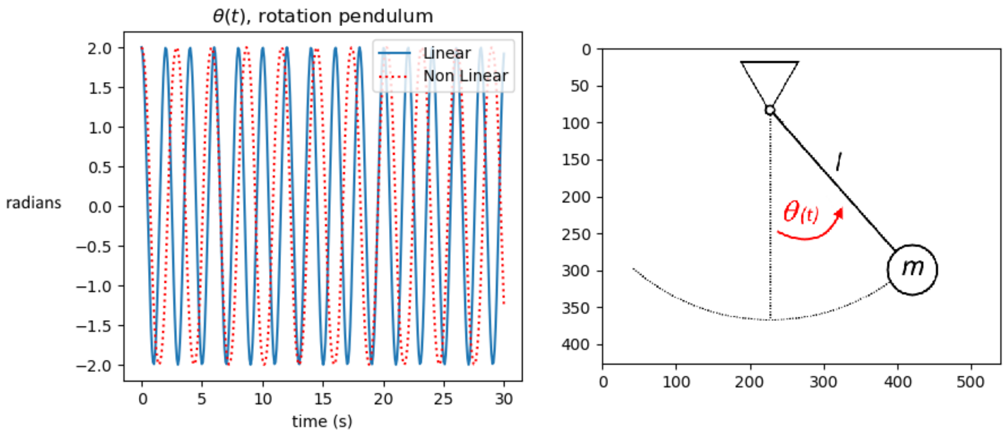


Figure 2.6: Motion of the linear and non-linear pendulum due to a large initial displacement ($\theta_0 = 2rad$)

2.1.2. Fast Fourier Analysis

In the previous section the derivation and solution of the single pendulum are explained. In this part, the natural frequency of the pendulum will be determined by the use of the FFT. The FFT is an algorithm that improves the efficiency for computing the discrete Fourier transform (2012,Vanderbei). With this transform, a signal in the time domain is decomposed to the frequency domain. It is known how to find the natural frequency analytically for this particular system, but for the understanding of the other systems, it is performed with an FFT as well.

The FFT in figure 2.7 is from the solution that is presented in figure 2.5. In this case, the system is given a small displacement, and both solutions are not differing significantly. This can also be seen in the FFT in 2.7. The both peaks in the FFT is at a frequency of $0.5rad/s$. This is, as expected, corresponding with the calculated natural frequency $0.495rad/s$ ($\omega_\theta = \sqrt{\frac{g}{l}}$).

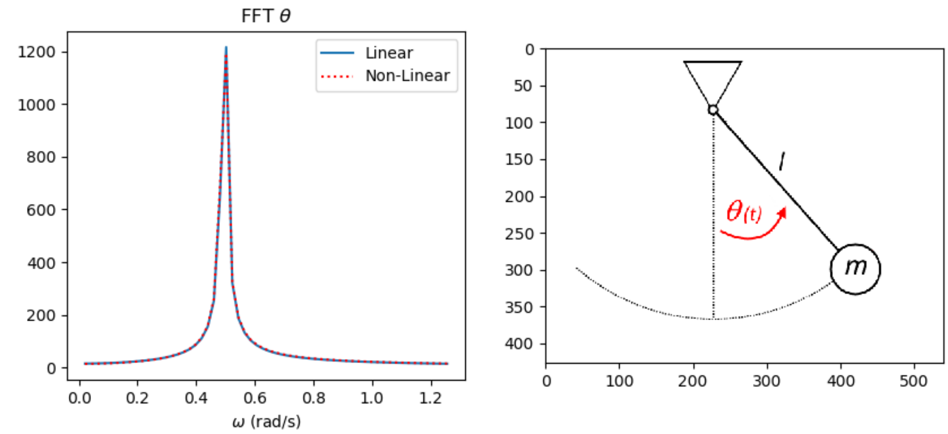


Figure 2.7: FFT - Lin and Non -Linear due to a small initial displacement($\theta_0 = 0.1rad$)

Figure 2.8 shows the FFT from the case where the system is given a large initial displacement.A difference between the linear and nonlinear solution was already noticed, but in the FFT it appears obvious. The peak of the nonlinear solution is shifted with respect to the peak of the linear solution. The results of the natural frequencies are summarized in table 2.3.

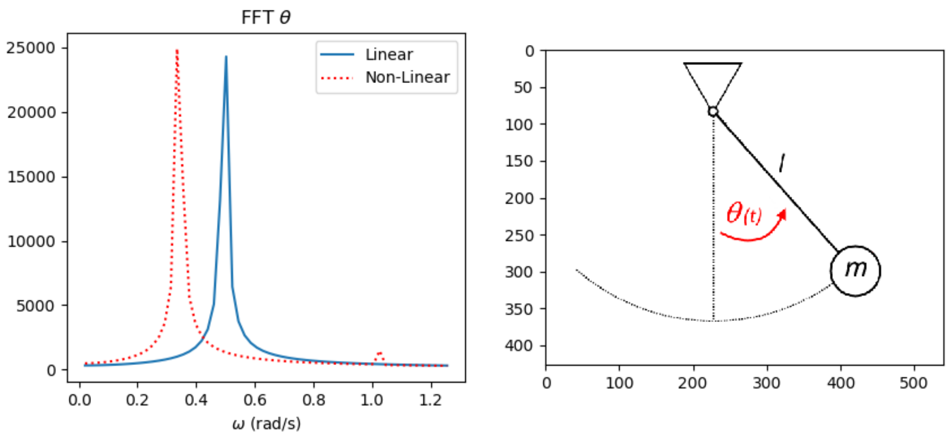


Figure 2.8: FFT - Lin and Non -Linear due to a large initial displacement ($\theta_0 = 2rad$)

	Linear	**Non-Linear**	
ω_θ	0.495	0.356	[rad/s]
T_θ	12.69	17.65	[s]

Table 2.3: Comparison between natural frequencies

The "large" initial displacement is extremely large with regarding the small initial displacement and results in a large shift of the peak in the FFT. To see how the peak shifts, in figure 2.9, the FFT for more initial displacements is given. In this plot, two things are remarkable, the shift of the peak and the second small peak in the FFT. Both appearances are because of the nonlinearity. The second peak is because of the higher order effects (2018, Suzuki). During this thesis, the focus will not be on these effects but it is good to understand where these peaks come from so in the later phase the origin of appearing peaks is understood.

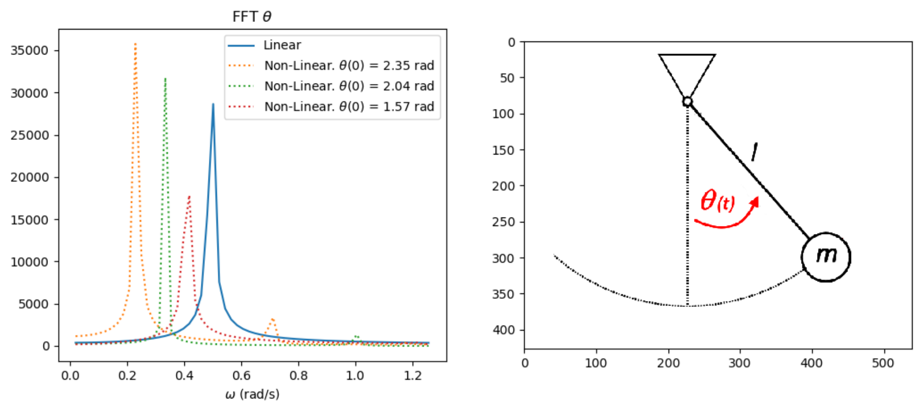


Figure 2.9: FFT - Multiple initial displacements

2.2. Spring Pendulum

The next step is adding an extra degree of freedom that simulates the elasticity of the hoist cable. This is done by replacing the massles rigid bar for a spring. Now the system can rotate ($\theta(t)$) and elongate($r(t)$) as shown in figure 2.10. In table 2.4 the natural frequencies and periods are given calculated using the equations: $\omega_\theta = \sqrt{\frac{g}{l}}$ and $\omega_r = \sqrt{\frac{k}{m}}$.

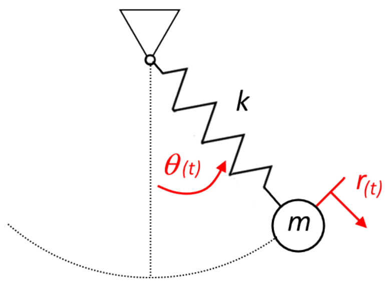


Figure 2.10: Spring-Pendulum

ω_r	8.66	[rad/s]
ω_θ	0.495	[rad/s]
T_r	0.75	[s]
T_θ	12.66	[s]

Table 2.4: Theoretical calculated natural frequencies and periods

2.2.1. Modal Analysis

As in the previous section, a linear and nonlinear solution will be given in this section. Because of the extra degree of freedom, the Lagrangian (2.7) and the equations of motion become more complex.

Linear

The lagrangian and the equations of motion are given below:

$$L = \frac{1}{2}m(\dot{r} + (l+r)\dot{\theta})^2 - \frac{1}{2}kr^2 + mg[(l+r)cos(\theta) - l] \tag{2.7}$$

Linearized equations of motion:

$$\ddot{r} + \frac{k}{m}r - g = 0 \tag{2.8}$$

$$\ddot{\theta} + \frac{g}{(l+r_s)}\theta = 0 \tag{2.9}$$

Where r_s is the static elongation ($\frac{mg}{k}$) and l the unstressed length of the spring. In figure 2.11 and figure 2.13 the difference between the linear and the nonlinear approach when a small initial displacement is given to the model is displayed.

Non-Linear

There is not only non-linearity because of the sin and cos terms in the Lagrange equation, but also because of the multiplication between the time dependent degrees of freedom ($r(t)$ and $\theta(t)$.

$$L = \frac{1}{2}m(\dot{r} + (l+r)\dot{\theta})^2 - \frac{1}{2}kr^2 + mg[(l+r)cos(\theta) - l] \tag{2.10}$$

The system has two degrees of freedom, $\theta(t)$ and $r(t)$, therefore, the following two equations of motions are the result of the Lagrangian derivation.

$$m[\ddot{r} - (l+r)\dot{\theta}^2] = mgcos(\theta) - kr \tag{2.11}$$

$$m[(l+r)\ddot{\theta} + 2\dot{r}\dot{\theta}] = -mgsin(\theta) \tag{2.12}$$

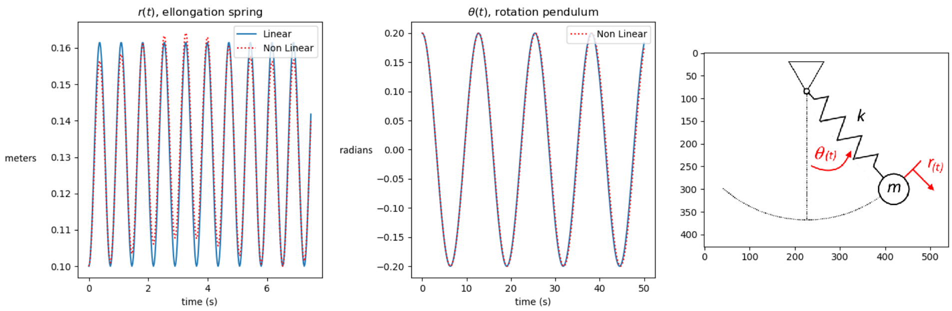


Figure 2.11: Motions of linear and non linear spring pendulum $r_0 = 0.1m, \theta_0 = 0.2rad$)

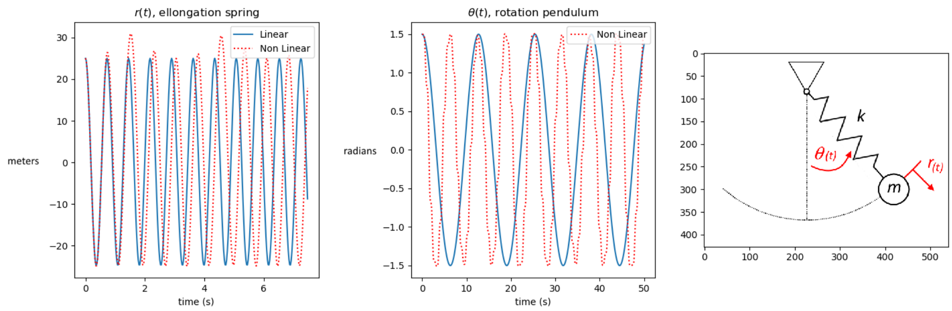


Figure 2.12: FFT of linear and non linear spring pendulum ($r_0 = 25m, \theta_0 = 1.5rad$)

2.2.2. Fast Fourier Analysis

For the solutions presented in figures 2.11 and 2.12 a FFT is done as well. The main difference with the previous system is the extra degree of freedom. This results not only in two natural frequencies but also in the possibility of coupling between the Degrees Of Freedom (DOF).

The first priority is to find natural frequencies of the discussed models. Natural frequencies give insight at which exciting frequency the system will respond the most. However, with every natural frequency comes a mode. At the pendulum of previous part, this mode is the rotation θ. This system has only one degree of freedom. When a system has more degrees of freedom, like the current spring pendulum, the mode corresponding with one of the natural frequencies can consists of a combination of different degrees of freedom. This means that it is a coupled mode.

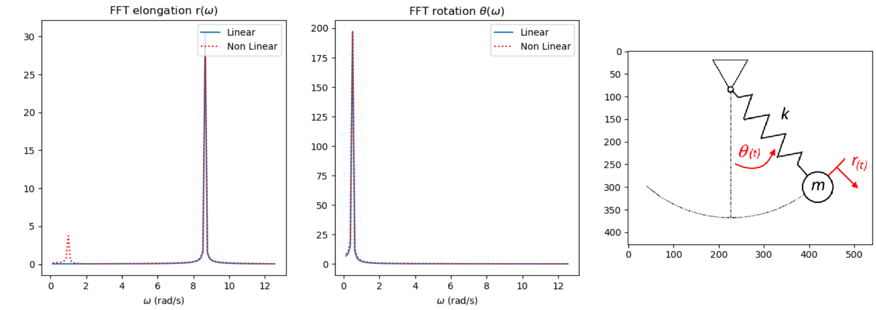


Figure 2.13: FFT of linear and non linear spring pendulum ($r_0 = 0.1m, \theta_0 = 0.2rad$)

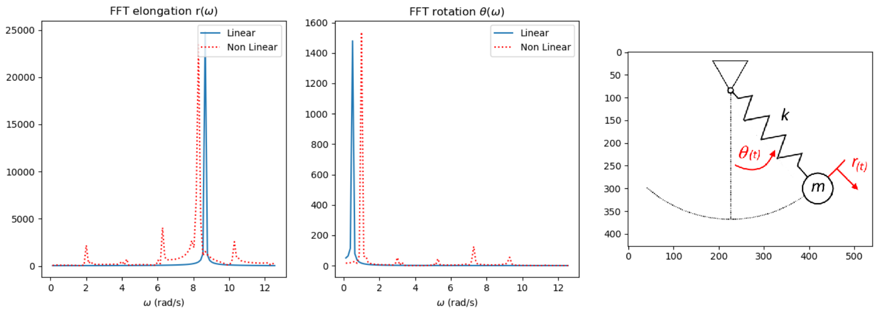


Figure 2.14: FFT of linear and non linear spring pendulum ($r_0 = 25m, \theta_0 = 1.5rad$)

In figure 2.13 the origin of the small peak in the FFT of the elongation is because of the coupling between the two degrees of freedom. The small peak is exactly at the natural frequency of the rotation DOF.

The peaks noticed in figure 2.14 are because of the earlier mentioned higher order harmonics. The non-linear effects are appearing in this FFT. Differences in natural frequencies and periods between the linear and nonlinear solutions are presented in table 2.5.

	Linear	**Non-Linear**	
ω_r	8.66	8.29	[rad/s]
ω_θ	0.495	1.00	[rad/s]
T_r	0.75	0.76	[s]
T_θ	12.69	6.26	[s]

Table 2.5: Natural frequencies comparison

2.3. Final Simplified Model

Now the basic systems are explained, the step to the simplified model is made. As explained earlier in this section this simplified model contains a rigid body (the monopile), two springs (the tackles) and to exited ends of the two springs (the crane tips). The solution for this system is solved exactly the same as the other systems, by solving the Lagrangian. Also here both solutions are given and compared. To find the natural frequencies of certain modes, the basic analytical solutions are used from the earlier section and fast Fourier transforms are executed. Additional, another method is used to define the frequency at which the mode will response the most (resonance). This is realized by exciting the system at the crane tips with different frequencies. The maximum displacement per exciting frequency is plotted. A peak will occur at the natural frequency of the mode.

2.3.1. Modal Analysis

The final simplified model is shown in figure 2.15. There are two cross sections given where all the five degrees of freedom are shown. The sixth degree of freedom of the monopile (roll, rotation around the X-axis) is not considered.

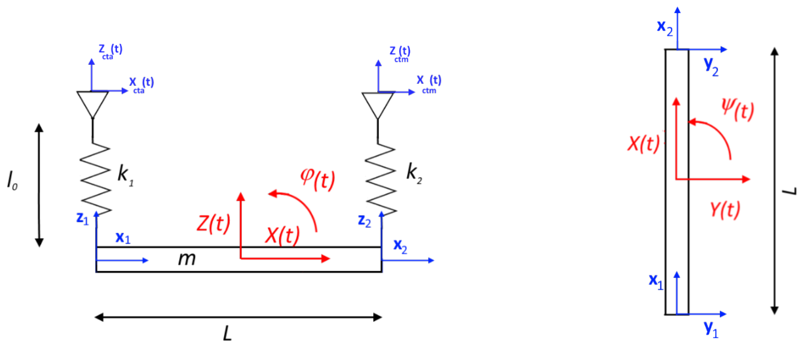


Figure 2.15: Simplified model

The Lagrangian equation in this case consists of the kinetic energy of the rigid body, and the potential energy due to the elongation of the springs and the gravity.

$$L = \frac{1}{2} m\ddot{X}^2 + \frac{1}{2} m\ddot{Y}^2 + \frac{1}{2} m\ddot{Z}^2 + \frac{1}{2} J\ddot{\varphi}^2 + \frac{1}{2} J\ddot{\psi}^2 - \frac{1}{2} k(\Delta u_1)^2 - \frac{1}{2} k(\Delta u_2)^2 + mgZ \tag{2.13}$$

where:

$$\Delta u_1 = \sqrt{(l+z_1)^2 + x_1{}^2 + y_1{}^2} - l$$
$$\Delta u_2 = \sqrt{(l+z_2)^2 + x_2{}^2 + y_2{}^2} - l$$

The $x_{1,2}$, $y_{1,2}$ and $z_{1,2}$ are local coordinates on the left and right end of the monopile as shown in figure 2.15 and can be expressed in the five degrees of freedom: $X(t)$, $Y(t)$, $Z(t)$, $\varphi(t)$ and $\psi(t)$.

$$x_1 = X(t) - \frac{1}{2}\cos\left(\psi(t)\right)\cos\left(\phi(t)\right)L$$
$$x_2 = X(t) + \frac{1}{2}\cos\left(\psi(t)\right)\cos\left(\phi(t)\right)L$$
$$y_1 = Y(t) - \frac{1}{2}\sin\left(\psi(t)\right)\cos\left(\phi(t)\right)L$$
$$y_2 = Y(t) + \frac{1}{2}\sin\left(\psi(t)\right)\cos\left(\phi(t)\right)L$$
$$z_1 = Z(t) + \frac{1}{2}\sin\left(\phi(t)\right)L$$
$$z_2 = Z(t) - \frac{1}{2}\sin\left(\phi(t)\right)L$$

The equations of motion, resulting from solving the Lagrangian, are executed with the use of Maple. The system is given initial displacements to analyze the behavior, also known as a decay test. To find the natural frequencies per mode, an FFT is performed in the next section. The motions as a result of the initial displacements, are shown in figure2.16. In this figure, the two solutions are presented. The Nonlinear solution is the exact solution of the problem, and a linear solution is an approach by using the spring, rotational spring, pendulum model. This particular intermediate model is described in appendix A and is the linearized decoupled approximation of this system.

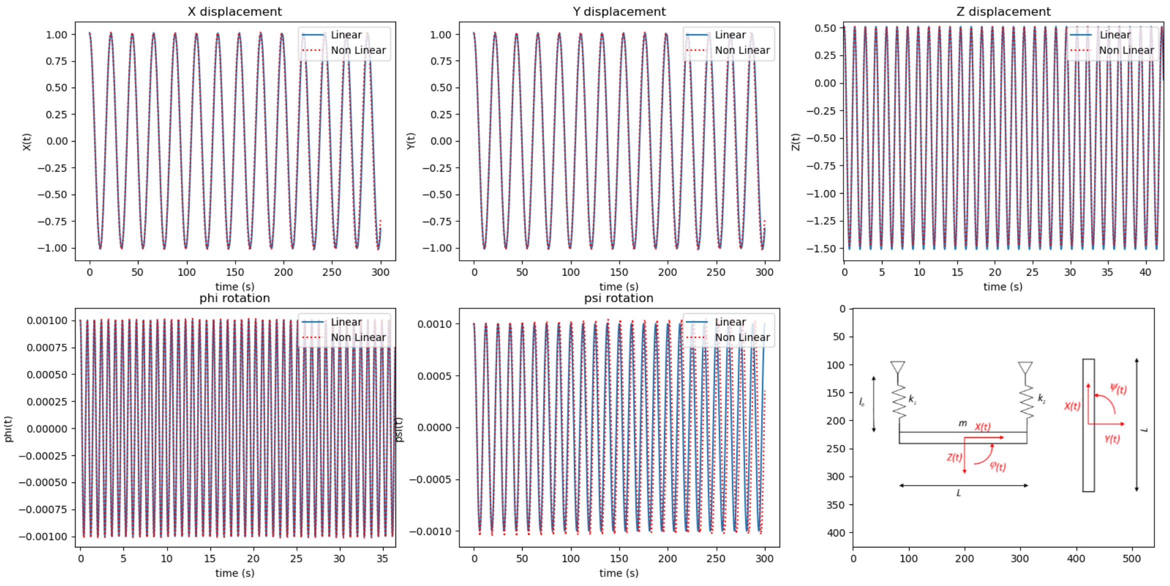


Figure 2.16: Motions- x(0) = 1m, y(0)= 1m, z(0)= 1m, $\phi(0)$ = 0.001rad and $\psi(0)$= 0.001rad

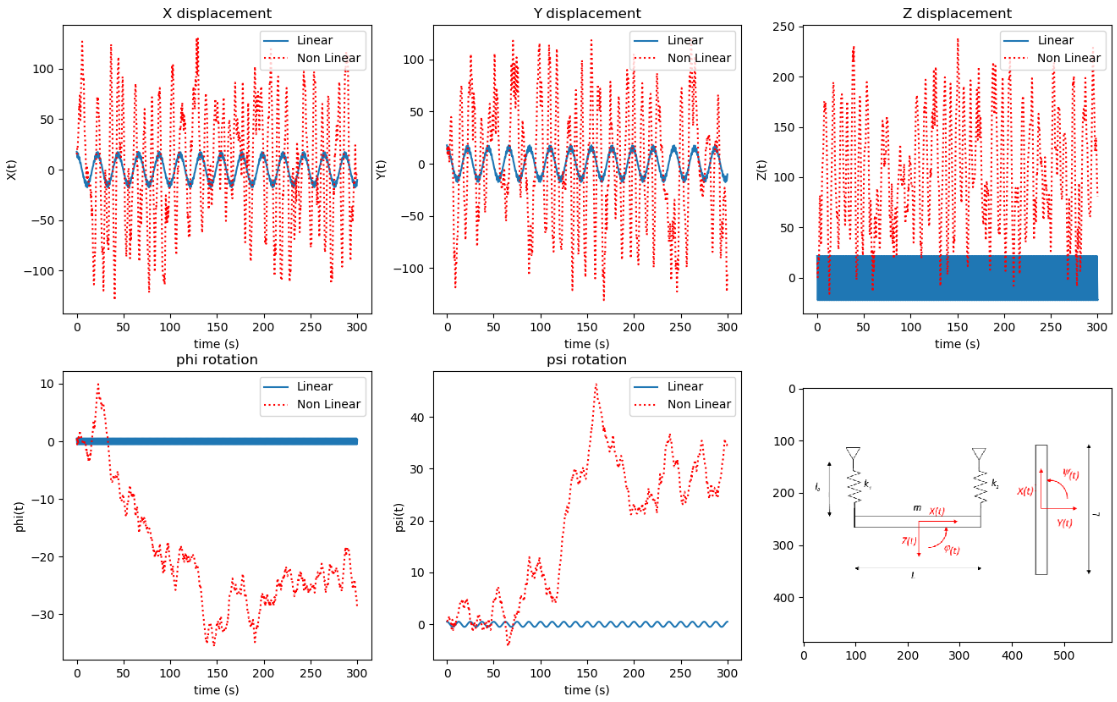


Figure 2.17: Chaotic Motions- x(0) = 15m, y(0)= 15m, z(0)= 15m, $\phi(0)$ = 0.5rad and $\psi(0)$= 0.5rad

The motions in figure 2.16 are the solutions due to the initial conditions given in the title of the figure. If the initial conditions are becoming larger, at a certain point the non-linaer solution is becoming chaotic (2002, Thompson). This is presented in figure 2.17. When the solution is chaotic it is not usable anymore. However, the initial conditions to reach chaos are much larger than the limits of a real lift operation. For example, with a X(t) motion of 15 meters, the sidelead angle will be 7 degrees. The limit for the sidelead angle is 4 degrees (2017,Boskalis). When the limits are used as the initial displacement the system behaves not chaotic. This can be seen in figure 2.18. The limits that are considered are the off-lead and side-lead angle of the tackles at the crane tips.

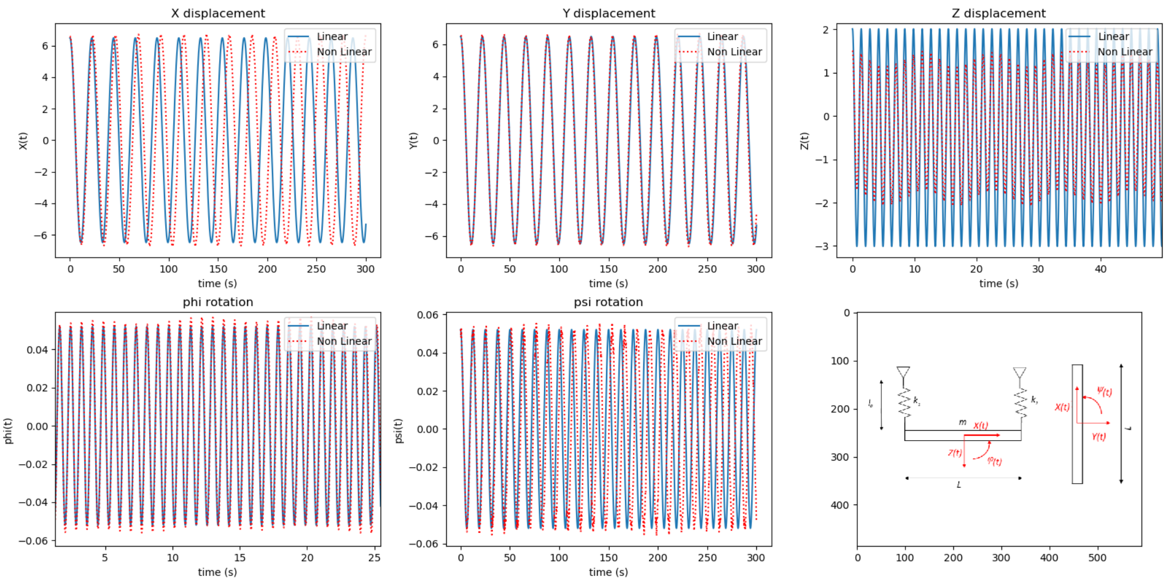


Figure 2.18: Motions with Limits as initial values x(0) = 6.5m, y(0)= 6.5m, z(0)= 2m, $\phi(0)$ = 0.052rad and $\psi(0)$= 0.052rad

2.3.2. Fast Fourier Analysis

As performed in the previous sections, a FFT is performed at the motions given in figure 2.16 as well. With this FFT the different natural frequencies are determined. The FFT of the linear and non linear solutions are displayed in the same figure, and are very similar. The analytical calculated natural frequencies calculated with the known equations are plotted as green vertical dotted lines.

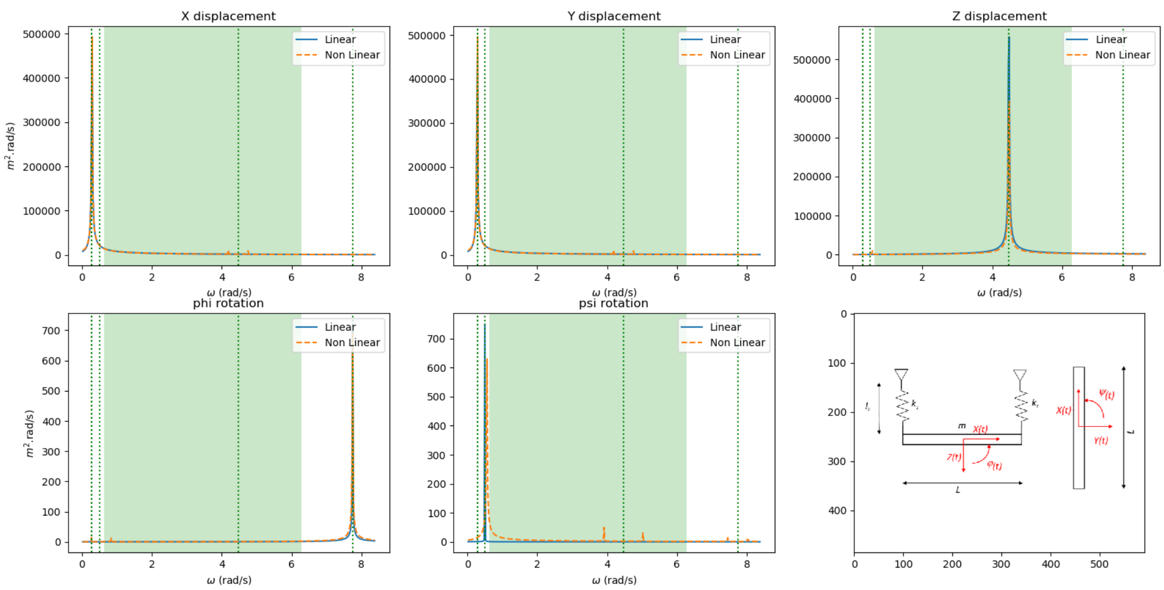


Figure 2.19: FFT - x(0) = 1m, y(0)= 1m, z(0)= 1m, $\phi(0)$ = 0.001rad and $\psi(0)$= 0.001rad

2.3.3. Forced Excitation

Another way to determine the natural frequencies is to give the system a forced excitation at the crane tips. This excitation is based on the Response Amplitude Operators of the vessel due to a certain incoming wave (vessel still not included in model). By taking the maximum displacement as result of the different sea state, a

	X	Y	Z	phi	psi
ω_0 (rad/s)	0.28	0.28	4.47	7.75	0.50
T (s)	22	22	1.4	0.8	12.5

Table 2.6: Natural frequencies and periods

line can be plotted and a peak is expected around the natural periods of the eigen modes. In figure 2.20 these lines are plotted for different wave heights, what results in a shift of the peak because of the non linearity.

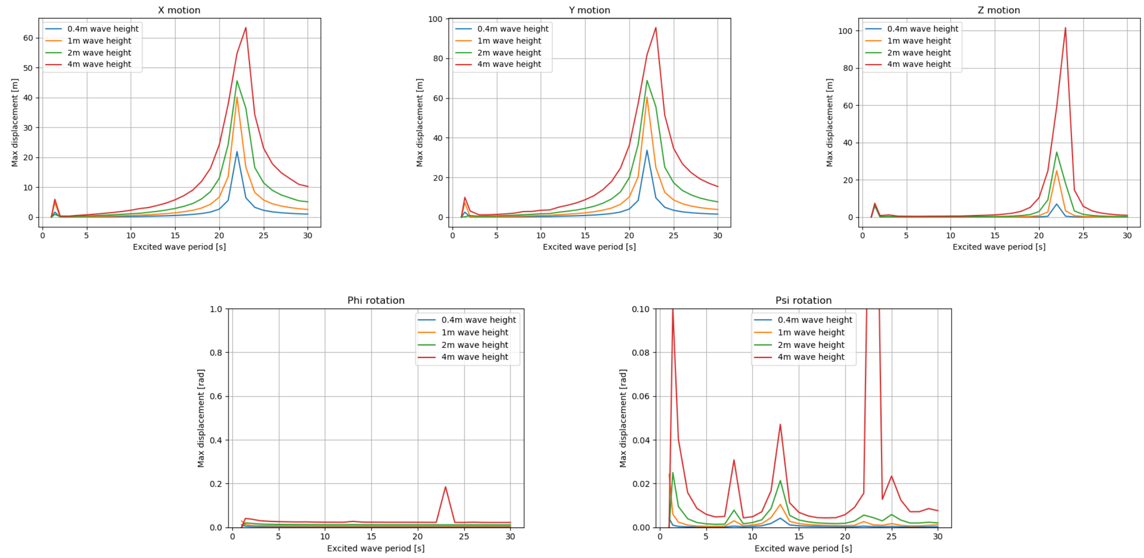


Figure 2.20: Max displacement due to different wave periods

It has to be noted that the peaks are becoming really large and unrealistic because of the resonance and the chaos that occurs in the system. The system is undamped and this results in extreme large motions when resonance occurs. Until a wave height of 3m, it behaves realistically. In the graphs, a peak around the 22s is seen in every motion and rotation. This is the natural period of the X and Y pendulum. These motions are becoming really significant around their natural periods and they affect the other motions as shown in the other plots.

2.4. Discussion

During this chapter, the focus was on the calculation of the modes and natural frequencies of the lifted object, the monopile. With the use of simple examples the difference and effect of linear and nonlinear solutions are explained. The goal of this was to research if the small displacement method is applicable. However, during these examples and the final simplified model, assumptions are made. In this section, it will be discussed if these assumptions are correct and what can be concluded from the results of this chapter.

Step by step a simple model is expanded to the simplified model, with two different solutions per model. With the use of decay tests and fast Fourier transform the natural frequencies have been determined, the two solutions have been compared and the effect of large amplitudes at nonlinear systems has been introduced. With this knowledge, an estimation can be made if the final simplified model can be approached with the small displacement method or not. Comparing the linear and nonlinear solutions to this system of different initial conditions it can be assumed that the small displacement method is applicable since the nonlinear effects start to appear when the initial displacements are significantly larger than the limits during real operations.

With the method mentioned in the previous part, the final simplified model is used to calculate the natural frequencies of the modes. However, the vessel is not included in this chapter and the question is of the calculated natural frequencies are representative. This will be tested in the next chapter by comparing the natural frequencies with and without a vessel.

When a close look is taken to the FFT of the final simplified model (figure 2.19), some small peaks are noticed only in the nonlinear solution. These peaks are not occurring at natural frequencies of the other modes. An explanation could by that these are because of nonlinear higher order harmonics (2018, Suzuki). These higher harmonic effects are also noticed in the example of the spring pendulum in figure 2.14. If peaks are showing up at a frequency that is the natural frequency of another mode, the coupling between the degrees of freedom will be the explanation for that.

It is assumed is that the monopile will be completely rigid. However, this will not be the case in reality. For this study, the choice of assuming it rigid is to keep the calculations simple and focus on the multibody dynamics. Although the monopile is assumed rigid, it is advised to study the structural dynamics of the monopile.

The results of this chapter are: the first impression of the natural frequencies of the lifted object (XXL Monopile) and an explanation about the differences between the linear and non linear solution.

3

Coupled Vessel and Lift Dynamics

In this chapter the following steps are made: First, the final simplified model of the previous chapter is verified with a model made in OrcaFlex. Second, the model is expanded with the vessel. From there, every analysis is done on the model which consists of the lifted object and the vessel. By doing a modal analysis of this complete system, global modes are determined with their corresponding natural frequency. The influence of different setups is investigated and at the end of this chapter, the biggest coupled modes are determined and explained. The purpose of this chapter is to verify the first impression of the natural frequencies of the lifted object (from the previous chapter), identify global modes and natural frequencies of the complete system (vessel included) and determine where coupling occurs and how strong this coupling is. The structure of this chapter is schematized in figure 3.2.

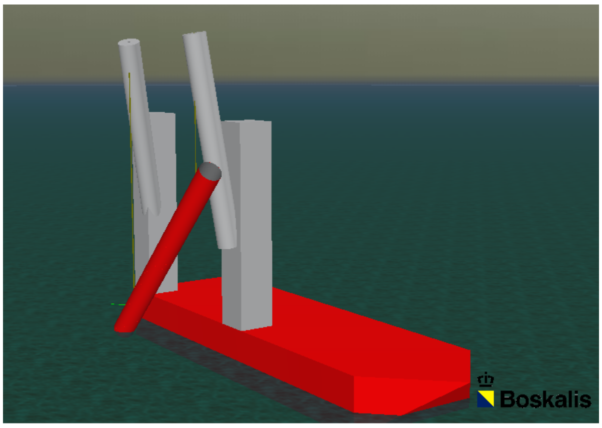


Figure 3.1: OrcaFlex model

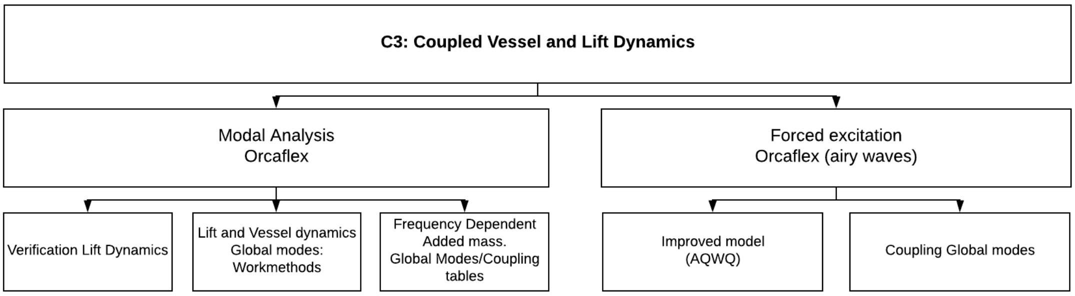


Figure 3.2: Chapter 3 structure

3.1. Lift Dynamics

With Lift Dynamics is meant the dynamics of only the lifted object. In this part, the verification will take place between the final simplified model form the previous chapter and a similar model made in OrcaFlex. The comparisons are made for different tackle lengths.

3.1.1. Verification

It is noticed that no vessel is included in this part of the calculations. The expansion with the vessel will be performed in the next section of this chapter.

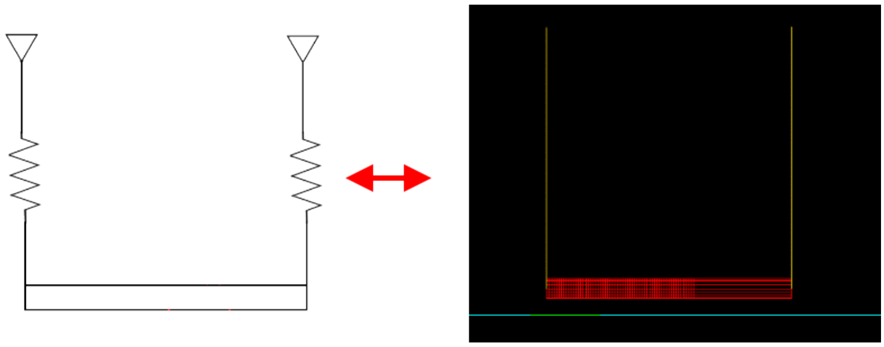


Figure 3.3: Verifying

The different set ups for comparing the models are based on possible installation methods and positions of the monopile. In table 3.1 for example, the natural frequencies at a tackle length of 78 are presented. More comparisons are given in Appendix B.

Tackle length = 78m				
Mode	*Orcaflex [s]*	*DAJU model [rad/s]*	*DAJU [s]*	*Difference[s]*
X pendulum	17.73	0.354	17.75	0.02
Y pendulum	17.73	0.354	17.75	0.02
Pitch	0.649	9.71	0.65	0.00
Yaw	14.32	0.46	13.66	0.66
Heave	0.803	7.84	0.80	0.00

Table 3.1: Natural Frequencies at 78m tackle length

The differences in natural frequencies between the two models are minimum. The biggest difference is at the Yaw mode. This can be explained because of the uncertainty from determining this natural frequency from the FFT. The other frequencies are determined with analytical equations as explained before. However, the 0.66s difference is assumed to be acceptable.

3.2. Vessel and Lift Dynamics

Now the verification is complete, the vessel will be added to the system in the OrcaFlex model. With this model, modal analysis and time domain simulations will be performed. During these analyses, the goal is to determine the natural frequencies of this system and to understand where and how strong coupling occurs.

3.2.1. Modal Analysis

First, a modal analysis is done for one set up. With set up is meant the position of the lifted object (XXL Monopile). This can be positioned in different ways and will affect the natural frequencies. These different positions will be researched in the following section. For this part, the set up is taken where the XXL monopile is hanging with an inclination of 45 degrees and will have a draft of 10m. With draft is meant the submerged length of the monopile. In figure 3.4 this set up is shown. The symbol for inclination in this thesis is alpha α and for the draft it is L.

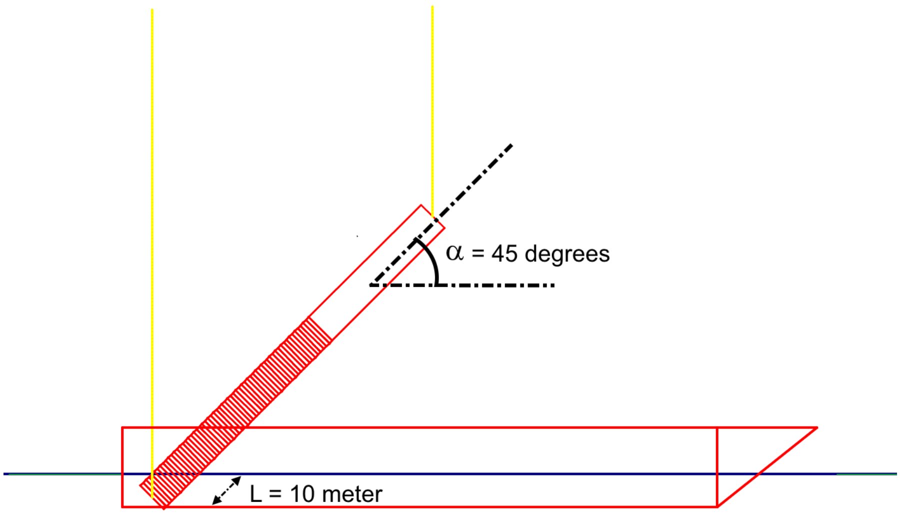


Figure 3.4: Set up model for this section

Global Modes

For the understanding of the following sections, the global modes are determined with their corresponding natural frequencies. With global modes are meant the modes of the vessel and monopile system. Every mode is given a name. The names of these modes are based on the pure modes of the monopile. In figure 3.5 these modes and how they are named are presented. However, the monopile will nut purely move in these modes since the vessel is included in the system, there will be motions of the vessel as well. This is what meant with vessel object coupling. Namely, when a mode consists of monopile and vessel motions this mode is a coupled mode. This coupling will be presented in a table as well.

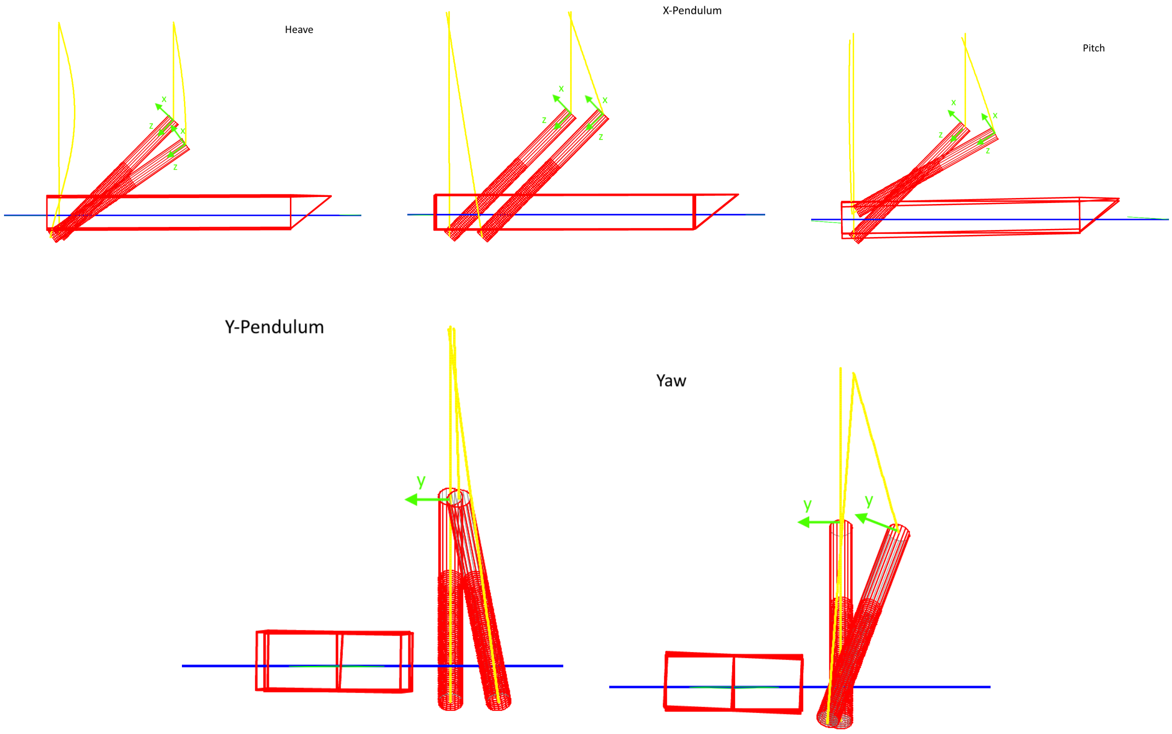


Figure 3.5: Naming modes MP

During the modal analysis, it is noticed that besides the modes which are defined in figure 3.5, two other modes are interesting to look as well. These modes also contain a "Yaw" motion of the monopile, and will be called YawPitch or YawRoll. The Pitch an Roll in these names are referring to the vessel motion in this mode. In figure 3.6 these modes are schematized. These modes are considered interesting because of their natural frequency in the range of the wave excitation.

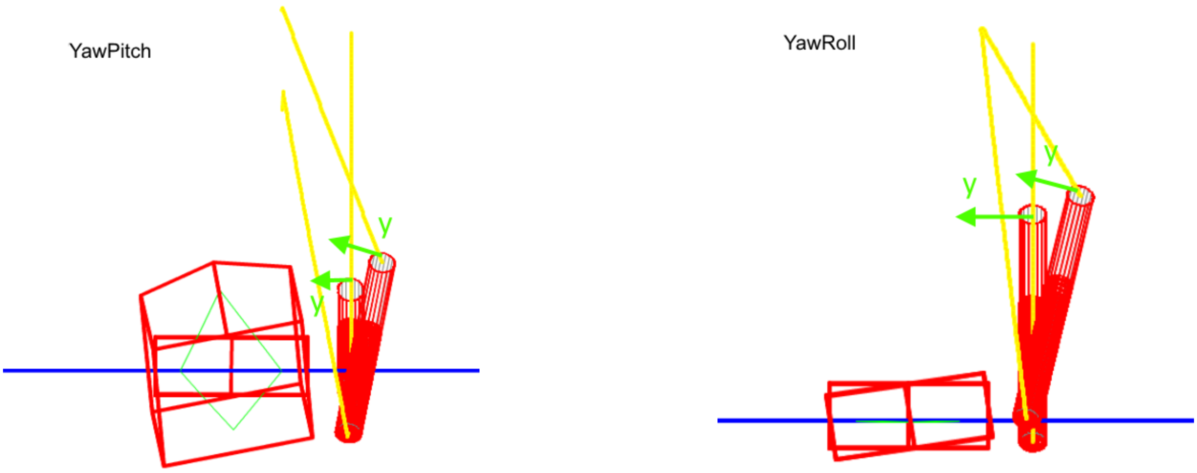


Figure 3.6: The YawPitch and YawRoll modes

The natural frequencies of the modes are shown in table 3.2. Not only the natural periods are presented here, but the modestabel is given as well. In this table the modes are defined by the contribution of the vessel motions an the motions of the monopile. With this table, a first impression can be obtained about the coupling between the monopile and the vessel. The local axis system of the monopile is given in every mode figure with the color green.

	y-Pendulum	**X-Pendulum**	**Yaw**	**YawRoll**	**YawPitch**	**Pitch**	**Heave**
Frequency (rad/s)	**0.18**	**0.29**	**0.43**	**0.62**	**0.67**	**3.09**	**6.61**
Period (s)	**35.30**	**21.50**	**14.55**	**10.06**	**9.44**	**2.03071**	**0.95026**
Monopile; x (m)	-0.01	**-0.72**	0.04	-0.06	**-0.35**	**-0.81**	**0.95**
Monopile; y (m)	**0.21**	-0.07	**0.99**	**0.98**	**0.81**	0.01	0.00
Monopile; z (m)	-0.07	**-0.69**	-0.13	0.18	0.01	**-0.56**	**-0.31**
Monopile; Rx (deg)	**-0.38**	-0.05	**0.63**	**0.55**	**0.44**	0.00	0.00
Monopile; Ry (deg)	0.00	0.00	0.00	-0.03	**0.34**	**0.79**	**-0.54**
Monopile; Rz (deg)	0.00	0.00	0.01	0.01	0.00	0.00	0.00
Vessel; x (m)	0.00	-0.04	0.00	0.01	-0.09	0.00	0.00
Vessel; y (m)	-0.13	0.06	-0.04	0.05	0.06	0.02	0.01
Vessel; z (m)	0.00	0.01	0.00	-0.03	**0.49**	-0.05	-0.01
Vessel; Rx (deg)	-0.05	0.02	**-0.15**	**0.21**	**0.25**	0.09	0.05
Vessel; Ry (deg)	0.00	0.01	0.00	-0.02	**0.33**	-0.02	0.00
Vessel; Rz (deg)	0.05	-0.04	-0.01	0.00	0.00	0.00	0.00

Table 3.2: Modes Table

The modes from table 3.2 are explained and discussed in the following section. The local axis systems per object are from great importance to understand what exactly is expressed with the coupling coefficients.

y-pendulum

The mode named 'Y-pendulum' is the mode where the monopile is mainly swinging in the global YZ-plane. In the mode table 3.2 the significant coupling coefficients are presented boldly. Because of the inclination, a rotation around the local x-axis of the monopile is present. This mode doesn't consist of large vessel motions.

x-pendulum

The 'X-pendulum' named mode is the monopile swinging in the global XZ-plane. Because of the local axis system of the monopile, a z motion is present in this mode. This mode doesn't effect the vessel since the coupling coefficients are very small relative to the coefficients of the monopile.

Yaw

The 'Yaw' motion is called this way since the monopile is mainly yawing at this mode. The 0.99 coupling coefficient in the table at the y motion is there because the local axis system of the monopile is located at an end of the monopile. But the monopile is mostly rotating and not translating in this mode. The first coupling case is present here. The vessel is showing roll motions in this mode and the monopile is yawing.

YawRoll

The name of this mode implies the significant Yaw motion of the monopile and the roll motion of the vessel. In the mode table can be seen that this mode is comparible with the previous described 'Yaw' mode. However, in this mode, the vessel roll is counteracting which results in four seconds shorter natural period. An interesting mode since the natural period lies within the wave excitation region (3s - 12s).

YawPitch

The 'YawPitch' mode is referring to the yaw motion of the monopile and the pitch motion of the vessel. To interpret the coupling coefficients the local axis of the monopile needs to be considered.

Pitch

This mode is driven by the elasticity of the tackles and has a short natural period. Noticing that the coupling coefficients of the vessel are small with respect to the rotation around the local y-axis of the monopile

Heave

The 'Heave' mode is comparable with the 'Pitch' mode in terms of coupling. This mode consists only of motions of the monopile and the vessel does not contribute to this mode.

Influence Inclination and Draft

In the earlier section, something is mentioned about the inclination and the draft of the monopile. The reason for this is because these two parameters are researched in the following part of this chapter. During the installation of the monopile, the monopile needs to be upended. In figure 3.7 the way of upending that is considered is shown. Stage 2 and 3 in figure 3.7 will be analyzed in particular during this thesis.

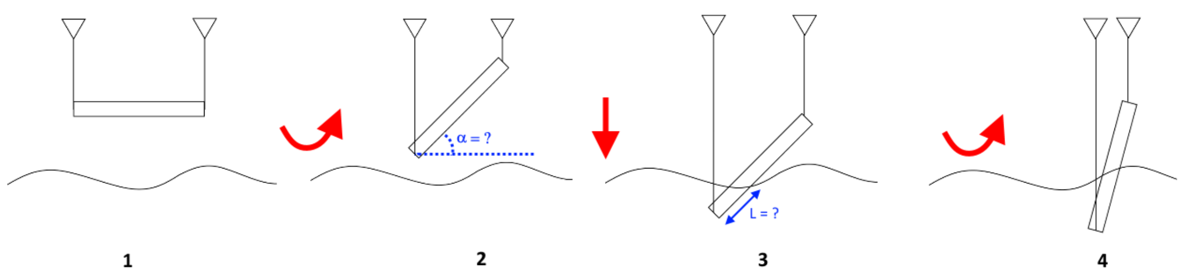


Figure 3.7: Work method

1. Pick up the monopile and slew it long side the vessel.
2. Upend the monopile in air until a certain "inclination angle α".
3. Lower the monopile until a certain "draft L".
4. Continue the upending.

To examine the influence of the upending angle "α or the draft L. The natural frequencies for every mode during these different setups are obtained and plotted. In figure 3.8 the natural periods of the modes are plotted against the inclination α for a drafts: $L = 0m, 10m, 20m$ and $30m$. The red region is the wave excitation region during operation, from 3sec to 12sec. When the natural period of a mode is in this region, resonance may occur what means unwanted large motions of the object or vessel.

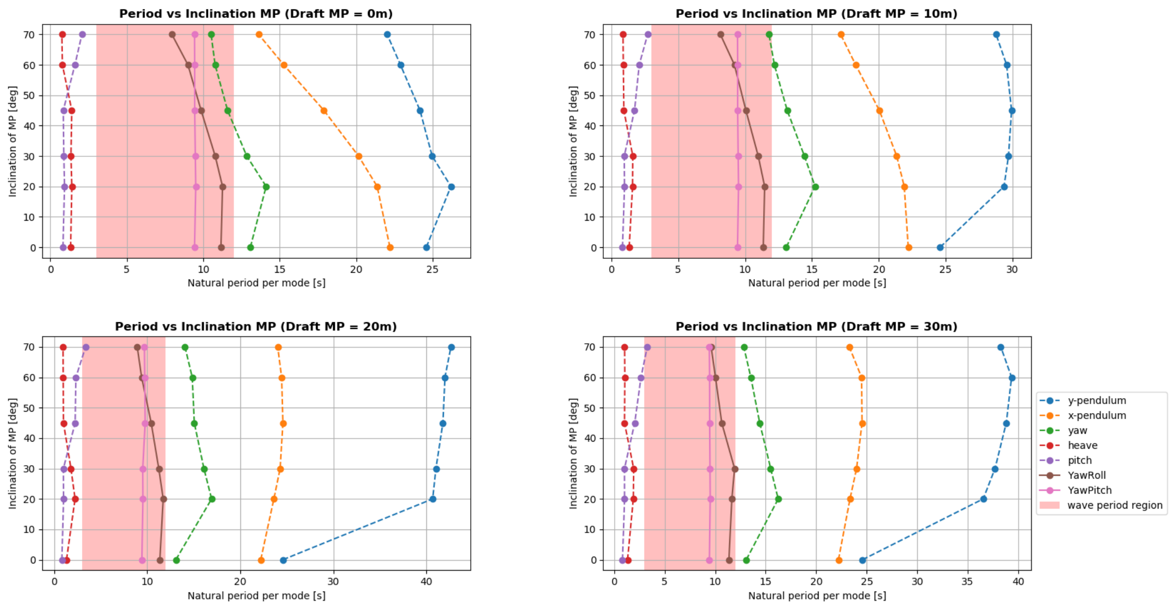


Figure 3.8: Natural period per mode at different inclination angles

Frequency Dependent Added Mass

The BL2 vessel has frequency dependent added mass. However, for the modal analysis, OrcaFlex is not taken the added mass into account at all. To check if this causes a significant difference in the natural frequencies, three modes are analyzed with the different added masses. The three modes that are considered are the modes were the biggest coupling seems to occur. Explanation modes nomenclature: "Mode 7" is the so called "Yaw" motion of the monopile. "Mode 8" is the YawRoll motion and "Mode 9" is the YawPitch motion as earlier mentioned. The natural periods of these modes are plotted for different added mass taken into account. The period for which the added mass is taken into account is the value on the x axis of the graphs in figure 3.9. With taken into account is meant that the added mass is implemented manually to the mass matrix of the vessel (BL2).

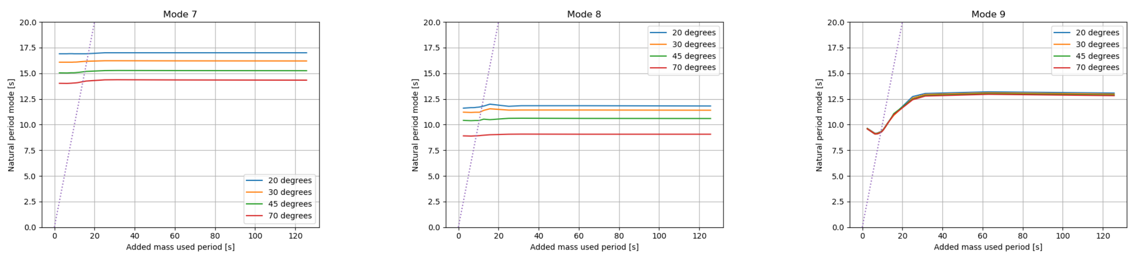
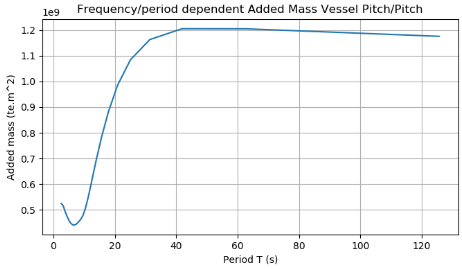


Figure 3.9: Natural period mode vs addedmass period

In mode 7 and mode 8 there is no much different with the change off added mass. This is because the added mass in roll direction is not significant. The influence of the frequency dependent added mass at mode 9 is significant in a particular region. Mode 9 is the YawPitch mode and the added mass in pitch direction is bigger with comparison to the roll part, this is shown in figure 3.10. To find the correct added mass that needs to be included in the mass matrix, a $y = x$ line is added to the plot to find the period.



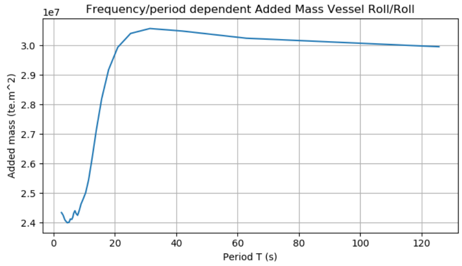


Figure 3.10: Pitch and Roll Frequency Dependent Added Mass: vessel BL2

3.2.2. Forced Excitation: Airy Waves

As a result of the modal analysis, the work methods with $L = 0m$ and $L = 10m$ are examined as not favorable because of natural periods being close to or within the wave period region. The 20m and 30m draft are analyzed further with the use of time domain simulations. For each draft the following inclinations are considered: 20, 30, 45 and 70 degrees. The sea states are given in table 3.3. For the first impression, regular waves are used in order to keep the simulation time short.

Time domain simulations		
Wave Height	1	*[m]*
Wave periods	3 t/m 18	*[s]*
Wave Directions	150,180 and 210	*[degr]*

Table 3.3: Sea States

During the simulations, data like displacements, velocities, accelerations, and forces are obtained. The maximum value of these data is stored. For every sea state and every setup, two different settings are applied. Since the dynamics of the Bokalift 2 vessel can be calculated in two ways, with displacement response amplitude operators (RAO's) or force ROA's. When displacement RAO's are used, the motions of the vessel are only influenced by the amplitude of the incoming waves (2019, Orcina). The free hanging monopile only influences the static equilibrium of the system, not the dynamic behavior of the vessel. With the use of force RAO's of the Bokalift 2, a force calculation per timestep is made which includes the free hanging monopile (static and dynamic). If these two settings are compared, the influence of the monopile on the vessel can be determined.

Improved model

In order to start with the time domain simulations, the model needs to be improved. Orcaflex is calculating the hydrodynamics on structures with use of an extended version of the Morison equation (1950, Morison). However, this is not applicable for the model that is considered. The Morison equation can be used if the structure does not affect the waves, a rule of thumb is (2014, Veritas):

$$\lambda > 5D \tag{3.1}$$

where λ is the wavelength and D is the diameter of the projected cross section. The monopile has a diameter of 10m and To make sure the forces are calculated correctly in OrcaFlex, another method is used. This method makes use of a "fake vessel" (2016, van Steensel). A vessel building block is used to simulate the behavior of the submerged part of the monopile. This vessel block is called the fake vessel. The specifications of this fake vessel are obtained with AQWA, diffraction software, by making use of potential theory (2001, Journee). By doing this, the hydrodynamic forces are calculated more accurate than with use of the Morison equation because diffraction is taken into account (2012, AQWA). For every monopile set up, a fake vessel is created. The drawback of this method is that the water inside of the monopile is not taken correctly into account. The added mass of this water is included in the calculations. However, fluctuations of this mass (Piston mode for example) and the in and outflow are not taken into account. In the next chapter, a solution for taking these phenomena into account is explained. The set up with the fake vessel is shown in figure 3.11.

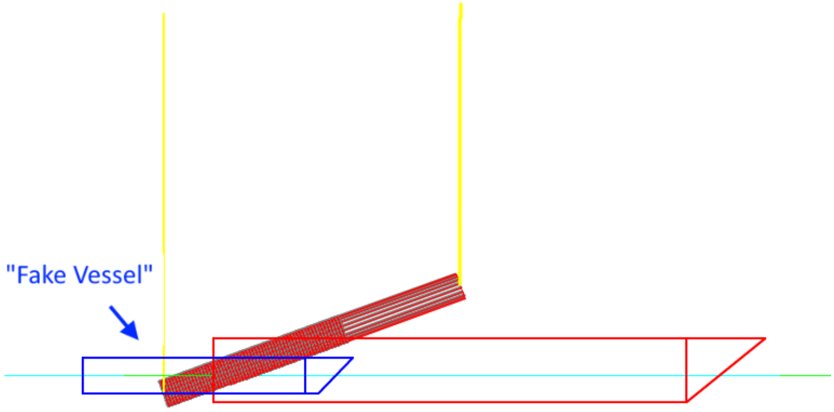


Figure 3.11: Model including "fake vessel"

Determine magnitude of Coupling

The main objective of this thesis is to find out how the system behaves, where is it coupled and where to mitigate if necessary. To identify coupling the solutions with the use of the displacement RAO's and the force RAO's are compared. This because as explained earlier, with the use of the displacement ROA's the vessel is not experiencing any forces of the monopile during the dynamic calculations. For example the maximum pitch motion of the Bokalift 2. For every sea state, this is plotted in figure 3.12. Both solutions are not differing much and this makes sense. Since the BL2 is a long vessel what means a lot of added mass at pitch rotation, which makes the weight of the monopile minimal with compared to these added mass. However, for the roll motion. The influence of the monopile is expected. This because of the moment the monopile creates since its hanging long side and the BL 2 is a monohull vessel. This influence can be seen in the figure 3.13. The peaks in figure 3.13 are around the natural periods of the modes where the monopile is "yawing" as mentioned in previous part. These natural periods are given in table 3.4.

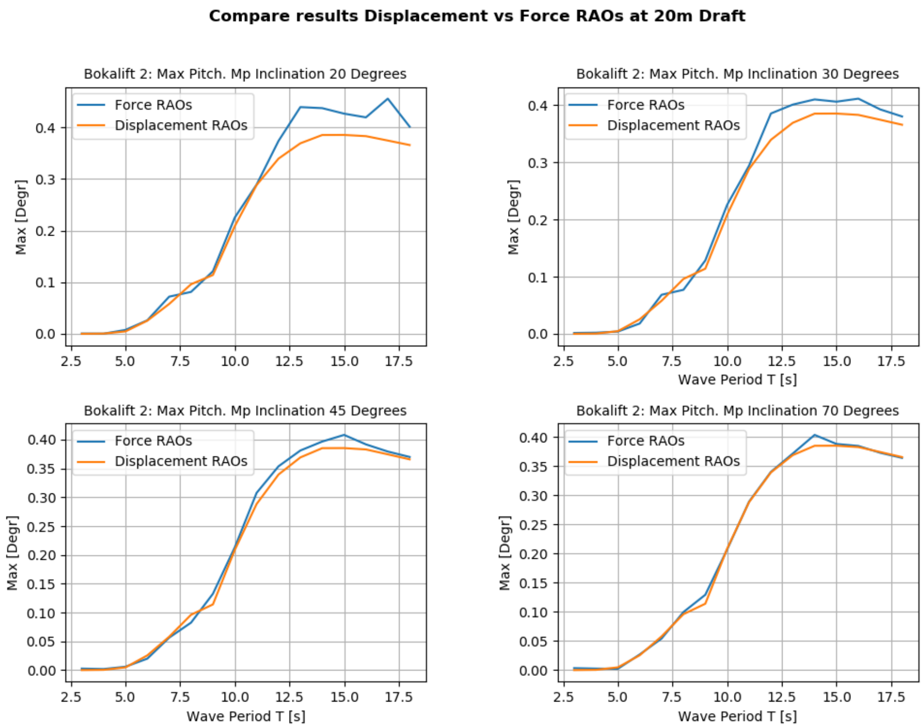


Figure 3.12: Example Coupling 1

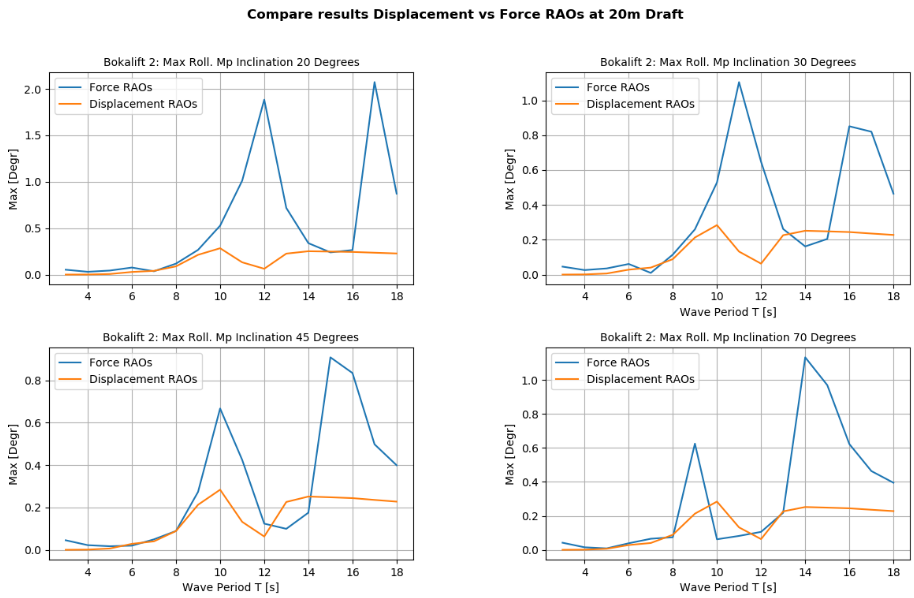


Figure 3.13: Example Coupling 2

With these graphs, the magnitude of the coupled modes is confirmed. Apparently, the yawing of monopile affects the roll motion of the vessel significant. The reason that the yawing of the monopile is affecting the roll motion of the vessel is because of a horizontal force acting on the crane tips. This horizontal force causes an additional overturning moment. To confirm this, the maximum overturning moment is calculated and compared with the Force RAO: Roll Response Amplitude Operator.

Draft: 20m			
	Yaw (s)	**YawRoll (s)**	**Yaw Pitch (s)**
Inclination: 20 (degr)	16.92	11.72	9.54
Inclination: 30 (degr)	16.09	11.24	9.52
Inclination: 45 (degr)	15.06	10.43	9.78
Inclination: 70 (degr)	14.06	8.93	9.7

Table 3.4: Natural Periods per Mode

To confirm this, a calculation of the overturning moment is performed and this is compared with the roll motion RAO of the vessel at the same period. The calculations of the overturning moment are based on the vertical Force (F_{weight}) horizontal Force (F_{yaw}) as shown in figure 3.14. As presented in table 3.5, the overturning moment is approximately ten times larger than the moment causes by the wave excitation. This implies that the moment caused because of the yawing of the monopile is affecting the roll motions of the vessel significant. These values are obtained during the sea state with a wave excitation period around one of the natural periods presented in table 3.4. Has to be noted that with "yaw" is meant the yawing of the monopile.

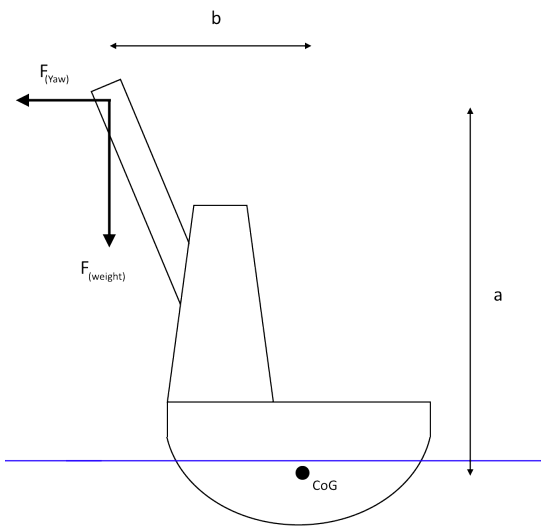


Figure 3.14: Overturning Moment

$$M_{total} = F_{yaw}a + F_{weight}b \tag{3.2}$$

Overturning Moment			Moment applied by the waves		
M_{yaw}	72900	kNm	Roll Moment Waves	9.9E+04	kNm
M_{weight}	1175000	kNm			
M_{Total}	1.25E+06	kNm	Total	9.9E+04	kNm

Table 3.5: Overturning Moment during wave period = 10s

Another way to determine the influence of the monopile to the complete system is to compare the moments of inertia of only the vessel and the monopile. In table 3.6 the both are presented and the part of the monopile is more then 10 percent of the vessel part. This also indicates the effect of the object to the vessel motions.

Moment of Inertia		
Vessel Rx	40250000	$te.m^2$
Mp	5522500	$te.m^2$

Table 3.6: Moments of Inertia

3.3. Discussion

In the discussion of chapter 2 is discussed how representative the calculated natural frequencies are. In this chapter, it is confirmed that a good first impression was obtained about the natural frequencies of the lift modes. However, it was agreed that since the vessel is not included it is not enough to stop the research at this level.

With the use of modal analysis, different installation methods are analyzed with the vessel included. For different drafts and inclination angles the natural frequencies are compared. Although OrcaFlex calculates the natural frequencies with a modal analysis, the added mas is not taken into account. The added mass for the vessel BL2 is frequency dependent and is a significant part of the mass for certain motions. The modal analysis is done again but now with the added mass included for different frequencies. The results show that the frequency dependent added mass does not affect the natural frequencies for the modes which consist of mainly roll motions of the vessel. The modes which consist partly of pitching of the vessel are changing if the different added masses are included. As a solution for this, the added mass in pitch direction at the natural pitch frequency is taken into account.

Since the upending through the splash zone is considered in this chapter, the hydrodynamics on the submerged part of the monopile are from great interest. It is stated that the Morison equation is not suitable for this submerged part and diffraction software is used. However, the behavior of the water inside the monopile is not taken into account. Only the added mass created by this trapped water is taken into account. It is not clear if this assumption will represent a good estimation, especially since an XXL monopile is considered and the water inside the monopile will have a significant volume. To simulate this water in OrcaFlex, next chapter a new phenomenon will be added to the model.

Hydrodynamic behavior like sloshing, slamming and damping doesn't have the priority in this thesis. Because these are nonlinear or expected to be of no significant influence. However, some of these phenomena could be playing a role during the determination of the most favorable work method. Slam loads on the monopile with an inclination of 20 degrees are expected to be much larger than at an inclination of 70 degrees for example.

During the time domain simulations, only airy waves are used. This was to improve the calculation time of all the different setups and sea states. Airy waves are of course not an accurate estimation of the reality and irregular waves need to be used for next time domain simulations to determine the workability. However, the airy waves have given a good impression of the coupling and the magnitude of this coupling.

The complete model is a preliminary model to create an understanding of this system, only one phase is analyzed. During the upending, the monopile will be in many more different positions. These phases need to be investigated too. Also, other loads need to be taken into account like currents and wind.

4

Splash Zone Expansion

The model from the previous chapter is used to identify and determine the coupling of the system between the monopile and the vessel. The model is already improved with the use of the diffraction software. However, in the last year, a new phenomenon seems to occur at large monopiles (read diameter of 10m and larger). This phenomenon is called the piston mode. During this mode, the water inside the monopile, when entering the splash zone, is moving up and down because of the wave excitation (2018, Balkema). This phenomenon is researched earlier and the goal of this chapter is to implement this in the complete model to determine how much the piston mode affects the behavior of the system. It is explained how this is done. As the method in previous chapter, a modal analysis and a time domain simulation is performed. To verify if the piston mode is modeled correctly, complete upended monopile (inclination $\alpha = 90 degrees$) is used for comparison and verification. In figure 4.1 the structure of this chapter is summarized.

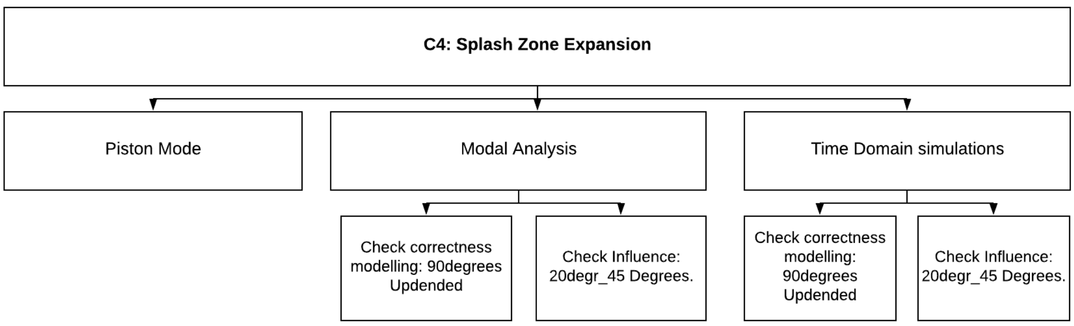


Figure 4.1: Chapter 4 structure

4.1. Piston Mode

In this section the piston mode is explained briefly and why this could be affecting the behavior of the system. As mentioned in the introduction of this chapter, the piston mode is the moving water inside the monopile. This movement is shown in figure 4.2. This movement is caused by the pressure difference in and outside the monopile. When the monopile is under a certain angle, this moving water will result in force and moment with respect to the COG of the monopile. When the frequency of this movement is nearby a natural frequency of another mode, resonance may occur which is not favorable.

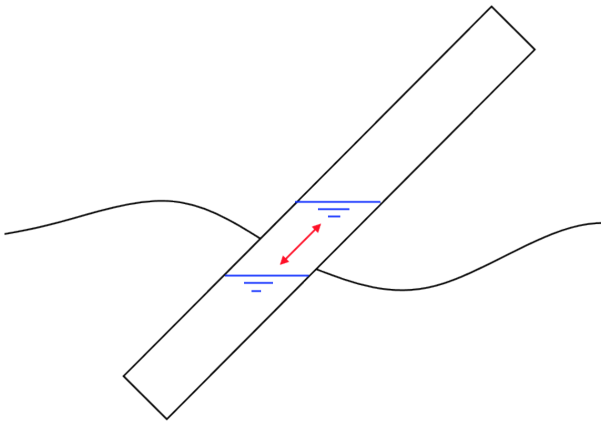


Figure 4.2: Schematized Piston Mode in Monopile

As shown in figure 4.2, the water is moving up and down with a certain frequency, amplitude and phase with respect to the surrounding water. The characteristics of this phenomena are described and tested in different theses (2018, de Vlieger) and (2018, Balkema). With the use of the following equation the Piston Mode periods are calculated and presented in tabel 4.1. The equation is derived in appendix D.

$$\omega_{pistonmode} = \sqrt{\frac{gsin(\alpha)}{L}} \tag{4.1}$$

Piston mode period (s)		
	Draft = 20m	**Draft = 30m**
Inclination: 20 degrees	15.3	18.8
Inclination: 30 degrees	12.6	15.5
Inclination: 45 degrees	10.6	13.1
Inclination: 70 degrees	9.25	11.3

Table 4.1: Piston Mode periods of the different set ups

4.1.1. Piston Mode modelled

The purpose of this chapter, is to implement the piston mode phenomena into the OrcaFlex model. The implementation will be like the simplification shown in figure 4.3. In the following section it is explained how the parameters from figure 4.3 are determined.

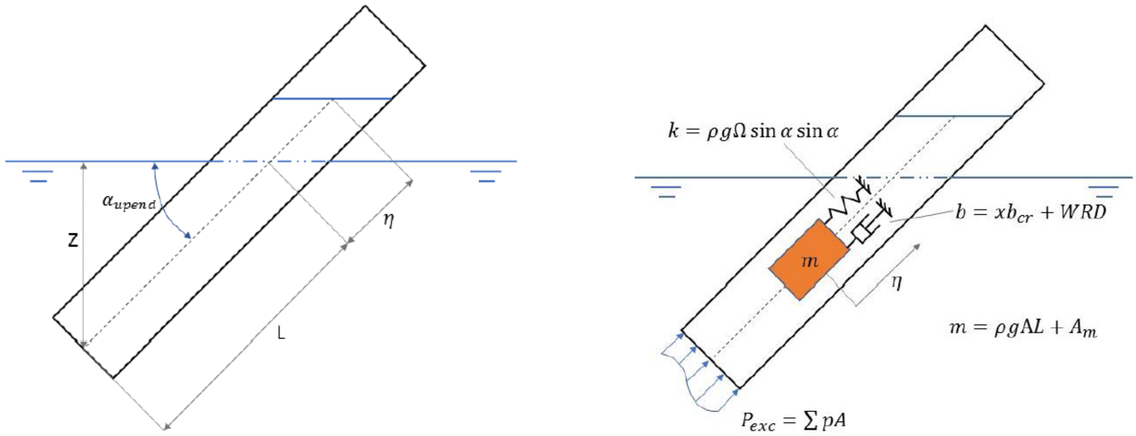


Figure 4.3: Representation of simplified system (2018, de Vlieger)

Stiffness

The stiffness of the spring is the hydrodynamic stiffness of the water column and is determined and explained in appendix D and the piston mode frequency as well.

The Mass and Added Mass

The water inside is the monopile will differ, however for the preliminary calculations of the model the mass of the water column in static equilibrium is taken into account. The added mass in the longitudinal direction (expressed with η in figure 4.3) is determined by taking the added mass in the heave direction of a vertical cylinder with the same diameter as the water column.

Damping

The damping in the system is caused because of different phenomena. These phenomena are in and outflow losses at the bottom of the monopile, viscous wall friction, and wave radiation damping. It is not in the scope of the thesis. However, for the calculations, 10% of the critical damping is considered

Force Excitation at Bottom

The water is excited because of a pressure difference between the in and outside of the cylinder. For the modeling in OrcaFlex, Force RAO's are preferred. These RAO's are created by a pressure integration over the bottom of the monopile at different frequencies. Thes pressures at the different frequencies are obtained with the use of diffraction software AQWA. The picture of the mesh is given in figure 4.4.

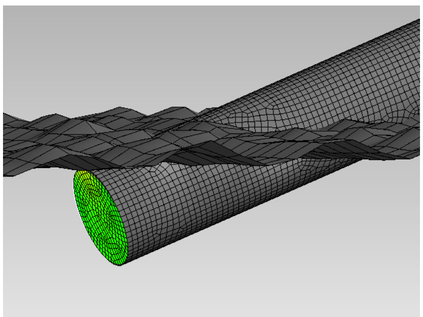


Figure 4.4: Using AQWA to define pressures at the bottom of Monopile

The goal of the modeling of the piston mode is to simulate the moving mass as precisely as possible. However, there are limits in OrcaFlex. The way that the mass will transfer its loads to the monopile is not complete modeled correctly. In ideal conditions, the mass will not transfer any of its loads in the longitudinal z-direction of the monopile. With the confinement of particular elements in OrcaFlex, it is not succeeded to model it such as mentioned. Although the model built is not ideal, it will be used to do comparison studies because the unwanted effect is not significant read Appendix F.

4.2. Modal Analysis

To determine the influence of the piston mode, a comparison is made. This comparison is between a model build like in Chapter 3 (XXL monopile, Vessel BL2, and Fake vessel for correct hydrodynamics submerged part monopile) and a model expanded with the piston mode. With a modal analysis, the natural frequencies per mode will be compared. Nevertheless, first, it will be checked of the piston mode is modeled correctly for the modal analysis. This is checked with a set up where the monopile is upended completely, inclination α = 90 degrees shown in figure 4.5. In this position the piston mode should affect the natural frequencies of the modes minimal. In table 4.2 several natural frequencies of different modes are given and compared. The difference is minimal, this implies that the Piston mode is modeled correctly for the modal analysis.

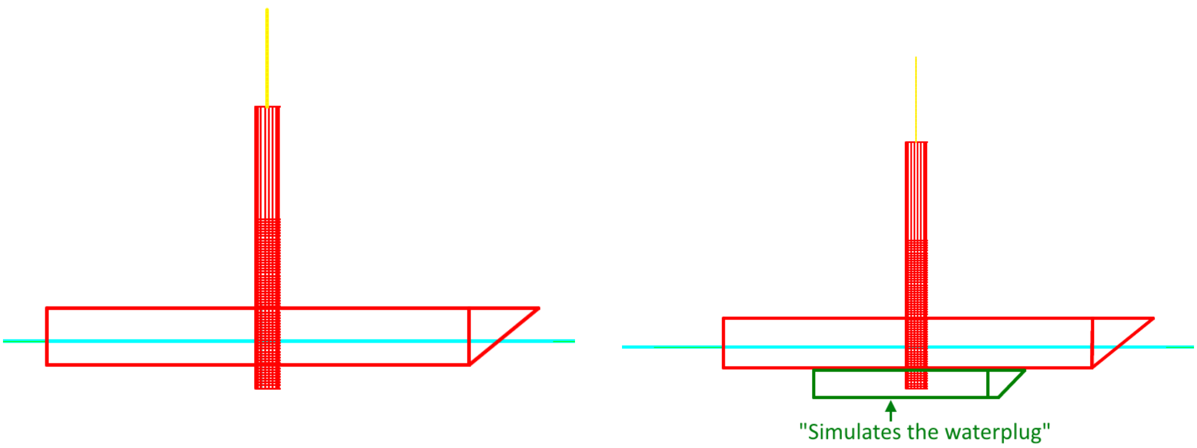


Figure 4.5: The two models compared: Completely Upended

	mode 1	mode 2	mode 3	mode 4	mode 5	mode 6	mode 7
Without Piston Mode (s)	35.60	32.77	6.49	5.95	5.75	5.26	1.08
With Piston Mode (s)	34.70	32.26	6.53	5.92	5.55	5.25	1.07
Difference (s)	0.90	0.51	0.04	0.03	0.20	0.00	0.02

Table 4.2: Naturel Periods comparing models: Completely Upended

Now the statics are correct, it is concluded that the differences between the natural periods can be neglected. The work method with a draft of 20m and inclination of 45 degrees is compared for the two models with a modal analysis as well. The model with piston mode include is called the “Plus model”. In table 4.3 the difference in natural frequencies per mode are given.

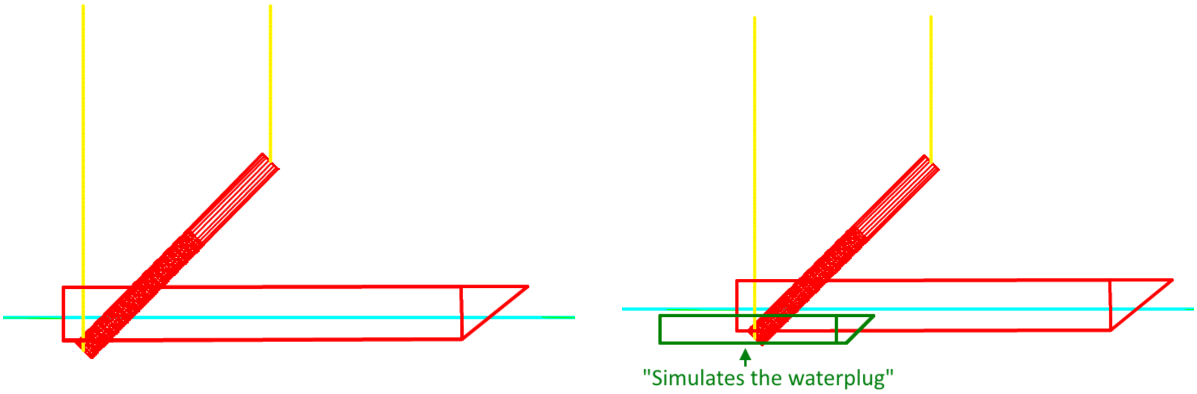


Figure 4.6: The two models compared: 45 degrees Upended

Nomenclature Chapter 3:	X pendulum **mode 1**	Y pendulum **mode 2**	Yaw **mode 3**	yawPitch **mode 4**	YawRoll **mode 5**	Pitch **mode 6**	Heave **mode 7**
Without Piston Mode (s)	24.81	42.62	15.26	12.48	10.62	2.27	1.01
With Piston Mode (s)	25.10	41.27	15.17	12.42	10.75	2.22	1.00
Difference (s)	0.29	1.35	0.09	0.06	0.13	0.05	0.01

Table 4.3: Natural Periods compared: 45 Degrees Upended

4.3. Time Domain Analysis

During the time domain simulations, the same sea states are used as in the time domain simulations in chapter 3 and are given in table4.4. The reason of this time domain simulation is to compare the output with the model in chapter 3, which is not including the piston mode phenomenon. With this comparison, the effect of the piston mode can be determined. However, since the modeling of the piston mode is novel in OrcaFlex, it has to be verified that the way of modeling is correct. In the previous section, verification is done by comparing the modal analysis. To do an extra verification with the time domain simulation, again the complete upended case is simulated where the monopile is at 90 degrees inclination. Here repeatedly, there should not be a significant difference in both solutions since the piston mode would not affect the behavior of the system. If there is a significant difference between the two solutions, it can be stated that the way of modeling is not correct. Although the modal analysis is showing small differences between with and without piston mode modeling, a difference is expected in the time domain simulations because of the incorrect load transfers of the mass explained in appendix F.

Time domain simulations		
Wave Height	1	*[m]*
Wave periods	3 t/m 18	*[s]*
Wave Directions	150,180 and 210	*[degr]*

Table 4.4: Sea States

The output that is plotted in figure 4.7 are the maximum values during a time domain simulation at wave periods from 3s t/m 18s. As shown, it is noticed that there is a difference between the two outputs. This difference occurs at the natural period of the roll and pitch mode of the vessel around 10s and 13s. This implies that the Piston Mode modeling is not correct for this case. However, the waterplug is simulated as a mass connected to the Monopile with stiffness and zero damping. This results in unrealistic large motions when resonance occurs and could be a reason for these differences between the two models.

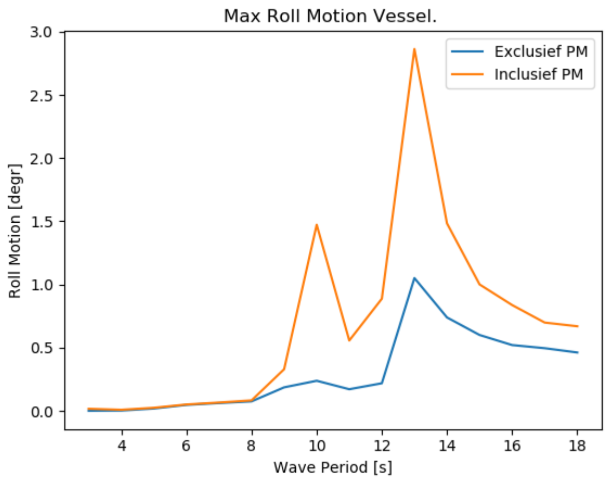


Figure 4.7: Verifying Piston Mode: 90 degrees upended Time Domain Simulation

When a closer look is taken to the output, it can be concluded that the waterplug indeed is moving with unrealistic large amplitudes around the natural periods. This is because the waterplug is only constraint with a stiffness and is not damped. In the introduction of this chapter is mentioned that, if damping is applied, 10% of the critical damping will be a good estimation (2018, (2018). If this 10% is applied, the solution is becoming more equal shown in figure 4.8. The difference between the two models begins at a 10s wave period. For Boskalis this is the maximum wave period for operations.

The difference in the region 10s - 18s could be the reason of the unwanted force transfer in longitudinal Z-direction of the monopile mentioned in Appendix F.

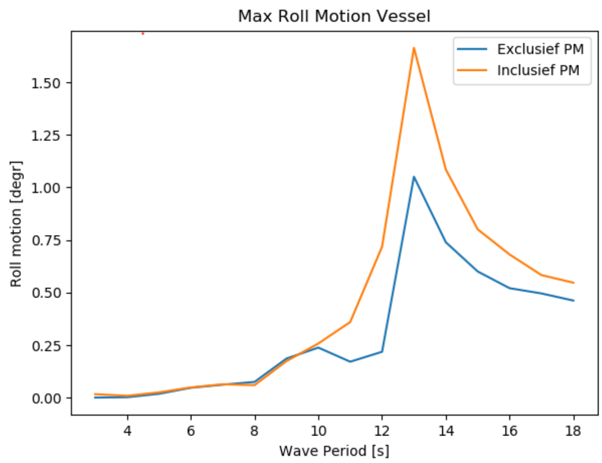


Figure 4.8: Verifying Piston Mode: 90 degrees upended Time Domain Simulation (0.1 * Critical Damping)

As mentioned in the introduction of this chapter, the waterplug is simulated at a way that its weight moves up and down is taken into account. And should not influence the behavior of the system when the monopile is upended at 90 degrees. With the damping added it is still clear that there is a difference between the two models. But the reason of this difference is explained and with this knowledge, a time domain analysis is done for the favorable work method: Draft 20m, inclination 45degrees.

Time Domain Simulation 20m Draft, 45 degrees inclination Comparison

As mentioned in earlier sections, the piston mode modeling is not modeled complete correct. However, for the operational wave periods (0 - 10s) the model examines reliable. For this reason, the most favorable work method is considered and compared for the with and without piston mode modeling.

Before doing the comparison, the method of including the damping of the piston mode is studied. In figure 4.9 the effective tension of the mid crane is plotted with different percentages of the critical damping.

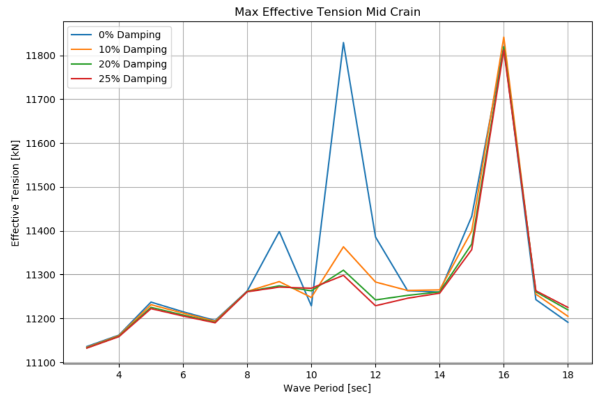


Figure 4.9: Different percentages of the critical damping- Effective Tension

In figure 4.10 and 4.11 the maximum roll and pitch motion of the vessel is presented respectively. Although the model with the piston mode included differs for the roll motion of the vessel, the pitch motion of the vessel is not affected by the piston mode.

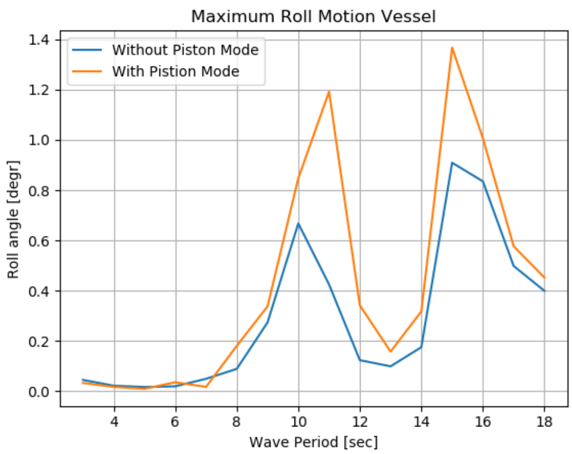


Figure 4.10: Draft: 20m. Inclination: 45degrees. Time Domain Results maximum Roll motion vessel BL2

The first peak in figure 4.10 occurs between the 10s and 11s wave period. The piston mode is 10.6s as shown in table 4.5. The YawRoll mode for this set up has a natural period of 10.4s. These two modes are close and are the reason of the larger peak of the model with Piston Mode included.

Piston mode period (s)		
	Draft = 20m	**Draft = 30m**
Inclination: 20 degrees	15.3	18.8
Inclination: 30 degrees	12.6	15.5
Inclination: 45 degrees	10.6	13.1
Inclination: 70 degrees	9.25	11.3

Table 4.5: Piston Mode periods of the different set ups

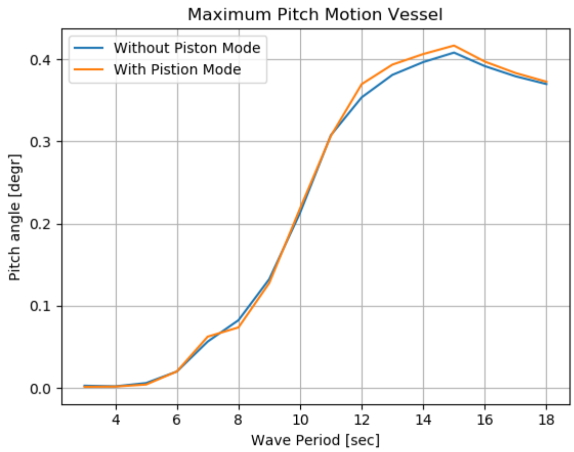


Figure 4.11: Draft: 20m. Inclination: 45degrees. Time Domain Results maximum Pitch motion vessel BL2

4.4. Discussion

This chapter was about the modeling of the piston mode phenomenon. By performing modal analysis and time domain simulations for different setups, the correctness of the modeling is examined. The way of modeling is questioned mainly regarding transferring the loads caused by the piston mode correctly to the monopile. When the correctness of the model is determined, the effect of the piston mode is researched.

Because the model consists of a solid cylinder, the weight of the mass that simulates the waterplug will compensate for the buoyancy. The mass that simulates the waterplug will have no volume and supposes to apply only loads normal to the wall of the monopile. Although the mass is constraint to the monopile with a "free" degree of freedom in the longitudinal Z-direction of the monopile, it still transfers forces in this direction. This because stiffness is added in this direction. This stiffness simulates the hydrostatic stiffness of the waterplug. Because of this, dynamic unwanted forces are transferred during the time domain analysis. After an analyze, appendix F, it is confirmed that this forces are small with respect to the forces occurring in the complete system and therefore will be neglected.

If the unwanted forces indeed can be neglected, the piston mode phenomenon clearly effects the behavior of the complete vessel and monopile system at certain ways. When the sea state has a period of 10s, the piston mode results in larger roll motions of the vessel. However, at this sea state no operations will be executed.

5

Required Crane Capacity and Workability

During the previous chapters a most favorable work method is determined and the piston mode is modeled in OrcaFlex. With these results, the workability of the operation is determined in this chapter. This chapter will consists of three parts. The first part is about the influence of the Piston mode. In previous chapter, this is determined briefly with the use of regular sea states. However, in this chapter irregular sea states are taken into account and will result in a better simulation of the reality. After this part, a study is done to the required crane capacities for the installation and upending in particular. The last part of this chapter is about the workability and mitigation with the use tuggerlines. A tuggerline is a winch that is connected to the object and vessel or crane which applies a certain tension based on the velocity of the object which results in damping.

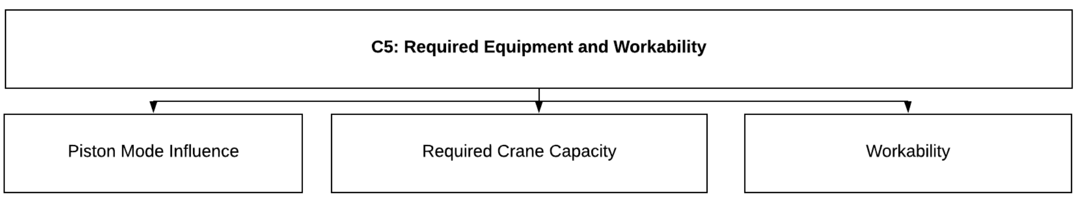


Figure 5.1: Chapter 4 structure

5.1. Piston Mode Influence

The piston mode is the moving waterplug inside the submerged part of the monopile. This will result in a various moment applied to the monopile. The influence of this varying moment is determined in this section. In figure 5.2 this phenomena is presented. The differences because of this piston mode are expected directly in the tackles.

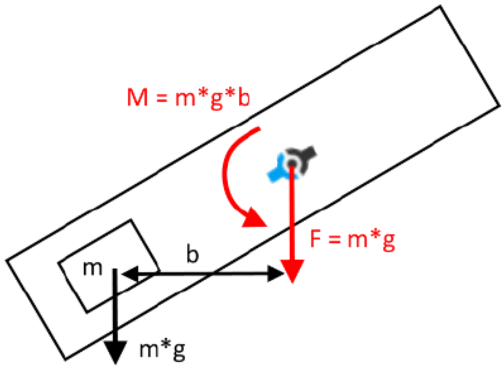


Figure 5.2: Piston Mode

The effective tension is plotted in figure 5.3 for a model with and a model without the piston mode. The sea states which are used for the simulations are presented in table 5.1. The effect of the piston mode could result in a decrease of the workability, especially during seas states with the period close to the piston mode period. The effect of the piston mode on the system is not only determined with the effective stress but the roll of the vessel as well. In figure 5.4 the effect of the piston mode presented on the roll motion of the vessel.

Direction [degr]	150	180	210
Tp [s]		3 - 10s	
Hs [m]	1.5	1.75	2

Table 5.1: Sea States

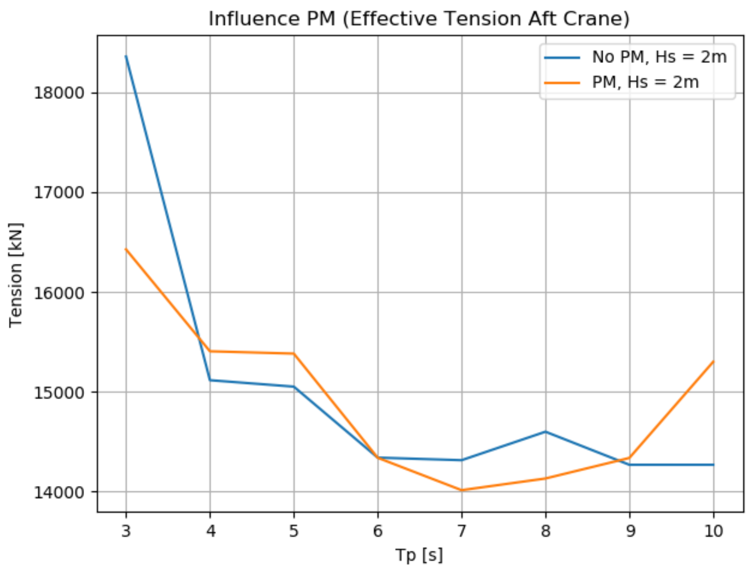


Figure 5.3: Piston Mode Influence - Effected Tension [kN]

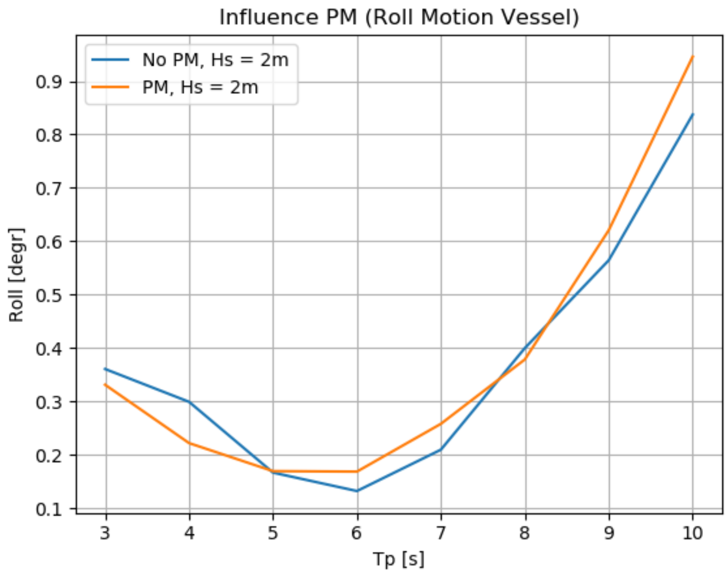


Figure 5.4: Piston Mode Influence - Roll Motion Vessel

It is noticed that the model with the piston mode included results in a slightly different behavior. However, this difference is not significant in the region between a T_p of 3s and 10s. When the waveperiod is becoming close to the piston mode period (10.6s in this case), an increase is noted. At smaller period the model without the piston mode results in larger loads. Which is not expected. An explanation for this is because of the not complete correct way of modeling the piston mode. In previous chapter the "Unwanted" transferred load is mentioned. which results in a different weight distribution. At figure 5.5 the tension at the mid crane tackle is presented.

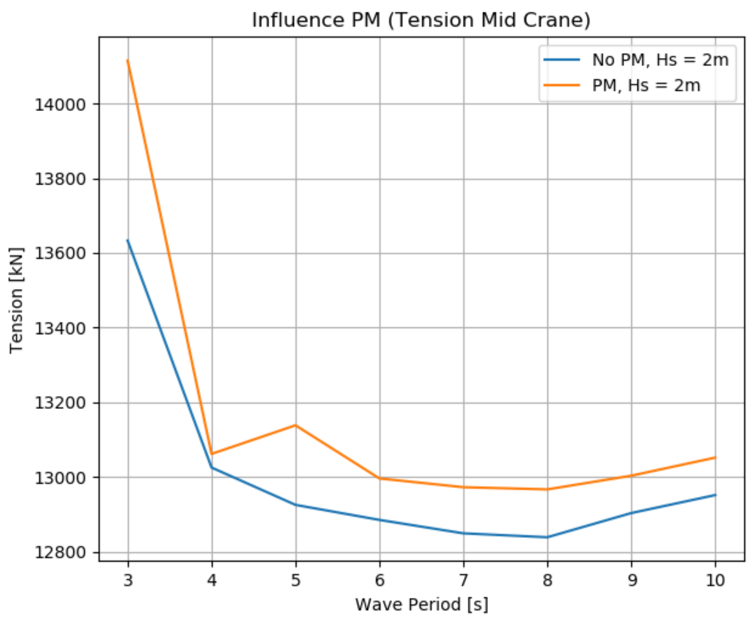


Figure 5.5: Piston Mode Influence - Mid Crane Tension

The large tensions at the short period waves in general are occurring because of the pitch and heave mode of the monopile. These modes have natural periods of 0.99s and 2.11s respectively. However, the significant wave height used in these cases are unreal because of the breaking limit of this short period waves.

5.2. Required Crane Capacity

Before defining the required crane capacity and workability two comments need to be made. First, it has to be remarked that the capacity of the middle crane needs to be determined during the vertical lowering of the monopile. The last phase of the work method, explained earlier in this thesis, consists of lowering the monopile with only the crane in the middle. Since the weight of the monopile is 2500 tons, the crane capacity of the middle crane needs to be at least 2500tons. In this section it is only about the determination of the required crane capacity for the aft crane. Which is used for the upending. Second, the workabilty that is determined will be the workability of the upending through the splash zone. To have a global idea of the complete workability of the installation of an XXL monopile, the complete operation needs to be analyzed. Since the focus of this research was on the upending through the splash zone, only this phase is considered.

The workability depends on different limits. The vessel has its limits in terms of maximum roll motion for example. The object has its limits with respect to the maximum allowable motion (safety) and the equipment comes with a certain limits in terms of forces. The crane has limiting forces at the crane tip. In the first part of this section these forces are determined. After, it is checked if the limits of the motions of the monopile are reached and how to mitigate/reduce these motions.

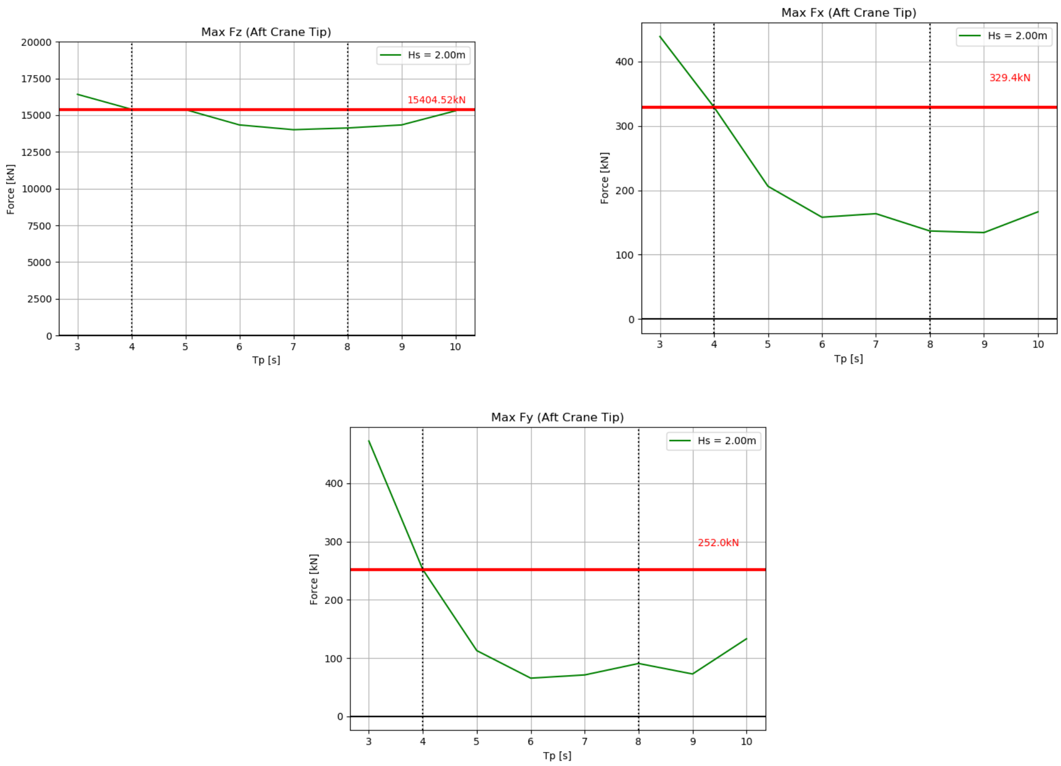


Figure 5.6: Max Forces at Aft Crane Tip

Figure 5.6 implies that a significant waveheight of 2m with T_p between the 4s and the 8s is requisite. Based on this, the crane capacity is determined at 2000tons for the aft crane. With use of this capacity the max oflead and sidelead angles are determined. These values are presented in table 5.2. In the figure is also presented that the max force for $T_p =< 4s$ is increasing. However, a wave with a $T_p =< 4s$ will never reach a significant waveheight of 2m because of breaking (1944,Miche). See table 5.3

	Mid	Aft	
Crane capacity	2500	2000	*[tons]*
Of/Side lead angle	2/2	2/2	*[degr]*

Table 5.2: Required Cranes

Tp [s]	Wave Length [m]	Max Wave Height [m]
1	1.6	0.2
2	6.2	0.9
3	14.1	2.0
4	25.0	3.5

Table 5.3: Max Wave height

5.3. Workability

The workablity in general is the maximum sea state where the operation is not exceeding any limits or criteria. For the upending case the aft crane capacity is determined in the previous section and the criteria and limits are present in table 5.4

Criteria		
	Limiting value	
Roll Motion Vessel	1.5	*[degr]*
Horizontal Motion Object	3	*[m]*
Crane Capacity		
Engineered Curve	2/2	*[-]*
Crane Capacity	2000	*[tons]*
Fz	19620	*[kN]*
Fy (offlead load)	684.73	*[kN]*
Fx (sidelead load)	684.73	*[kN]*

Table 5.4: Criteria

To improve the workability, the monopile can be constraint using a whinch in OrcaFlex. This winch is simulation a tuggerline (TL). These kind of lines are uses to control a lifted object by adding damping to the system. In this case a simple winch configuration is applied and the similar sea states as in previous section are simulated. With a comparison the effect of the mitigation is presented and the increase of the workability is shown.

Besides the maximum allowable offlead angle and forces at the crane tips, the clearance between the object and vessel is also a limitation. In previous chapters the yaw motion of the monopile is found to be of concern. The top or bottom of the monopile is moving from and towards the vessel in this motion. To controll this, tuggerlines can be connected to the top of the monopile and the vessel for example. The effect of the tuggerline is presented in figure 5.7.

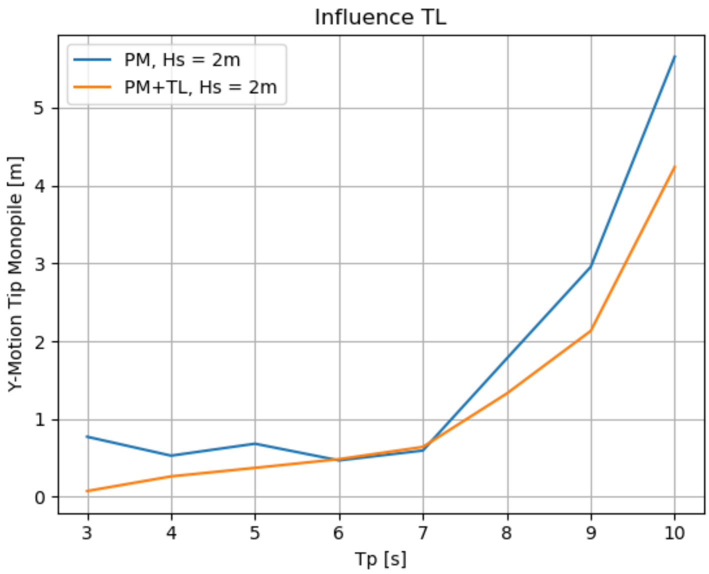


Figure 5.7: Horizontal motion Monopile

PM	**Unity Check: Hs = 2.0m**							
	Tp = 3	4	5	6	7	8	9	10
Roll Motion	0.22	0.15	0.11	0.11	0.17	0.25	0.41	0.63
Fz	0.84	0.79	0.78	0.73	0.71	0.72	0.73	0.78
Fy	0.69	0.37	0.17	0.09	0.10	0.13	0.11	0.19
Fx	0.64	0.48	0.30	0.23	0.24	0.20	0.20	0.24
Horizontal motion Object	0.26	0.18	0.23	0.15	0.20	0.59	0.98	1.88

Table 5.5: Unity Check

PM + TL	**Unity Check: Hs = 2.0m**							
	Tp = 3	4	5	6	7	8	9	10
Roll Motion	0.19	0.15	0.10	0.10	0.15	0.22	0.35	0.47
Fz	0.81	0.80	0.77	0.72	0.70	0.71	0.73	0.76
Fy	0.67	0.31	0.15	0.10	0.10	0.12	0.11	0.16
Fx	0.66	0.46	0.26	0.21	0.20	0.15	0.14	0.19
Horizontal motion Object	0.02	0.09	0.12	0.16	0.21	0.44	0.71	1.41

Table 5.6: Unity check - Tuggerline included

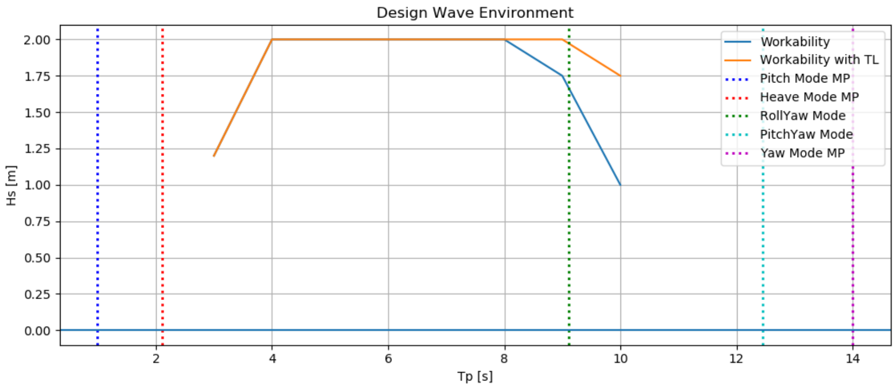


Figure 5.8: Design Wave Environment

5.4. Discussion

The purpose of this chapter was to determine the required capacity for the aft crane during the upending operation, and to define the workability. It has to be mentioned that during all the calculation no factors are used. The results are an estimation to give a first impression about the upending operation. The start of this chapter was about the influence of the piston mode. With the use of irregular sea states, the time domain simulations indicate that the influence of the piston mode is minimal in the operation window (T_p 3s - 10s). The influence of the piston mode becomes significant when the Tp is becoming larger, the influence of the piston mode becomes more significant. However, during these sea states is it unlikely that operations will be executed. Summarized, the piston mode is not of concern in the operation window. The upending operation with a dual lift is feasible with regarding to the unity checks.

6 Conclusions and Recommendations

The goal of this thesis is to get a better understanding of a dual lift, with the focus on the upending of an XXL monopile. With the use of different models and software this better understanding has been obtained. Based on the results and gained knowledge the conclusion of this thesis is formed. In the following section these conclusions are described.

6.1. Conclusions

This study considered three main steps, the Simplified Model Lift Dynamics, the Lift and Vessel Dynamics in Orcaflex and the Splash Zone Expansion. As result of these three steps a better understanding has been obtained and an improved model of an XXL Monopile installation has been developed. With the use of these better understanding a required crane capacity is determined and a workability for the operation is estimated. These findings led to the following conclusions:

Simplified Lift Dynamics

- The system with its parameters can be linearized since the nonlinear and linear solution are very close to each other for displacements within the operation limits, when the crane limits are used as initial displacements the difference between the solutions is neglectable. The assumptions of small displacements is, therefore, applicable.
- From the simplified model it can be concluded that the x and y pendulum modes are around 22s, which is not in the range of the wave expected wave periods during operations (3s-12s). The pitch and heave modes are also not in this range. The yaw mode of the monopile, however, should be of greater concern. This period is around the 8s and this is within the wave excitation region (3s – 12s). Attention is required for this mode.

Lift and Vessel Dynamics (OrcaFlex)

- From the lift and vessel dynamics in OrcaFlex it can be concluded that the simplified lift model results in a good first impression of the modes and frequencies. It is also confirmed that indeed the “Yaw” motion of the monopile plays a significant role in the coupled dynamics.
- A modal analysis revealed that the installation methods with a draft of 0 or 10m are not desired because the natural periods of these set ups are too close to or even within the wave excitation region. Based on the modal-analysis.
- Time domain simulations of four work methods were analyzed. Strong coupling was noticed between the monopile and vessel in the YawRoll mode. Because of the yawing of the monopile, horizontal forces at the crane tips result in an additional overturning moment which affects the roll motion of the vessel significant. As the weight of the monopile is small with regarding to the vessel, it can be concluded that it affects the behavior of the vessel much, especially during the so- called YawRoll mode.
- It is conlcuded that the larger the inclination, the shorter the natural period T_n of the modes. The larger the draft the longer the natural periods of the modes.

Splash Zone Extension

- It is possible to model Piston Mode in OrcaFlex. With the use of preliminary studies, the piston mode is modeled as a moving mass excited by the waves inside the monopile. From comparison studies it is

concluded that the Piston Mode is not modeled correctly. However, first impressions of the influence of the Piston Mode have been obtained. These impressions are that the effect of the piston mode does not affect the natural frequencies of the system and that the influence of the Piston mode is differing for different wave periods.
- It can be conclude that more research and simulations are needed to confirm if the piston mode is affecting the complete system and if the modeling is a good enough approximation of the reality.

Required Crane Capacity and Workability
- For the particular upending method for an XXL Monopile, the crane at the mid position should have at least a capacity of 2500tons and the crane at the aft position a capacity of 2000tons.
- The Piston mode phenomenon is not affecting the upending method significant at the operation window of 3s -10s wave period.
- The workability of the upending method is improved with the use of tugger lines. Especially in the longer wave period region. This because the yawing of the monopile is controlled with the winch.

First, with all these specific conclusions per phase, it can be stated that the objectives of this thesis have been achieved. There is a better understanding of the dynamics of the system. The lifted object and vessel: modes of interest have been determined. Coupling is determined and it is proven that even if the weight of the lifted object is small relative to the vessel, the lifted object can affect the behavior of the mono hull vessel.

Second, Upending through the splash zone is definitely possible. The piston mode phenomenon that occurs can be modeled in OrcaFlex and this model method can be used in other situations as well. There is still space for improvement since this is the first step of modelling the piston mode in OrcaFlex. The improvements will be in terms of damping and force transfer.

Whit all these knowledge the upending with a dual lift mono hull vessel is a step closer to reality. With applying the necessary mitigation of the modes of interest upending with a draft of 20m and inclination of 30/45 degrees is preferred.

6.2. Recommendations

During this thesis, some assumptions were made, to improve the accuracy and certainty of the model, the following recommendations are made:

- **Run irregular sea state in time domain simulations**. In this thesis only regular sea states have been considered. With simulations of irregular sea states a workability of the operation can be determined.

- **Run the model with mitigation methods**. Use tuggers or whinces to mitigiate the modes of interest and define the workability when these mitigations are applied.

- **Calculate more variations of the piston mode modelling method**. To verify this modeling and compare with a model without the Pison Mode included.

- **Do research to other installation methods**. Compare different installation methods. Float over monopils for example. Instead of lifting and upending partly in air, the monopile can be transported floating to the lift vessel and upendend completely in the water.

- **Simulate the complete installation**. In this research only the upending is considerd. However, the installation consists of more phases and it will be needed to analyze this complete work method.

- **Create a PM safety factor** For the time being, a piston mode safety factor could be applied for motion analyzes. This factor can be determined out of the comparison between the with and withoud models.

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A

Appendix A - Intermediate Spring, Pendulum and Rotational Spring Model

A.1. Spring, Pendulum and Rotational spring

A.1.1. Partly - Decoupled

The intermediate step to go from the spring pendulum configuration to the full simplified model will be explained in this part of the section. The intermediate step is a model that consists of a spring pendulum with an rotational spring on the point mass as shown in figure A.1. This spring will be simulating the rotational stiffness on the inertia of the mass J. The mass moment of inertia J is defined as the mass moment of inertia of a long cylinder: $J = \frac{1}{12}mL^2$. Like in the previous sections, the Lagrangian (A.1) and the equations of motions are presented. The idea of this intermediate step is to create a linear version of the final model.

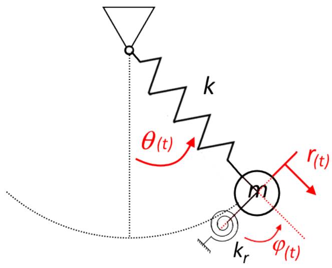

Figure A.1: Spring, Pendulum and Rotational spring

$$L = \frac{1}{2}m(\dot{r} + (l+r)\dot{\theta})^2 + \frac{1}{2}J\dot{\varphi}^2 - \frac{1}{2}kr^2 - \frac{1}{2}k_r^2\varphi + mg[(l+r)cos(\theta) - l] \tag{A.1}$$

Where

J = moment of inertia [kg m^2]

k_r = rotational stiffnes [Nm / rad]

To define this system as the linear version of the final model, the equations of motions are linearized and these are given below:

$$\ddot{r} + \frac{k}{m}r - g = 0 \tag{A.2}$$

$$\ddot{\theta} + \frac{g}{(l+r_s)}\theta = 0 \tag{A.3}$$

$$J\ddot{\varphi} = -k_r\varphi \tag{A.4}$$

B

Appendix B - Verification results simplified model

Tackle length = 78m				
Mode	*Orcaflex [s]*	*DAJU model [rad/s]*	*DAJU [s]*	*Difference[s]*
X pendulum	17.73	0.354	17.75	0.02
Y pendulum	17.73	0.354	17.75	0.02
Pitch	0.649	9.71	0.65	0.00
Yaw	14.32	0.46	13.66	0.66
Heave	0.803	7.84	0.80	0.00

Table B.1: Verification Tackle Length = 78m

Tackle length = 98m				
Mode	*Orcaflex [s]*	*DAJU model [rad/s]*	*DAJU [s]*	*Difference[s]*
X pendulum	19.80	0.32	19.95	0.15
Y pendulum	19.80	0.32	19.95	0.15
Pitch	0.73	8.67	0.72	0.00
Yaw	16.06	0.38	16.62	0.56
Heave	0.90	6.99	0.90	0.00

Table B.2: Verification Tackle Length = 98m

Tackle length = 122m				
Mode	*Orcaflex [s]*	*DAJU model [rad/s]*	*DAJU [s]*	*Difference[s]*
X pendulum	22.18	0.28	22.28	0.10
Y pendulum	22.18	0.28	22.28	0.10
Pitch	0.81	7.79	0.81	0.01
Yaw	17.90	0.34	18.76	0.86
Heave	1.01	6.27	1.00	0.00

Table B.3: Verification Tackle Length = 122m

C

Appendix C - Natural Periods per Mode

AH: Outreach 25M. Upending					
Draft = 0 m	Periods T [s]				
Inclination [deg]	**Y-pendulum**	**X-pendulum**	**Yaw**	**heave**	**pitch**
0	24.6	22.2	13.1	1.3	0.8
20	26.2	21.4	14.1	1.4	0.9
30	25.0	20.1	12.8	1.4	0.9
45	24.2	17.9	11.6	1.4	0.9
60	22.9	15.3	10.8	0.8	1.6
70	22.0	13.6	10.5	0.8	2.1

Table C.1: 0m Draft (Horizontal)

AH: Outreach 25M. Upending					
Draft = 10m	Periods T [s]				
Inclination [deg]	**Y-pendulum**	**X-pendulum**	**Yaw**	**heave**	**Pitch**
0	24.6	22.2	13.1	1.3	0.8
20	29.4	21.9	15.2	1.6	1.0
30	29.7	21.3	14.4	1.6	1.0
45	29.9	20.0	13.1	0.9	1.7
60	29.6	18.3	12.2	0.9	2.1
70	28.8	17.1	11.8	0.9	2.7

Table C.2: 10m Draft

AH: Outreach 25M. Upending					
Draft= 20m	Periods T [s]				
Inclination [deg]	**Y-pendulum**	**X-pendulum**	**Yaw**	**heave**	**Pitch**
0	24.6	22.2	13.1	1.3	0.8
20	40.7	23.6	16.9	2.3	1.0
30	41.0	24.3	1.0	1.8	1.0
45	41.8	24.6	15.1	1.0	2.3
60	42.0	24.4	14.8	1.0	2.3
70	42.6	24.0	14.1	1.0	3.4

Table C.3: 20m Draft

AH: Outreach 25M. Upending					
Draft = 30m	Periods T [s]				
Inclination [deg]	**Y-pendulum**	**X-pendulum**	**Yaw**	**heave**	**Pitch**
0	24.6	22.2	13.1	1.3	0.8
20	36.5	23.3	16.2	1.9	1.0
30	37.7	24.0	15.5	1.9	1.0
45	38.8	24.5	14.4	1.0	2.0
60	39.3	24.5	13.5	1.1	2.7
70	38.3	23.3	12.8	1.0	3.3

Table C.4: 30m Draft

D

Appendix D - Piston Mode Period

In this Appendix the analytical derivation of the piston mode is given. This is done by approach the problem as a simplified linear system. It can be compared with a mass spring system (1993, Faltinsen). Te assumptions for the analytical derivation are as follows:

- The water depth is infinite
- The monopile has a uniform diameter of 10m

Analystical derivation piston mode period inclined monopile

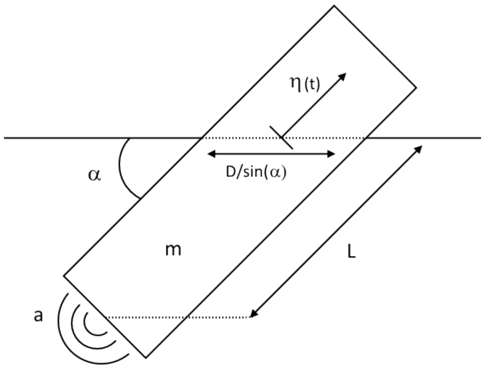

Figure D.1: Monopile and parameters

$$(m + a)\ddot{\eta} + B\dot{\eta} + C\eta = f_a sin(\omega t) \tag{D.1}$$

where:

m = Mass of water inside monopile		[kg]
a = Added mass		[kg]
η = Motion of the water in longitudinal direction of Monopile		[m]
B = Damping		[N s/m]
C = Stiffness		[N/m]
f_s = Excitation force amplitude		[m]

The stiffness can be written as:

$$C = \rho g A_s \tag{D.2}$$

where A_s is the pierced surface area of the cylinder, when the cylinder is under an angle, this area is not a circle but an ellipse and can be described as:

$$A_s = \frac{\pi}{4}\frac{D^2}{sin(\alpha)} \tag{D.3}$$

Now the pierced surface area is known, the natural frequency can be calculated. In general the natural frequency is calculated as follows:

$$\omega_n = \sqrt{\frac{C}{(m + a(\omega)}} \tag{D.4}$$

In general the assumption that $a(omega)$«m is made, or iteration is used to know which added mass is needed. With implementing the derivations for A_s, m and C, the natural frequency of the piston mode is given as follows:

$$\omega_n = \sqrt{\frac{\rho g \frac{\pi}{4} D^2}{(\frac{\pi}{4}\rho D^2 L + a(\omega))}} \tag{D.5}$$

Where $L = S/sin(\alpha)$ and S is the submergence. This part shows that the inclinination α has influence on the natural frequency (2009, Kristiansen).

E

Appendix E - Specifications and Conventions

In this appendix the specifications of the vessel, monopile and all the other things used in the model are given. The axis systems and nomenclature are explained as well.

Bokalift 2 Specifications		
Length	210	*[m]*
Beam	60	*[m]*
Draft	10	*[m]*
Displaced Volume	9.69E+04	*[m^3]*

Table E.1: Bokalift 2 Specifications

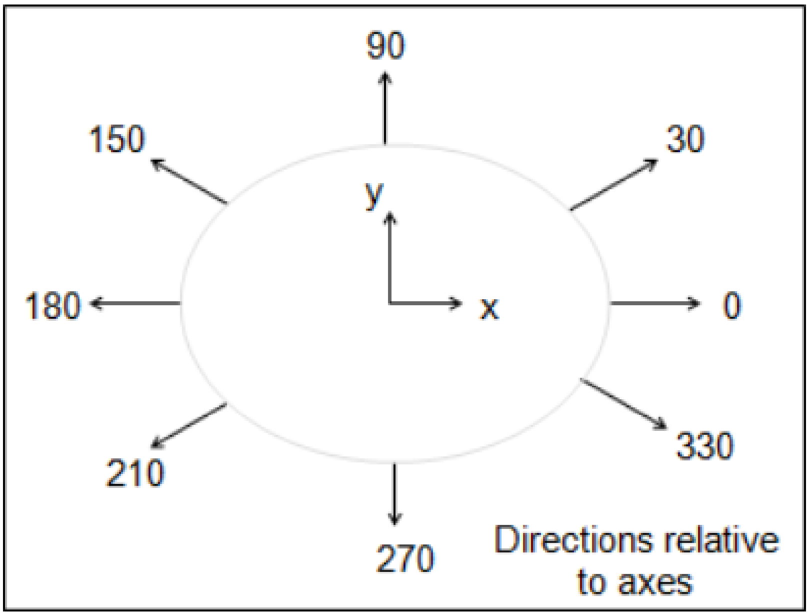

Figure E.1: Convention Wave Direction (2019, Orcina)

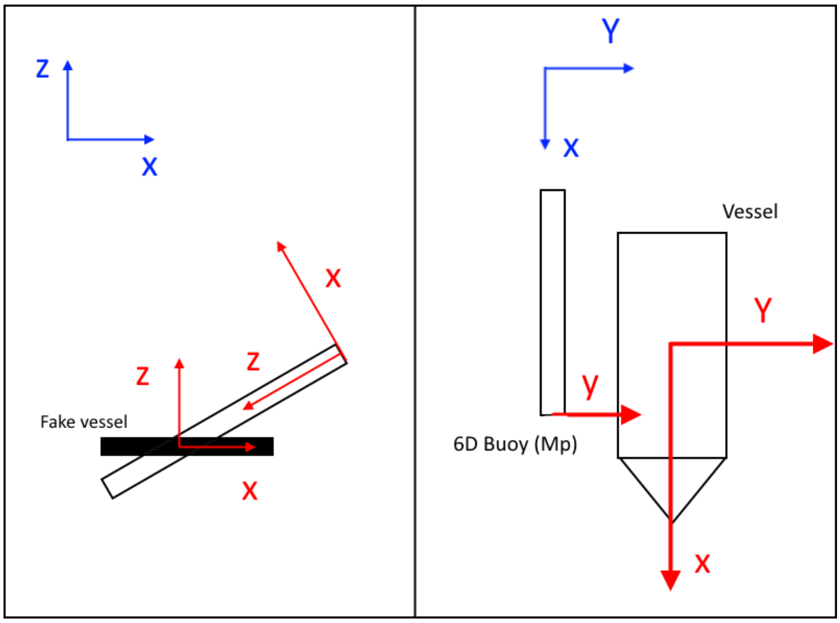

Figure E.2: Local Axis systems

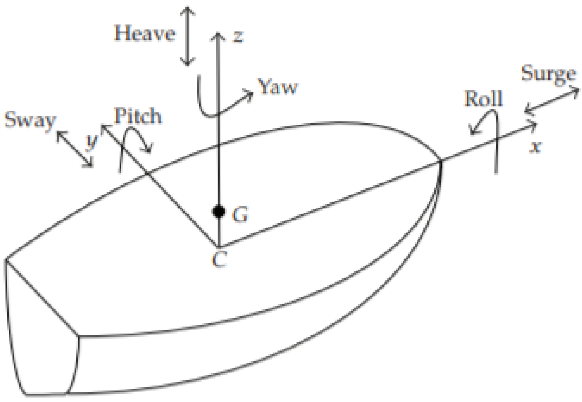

Figure E.3: Diagram of vessel DOF's

F

Appendix F - Piston Mode Modeling

In this Appendix the OrcaFlex modeling of the piston mode is explained detailed. Which kind of objects are used. What kind of values are given to the objects and what are the confinements and limitations.

The piston mode is a moving waterplug in the monopile. This waterplug is simulated as a mass. This mass has the weight of the water that is in the static equilibrium of the monopile. In reality, this mass wil differ over time. However, the amplitude of the excitation is assumed small relative to the water plug and the mass is considered constant. The waterplus is modeled with a "Vessel" component in OrcaFlex and is sketched red in figure F.1. This component is given the mass as mentioned earlier and zero volume so there is no buoyancy. The buoyancy of this waterplug is simulated as a stiffness. (WZAWZDKB)

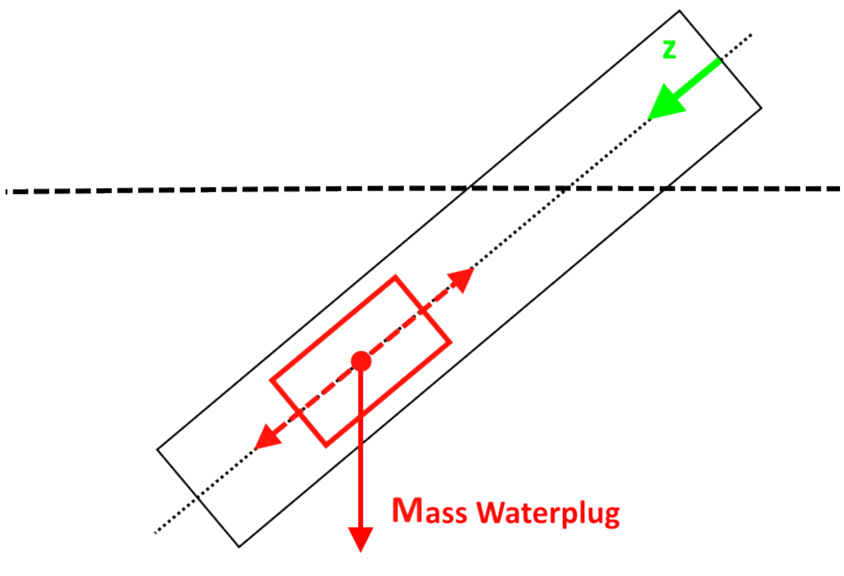

Figure F.1: Mass that simulates the weight of the waterplug

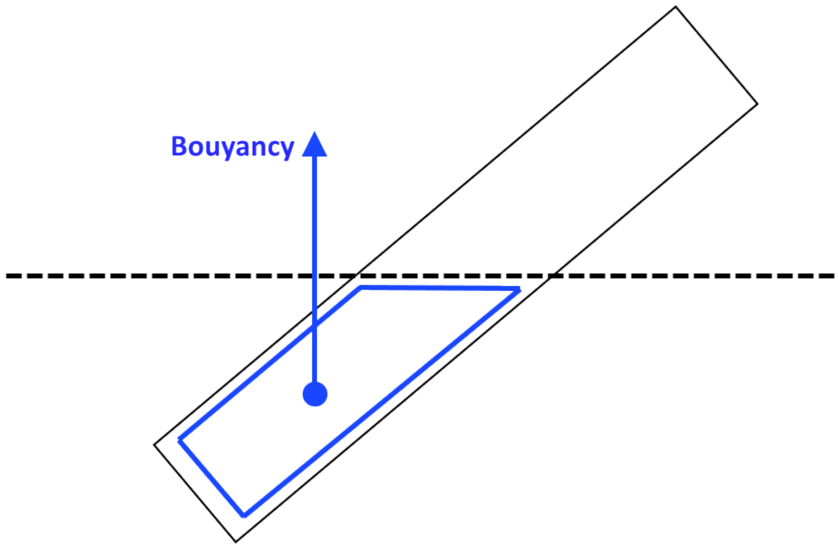

Figure F.2: Buoyancy of the submerged solid cylinder (monopile)

In figure F.3 the constraint is sketched. This constraint will arrange that the load of the moving waterplug is transferred to the monopile because the constraint is constraining the motions in x and y direction. The Z-direction (longitudinal of monopile) is free, a stiffness is applied to simulate the buoyancy of the waterplug. However, this results in a force in the Z-direction which will not be there in reality. The static force in this direction is cancelled out with the buoyancy of the submerged part of the monopile, but the dynamic force because of the motion of the waterplug will not be acting on the monopile in reality.

This can be questioned and the modeling is not completely correct. Since the dynamic extension of the spring is in the magnitude of 0 - 1m, and the stiffness is 555 kN/m, the additional unwanted occurring force is not significant with respect to the forces in the complete system.

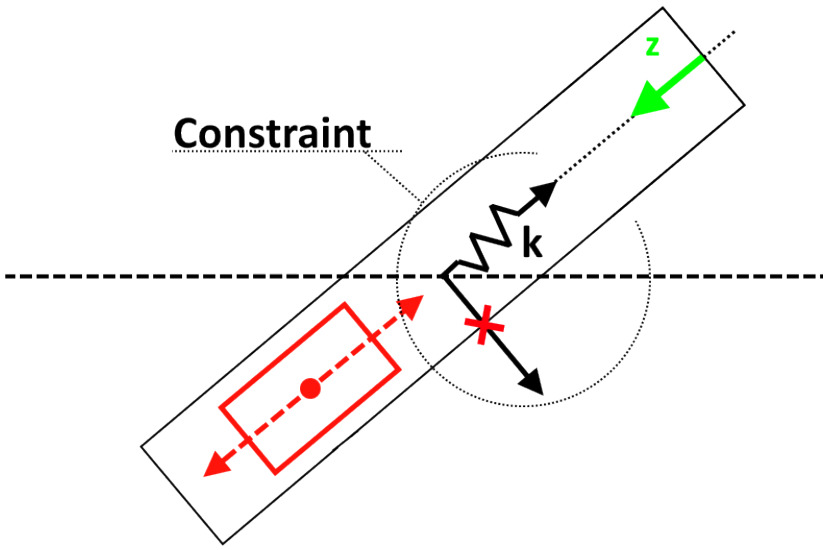

Figure F.3: Constraint that arranges the load transfer of the Mass to the monopile

The waterplug will be excited at the bottom of the monopile because of pressure differences. These pressures at the bottom at different wave frequencies are used to form Force Response Amplitude Operators and are assigned to the "Vessel" object mentioned earlier. Thise pressure differences per frequencies are obtained using the diffraction software AQWA where the submerged part of the 20m draft and 45degrees inclined monopile is modeled as a fixed structure F.4. In figure F.5 the final result of the model is shown in OrcaFlex.

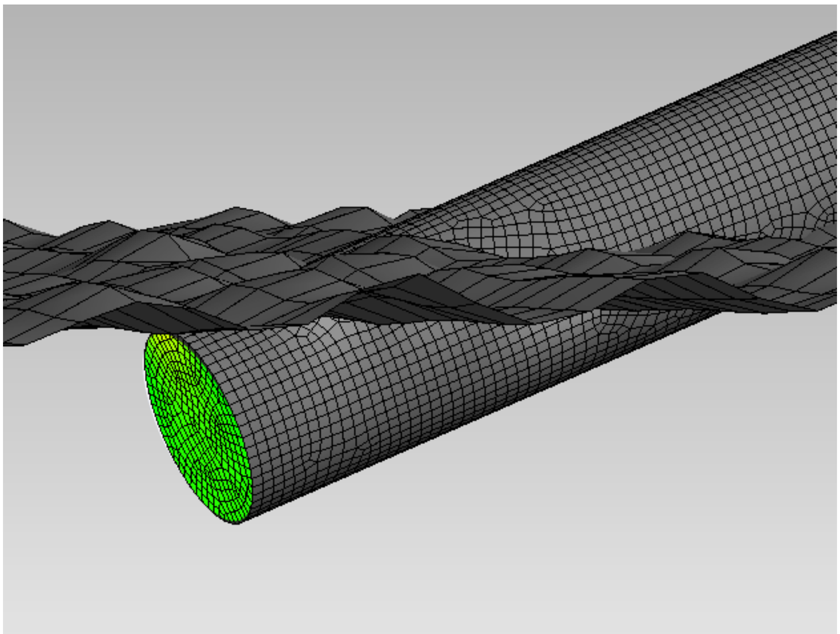

Figure F.4: Using AQWA to define pressures at the bottom of Monopile

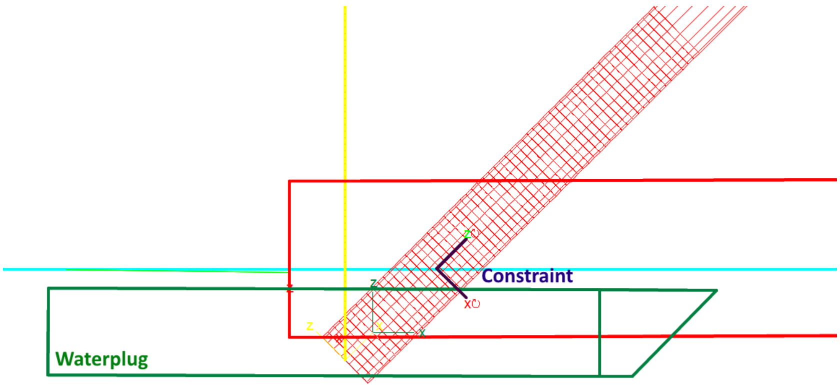

Figure F.5: PM modeled in OrcaFlex

By analyzing the force that is transferred in the longitudunal Z-direction of the monopile, it can be seen that it has a mean Force of the $cos(45) * F_{waterplug}$ where $F_{waterplug} = m_{waterplug}g$. This is the static force which should be transferred to cancel out the buoyancy of the solid cylinder. The unwanted dynamic force that is transferred in this direction has a magnitude of 100kN (figure F.6. This is small with respect to the occuring forces and moments in the complete model and so the effect of this unwanted transferred force will be minimal.

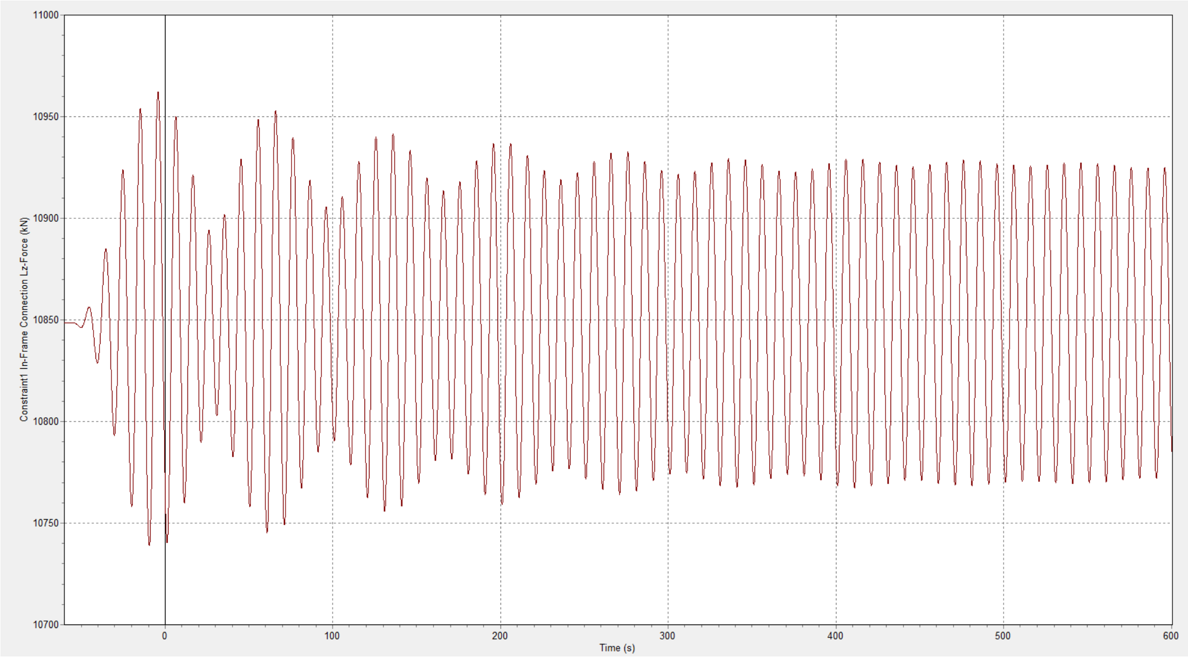

Figure F.6: "Force transferred in longitudinal Z-direction Monopile"